The American Society of Heating and Ventilating Engineers

GUIDE 1937

FOR

HEATING, VENTILATING, AIR CONDITIONING

An Instrument of Service prepared for the Profession—Containing a

Technical Data Section

of reference material on the design and specification of heating, ventilating and air conditioning systems—Based on the Transactions—the Investigations of the Research Laboratory and Co-operating Institutions—and the Practice of the Members and Friends of the Society

together with a

Manufacturers' Catalog Data Section

Containing Essential and Reliable Information Concerning Modern Equipment

also

The Roll of Membership of the Society

with

Complete Indexes

to Technical and Catalog Data Sections

Vol. 15

$5.00 Per Copy

Published Annually by

American Society of Heating and Ventilating Engineers

51 Madison Avenue ∴ New York, N. Y.

and by it
Dedicated
To the Advancement of
The Profession
and
Its Allied Industries

PREFACE TO THE 15th EDITION

IN the fifteenth edition of The American Society of Heating and Ventilating Engineers Guide, several innovations have been incorporated by the Guide Publication Committee to aid those individuals who are specializing in heating, ventilating, air conditioning and allied industries. Newly recognized standards of the profession have been carefully reviewed and added to the text section so as to maintain The Guide in its recognized role of presenting authoritative and fundamental data. Recent research developments have been summarized and added to this volume in order to maintain the original conception of The Guide formulated in 1922 of presenting authentic and current investigative results which may be of practical value to designing engineers.

The data for properties of dry and saturated air have been recalculated and tabulated based on the instantaneous specific heats of air. A new table has been added giving the true, mean and instantaneous specific heats of air over a wide range of temperatures.

The chapter has been completely rewritten which deals with the Physical and Physiological Principles of Air Conditioning covering the varied fundamental requirements which must be considered in properly designing a system to provide comfortable environments and thereby satisfy human needs.

Because of the introduction of new building materials and insulators, the tables of heat transmission coefficients have been enlarged and the text material has been rewritten and condensed with revised arrangements of fundamental formulae.

Problems of Humidification, Dehumidification and Water Cooling Equipment are discussed in a new chapter. Design tables and curves are included for estimating the number of natural draft cooling tower sections. Mechnical draft cooling tower design factors are presented with conversion factors corresponding to design wet-bulb temperatures.

New illustrations have been included in the chapter on Unit Heaters, Ventilators, Air Conditioning and Cooling Units and the text has been completely revised to incorporate the newer trends of unitary equipment.

A new chapter on Automatic Control is presented this year which concisely describes the several types of control for residential, commercial and industrial applications.

The new chapter on Sound Control has been prepared to present the subject material in a manner more suitable for interpretation by the heating and air conditioning engineer. A method is outlined for determining the proper amount of sound absorbent material to apply to a duct system based upon an experimental laboratory duct lining factor.

Another new chapter, Air Distribution, gives the latest information on air diffusion technique within enclosed spaces. The important consideration of air outlet noises is discussed in detail with charts and illustrative examples to indicate a method of grille or outlet selection.

The material on Gravity and Mechanical Warm Air Furnace Systems includes the latest results developed in the Research Residence of the *National Warm Air Heating and Air Conditioning Association.* A systematic approach to the design problems encountered in both summer and winter air conditioning is outlined in detail.

Inserted this year in the chapter on Chimneys and Draft Calculations is a method of separately determining duct and chimney friction factors as a function of the flow conditions, and specifically as a function of the Reynolds number and the relative chimney or flue pipe roughness.

The chapter on Drying Systems which was introduced last year has been improved by the addition of several new charts and tables applicable to varied systems of drier installations. Other chapters which have been revised and amplified with much new material are: Refrigeration, Cooling Methods, Air Cleaning Devices, Railway Air Conditioning, Boilers, Radiators and Gravity Convectors, Steam Heating Systems, Piping for Steam Heating Systems, Hot Water Heating Systems and Piping, and Electric Motors and Controls.

A new 16 x 20 in Bulkeley Psychrometric Chart with an additional directrix line will be found in a convenient envelope attached to the inside back cover, thus making it easily accessible for performing psychrometric calculations. All of the principal illustrations, curves and charts have been redrawn in order to present a more uniform appearance throughout the book. The drawing symbols for heating, ventilating and air conditioning applications have been revised and amplified to include the piping and fitting symbols recommended by the *American Standards Association* for drafting room practice. The Problems in Practice have been continued with new and practical solutions to expand the text material given in each of the 44 chapters.

Manufacturers recognizing the meritorious features of The Guide as an advertising medium have been responsible for an increased number of pages in the Catalog Data Section. Special efforts have been made this year to have the manufacturers present in their advertising pages of descriptive material, detailed technical data and equipment dimensions. This will enable the heating, ventilating and air conditioning engineer to have available in one volume a great deal of the essential data required for designing a complete system. The Committee is appreciative of the cooperative efforts of the manufacturers who have assisted in this program.

The continual demand for increasing copies of this handbook by colleges, universities, engineering and technical schools is further evidence of the recognition given to The Guide as the standard authority in the field of heating, ventilating and air conditioning.

It is hoped that the reader of this edition will be amply rewarded with the information contained in the 1180 pages of text and catalog data and that the conciseness and reliability of data presented will impress the user who desires a handbook for quick reference. It is again bound in a flexible blue cover with gold stamping.

In releasing this fifteenth edition of 14,500 copies, the committee hopes that The Guide 1937 will receive the same enthusiastic reception that has been accorded to previous volumes which have appeared annually since 1922.

J. H. Walker, *Chairman*

GUIDE PUBLICATION COMMITTEE

EDITORIAL ACKNOWLEDGMENT

CONTINUOUSLY for 15 years THE A.S.H.V.E. GUIDE has maintained leadership as the authoritative source of information on all phases of heating, ventilating and air conditioning and the profession with its allied industries have adopted it as their standard reference work. This achievement of world wide recognition has been possible because of the willingness of hundreds of engineers and technical experts to contribute freely from their knowledge and practical experience for the benefit of the entire profession.

The Guide Publication Committee is proud to acknowledge with gratitude and to earnestly commend the work of the following individuals who have cooperated in the production of the 1937 edition, and also to those contributors who have previously prepared a firm foundation for the addition of new and amplified material.

T. N ADLAM
O. W. ARMSPACH
C. L. ARNOLD
C. M. ASHLEY
J W. BENNER
J. L. BLACKSHAW
C. A. BOOTH
C. J. BRAATZ
PROF. A. I. BROWN
C. A. BULKELEY
W. H. CARLTON
SABIN CROCKER
D. N. CROSTHWAIT, JR.
J. M. DALLAVALLE
PHILIP DRINKER
E. V. ERICKSON
JOHN EVERETTS, JR
PROF. M. K. FAHNESTOCK
W. L. FLEISHER
W. G. FRANK
E. R. GOODRICH
W. A. GRANT
J. R. HERTZLER
C. G. HILLIER
J. H. HOLTON
F. C HOUGHTEN
PROF C. M. HUMPHREYS
H F. HUTZEL
E. F. JONES
C. F. KAYAN
R. T. KERN
R. E KEYES
PROF. V. O KNUDSEN
PROF. S. KONZO
A. W. KRAMER, JR.
PROF. A. P. KRATZ
C. H. LANKFORD
M. LAWRENCE
L. L. LEWIS
S R. LEWIS
H. A. LOCKHART
ARTHUR MCCUTCHAN
PROF. L G MILLER
H. C. MURPHY
PROF. D. W. NELSON
P. NICHOLLS
R. F. NORRIS
A. J. OFFNER
O. W OTT
J. S. PARKINSON
H. G RAPPOLT
PROF T. F. ROCKWELL
C. Z. ROSECRANS
J. O. ROSS
S. I. ROTTMAYER
PROF. F. B. ROWLEY
PROF W. M. SAWDON
PROF. L. E SEELEY
LESTER SEELIG
C G. SEGELER
A M. SELVEY
R W SHIELDS
A. L. SIMISON
D C. SIMPSON
A E. STACEY, JR.
D. J. STEWART
C. A. THINN
R J THOMPSON
W. W. TIMMIS
PROF G L TUVE
J. H. VAN ALSBURG
F. O. URBAN
A R. WALKER
B. S WEAVER
PROF C P. YAGLOU

The many readers of THE GUIDE 1937 and especially the members of the Society are deeply indebted to these engineers who have assisted in the preparation of this edition and the Guide Publication Committee wishes to pay tribute for this loyal cooperation and devotion to public service in advancing the art of heating, ventilating and air conditioning engineering. This 15th edition of THE GUIDE is hereby dedicated to the service of the profession.

GUIDE PUBLICATION COMMITTEE
J. H. WALKER, *Chairman*
ALBERT BUENGER C. H. B. HOTCHKISS
S. H. DOWNS W. H. SEVERNS
E. N. MCDONNELL, *Advisory*
JOHN JAMES, *Technical Assistant*

CODE *of* ETHICS *for* ENGINEERS

ENGINEERING work has become an increasingly important factor in the progress of civilization and in the welfare of the community. The engineering profession is held responsible for the planning, construction and operation of such work and is entitled to the position and authority which will enable it to discharge this responsibility and to render effective service to humanity

That the dignity of their chosen profession may be maintained, it is the duty of all engineers to conduct themselves according to the principles of the following Code of Ethics:

1—The engineer will carry on his professional work in a spirit of fairness to employees and contractors, fidelity to clients and employers, loyalty to his country and devotion to high ideals of courtesy and personal honor.

2—He will refrain from associating himself with or allowing the use of his name by an enterprise of questionable character.

3—He will advertise only in a dignified manner, being careful to avoid misleading statements.

4—He will regard as confidential any information obtained by him as to the business affairs and technical methods or processes of a client or employer.

5—He will inform a client or employer of any business connections, interests or affiliations which might influence his judgment or impair the disinterested quality of his services.

6—He will refrain from using any improper or questionable methods of soliciting professional work and will decline to pay or to accept commissions for securing such work

7—He will accept compensation, financial or otherwise, for a particular service, from one source only, except with the full knowledge and consent of all interested parties.

8—He will not use unfair means to win professional advancement or to injure the chances of another engineer to secure and hold employment.

9—He will cooperate in upbuilding the engineering profession by exchanging general information and experience with his fellow engineers and students of engineering and also by contributing to work of engineering societies, schools of applied science and the technical press.

10—He will interest himself in the public welfare in behalf of which he will be ready to apply his special knowledge, skill and training for the use and benefit of mankind.

CONTENTS THE A.S.H.V.E. GUIDE 1937

INDEX

THE A. S. H. V. E. GUIDE 1937

15th Edition

Technical Data Section

Chapters 1–44 and Pages 1–808

Cross Reference to Subjects in Chapters 1 to 44 Alphabetically Listed

A

B

D

E

H

I

J-K

L

M

N

O

S

T

U

V

W

Z

The
AMERICAN SOCIETY OF HEATING
AND VENTILATING ENGINEERS
GUIDE
Vol. 15 1937

Chapter 1

AIR, WATER AND STEAM

Dalton's Law, Temperatures, Air Properties, Humidity, Relative Humidity, Specific Humidity, Relation of Dew Point to Relative Humidity, Adiabatic Saturation of Air, Total Heat and Heat Content, Enthalpy, Psychrometric Chart, Properties of Water, Properties of Steam, Rate of Evaporation

AIR conditioning has for its objective the supplying and maintaining, in a room or other enclosure, of an atmosphere having a composition, temperature, humidity, and motion which will produce desired effects upon the occupants of the room or upon materials stored or handled in it.

Dry air is a mechanical mixture of gases composed, in percentage of volume, as follows[1]: nitrogen 78.03, oxygen 20.99, argon 0.94, carbon dioxide 0.03, and small amounts of hydrogen and other gases.

Atmospheric air at sea level is given in percentage by volume as: N_2 77.08, O_2 20.75, water vapor 1.2, A 0.93, CO_2 0.03 and H_2 0 01. The amount of water vapor varies greatly under different conditions and is frequently one of the most important constituents since it affects bodily comfort and greatly affects all kinds of hygroscopic materials.

DALTON'S LAW

A mixture of dry gases and water vapor, such as atmospheric air, obeys Dalton's Law of Partial Pressures; each gas or vapor in a mixture, at a given temperature, contributes to the observed pressure the same amount that it would have exerted by itself at the same temperature had no other gas or vapor been present. If p = the observed pressure of the mixture

[1]*International Critical Tables.*

and p_1, p_2, p_3, etc. = the pressure of the gases or vapors corresponding to the observed temperature, then

$$p = p_1 + p_2 + p_3, \text{ etc} \tag{1}$$

TEMPERATURES

Air is said to be saturated at a given temperature when the water vapor mixed with the air is in the dry saturated condition or, what is the equivalent, when the space occupied by the mixture holds the maximum possible weight of water *vapor* at that temperature. If the water vapor mixed with the dry air is superheated, *i.e.*, if its temperature is above the temperature of saturation for the actual water vapor partial pressure, the air is not saturated.

The starting point of most applications of thermodynamic principles to air conditioning problems is the experimental determination of the dry-bulb and wet-bulb temperatures, and sometimes the barometric pressure.

The *dry-bulb temperature* of the air is the temperature indicated by any type of thermometer not affected by the water vapor content or relative humidity of the air. The *wet-bulb temperature* is determined by a thermometer with its bulb encased in a fine mesh fabric bag moistened with clean water and whirled through the air until the thermometer assumes a steady temperature. This steady temperature is the result of a dynamic equilibrium between the rate at which heat is transferred from the air to the water on the bulb and the rate at which this heat is utilized in evaporating moisture from the bulb. The rate at which heat is transferred from the air to the water is substantially proportional to the wet-bulb depression $(t - t')$, while the rate of heat utilization in evaporation is proportional to the difference between the saturation pressure of the water at the wet-bulb temperature and the actual partial pressure of the water vapor in the air $(e' - e)$. Carrier's equation for this dynamic equilibrium is

$$\frac{e' - e}{t - t'} = \frac{B - e'}{2800 - 1.3t'} \tag{2a}$$

In the form commonly used,

$$e = e' - \frac{(B - e')(t - t')}{2800 - 1.3t'} \tag{2b}$$

where

e = actual partial pressure of water vapor in the air, inches of mercury.
e' = saturation pressure at wet-bulb temperature, inches of mercury.
B = barometric pressure, inches of mercury
t = dry-bulb temperature, degrees Fahrenheit.
t' = wet-bulb temperature, degrees Fahrenheit

Formula 2b may be used to determine the actual partial pressure of the water vapor in a dry air-water vapor mixture. Then, from Dalton's Law of Partial Pressures, Equation 1, it follows that the partial pressure of the dry air is $(B - e)$.

If a mixture of dry air and water vapor, initially unsaturated, be cooled

at constant pressure, the temperature at which condensation of the water vapor begins is called the *dew-point temperature.* Clearly the dew-point is the saturation temperature corresponding to the actual partial pressure, e, of the water vapor in the mixture.

AIR PROPERTIES

Density is variously defined as the mass per unit of volume, the weight per unit of volume, or the ratio of the mass, or weight, of a given volume of a substance to the mass, or weight, of an equal volume of some other substance such as water or air under standard conditions of temperature and pressure. The term *specific gravity* is more commonly used to express the latter relation but, when the gram is taken as the unit of mass and the cubic centimeter as the unit of volume, density and specific gravity have the same meaning. The term *specific density* is sometimes used to distinguish the weight in pounds per cubic foot; and as here used, *density* is the weight in pounds of one cubic foot of a substance.

The density of air decreases with increase in temperature when under constant pressure. The density of dry air at 70 F and under standard atmospheric pressure (29.921 in. of Hg) is approximately 0.075 lb (see Table 1), while that of a mixture of air and saturated water vapor at the same temperature and barometric pressure is only about 0.0742 lb. In the mixture the density of the dry air is 0.07307 and that of the vapor is 0.00115 lb (see Table 2).

In order to make comparisons of air volumes or velocities it is necessary to reduce the observations to a common pressure and temperature basis. The basic pressure is usually taken as 29.921 in. of Hg, but no basic temperature is universally recognized. Common temperatures for this purpose are 32 F, 60 F, 68 F, and 70 F. Since 70 F is the most commonly specified temperature to which rooms for human occupancy must be heated, it is usually understood, when no other temperature is specified, that 70 F is the basic temperature for measuring the volume or the velocity of air in heating and ventilating work.

The *specific volume* of air is the volume in cubic feet occupied by one pound of the air. Under constant pressure the specific volume varies inversely as the density and directly as the absolute temperature.

The *specific heat* of air is the number of Btu required to raise the temperature of 1 lb of air 1 F. Distinction should always be made between the instantaneous specific heat at any existent temperature and the mean specific heat, which is the average specific heat through a given temperature range. The mean specific heat is the value required in most calculations. The specific heats at constant pressure, C_p, and the specific heats, C_v, at constant volume are different. The specific heat at constant pressure is commonly used and it varies, under a pressure of one atmosphere, from a minimum at 32 F from which it increases with either increase or decrease of temperature. The value of 0.24, as the mean specific heat at constant pressure, is sufficiently accurate for use at ordinary temperatures. Values for instantaneous and mean specific heats are given in Table 3.

The *mean specific heat* of *water vapor* at constant pressure is taken as 0.45 for all general engineering computations.

Table 1 Properties of Dry Air[a]

Barometric Pressure 29.921 In. of Hg

Temperature Deg F	Weight Pounds per Cu Ft	Ratio of Volume to Volume at 70 F	Btu Absorbed by One Cu Ft Dry Air per Deg F	Cu Ft Dry Air Warmed One Deg per Btu
0	0.08633	0.8678	0.02077	48.15
10	0.08449	0.8867	0.02030	49.26
20	0.08273	0.9056	0.01986	50.35
30	0.08104	0.9245	0.01944	51.44
40	0.07942	0.9433	0.01905	52.49
50	0.07785	0.9624	0.01868	53.36
60	0.07636	0.9811	0.01832	54.44
70	0.07492	1.0000	0.01798	55.62
80	0.07353	1.0189	0.01765	56.66
90	0.07219	1.0378	0.01733	57.70
100	0.07090	1.0567	0.01702	58.75
110	0.06966	1.0755	0.01672	59.81
120	0.06845	1.0946	0.01643	60.86
130	0.06729	1.1133	0.01616	61.88
140	0.06617	1.1322	0.01589	62.93
150	0.06509	1.1510	0.01563	63.98
160	0.06403	1.1701	0.01538	65.02
180	0.06203	1.2078	0.01490	67.11
200	0.06015	1.2456	0.01446	69.24
220	0.05838	1.2832	0.01403	71.27
240	0.05671	1.3211	0.01364	73.31
260	0.05514	1.3587	0.01326	75.41
280	0.05365	1.3965	0.01291	77.46
300	0.05223	1.4344	0.01257	79.55
350	0.04901	1.5287	0.01181	84.67
400	0.04615	1.6234	0.01114	89.67
450	0.04362	1.7176	0.01054	94.87
500	0.04135	1.8119	0.01001	99.01
550	0.03930	1.9064	0.00953	104.93
600	0.03744	2.0011	0.00908	110.13
700	0.03422	2.1893	0.00833	120.05
800	0.03150	2.3784	0.00769	130.04
900	0.02911	2.5737	0.00713	140.25
1000	0.02718	2.7564	0.00668	149.70

[a]Compiled by W. H. Severns, based on the instantaneous specific heats of air. The values for the heats and for the cubic feet warmed one degree are for the temperatures stated and are not true over a temperature range of more than one degree above or below the temperatures stated.

TABLE 2 PROPERTIES OF SATURATED AIR[a]

Weights of Air, Vapor and Saturated Mixture of Air and Vapor at 29.921 In. of Hg

TEMP DEG F	Weight in a Cubic Foot of Mixture			BTU ABSORBED BY ONE CUBIC FOOT SAT AIR PER DEG F	CUBIC FEET SAT AIR WARMED ONE DEG PER BTU	SPECIFIC HEAT BTU PER POUND OF MIXTURE
	Weight of Dry Air Pounds	Weight of Vapor Pounds	Total Weight of Mixture Pounds			
0	0.08622	0.000068	0.08629	0.02078	48.12	0.2408
10	0.08431	0.000111	0.08442	0.02031	49.24	0.2406
20	0.08244	0.000177	0.08262	0.01987	50.33	0.2405
30	0.08060	0.000278	0.08088	0.01946	51.39	0.2406
40	0.07876	0.000409	0.07917	0.01908	52.41	0.2410
50	0.07692	0.000587	0.07751	0.01872	53.42	0.2415
60	0.07503	0.000828	0.07586	0.01838	54.41	0.2423
70	0.07307	0.001151	0.07422	0.01805	55.40	0.2432
80	0.07099	0.001578	0.07257	0.01775	56.34	0.2446
90	0.06877	0.002134	0.07090	0.01747	57.24	0.2464
100	0.06634	0.002851	0.06919	0.01721	58.11	0.2487
110	0.06361	0.003762	0.06737	0.01696	58.96	0.2517
120	0.06057	0.004912	0.06548	0.01675	59.70	0.2558
130	0.05712	0.006344	0.06346	0.01657	60.35	0.2611
140	0.05317	0.008116	0.06129	0.01642	60.91	0.2679
150	0.04863	0.010284	0.05891	0.01630	61.35	0.2767
160	0.04339	0.012919	0.05631	0.01624	61.58	0.2884
170	0.03733	0.016092	0.05342	0.01621	61.69	0.3034
180	0.03033	0.019888	0.05022	0.01624	61.58	0.3234
190	0.02228	0.024384	0.04666	0.01633	61.24	0.3500
200	0.01298	0.029700	0.04268	0.01649	60.64	0.3864
210	0.00230	0.035932	0.03616	0.01672	59.81	0.4624
212	0.00000	0.037286	0.03729	0.01818	55.01	0.4875

[a]Compiled by W. H. Severns, based on the instantaneous specific heats of air.

TABLE 3 SPECIFIC HEATS OF DRY AIR[a]

Constant Barometric Pressure of 29.921 In. of Hg

TEMPERATURE DEG F	INSTANTANEOUS OR TRUE SPECIFIC HEAT	TEMPERATURE RANGE DEG F	MEAN SPECIFIC HEAT
−301.0	0.2520	32 to 212	0.2401
−108.4	0.2430	32 to 392	0.2411
32.0	0.2399	32 to 752	0.2420
212.0	0.2403	32 to 1112	0.2430
392.0	0.2413	....	....
752.0	0.2430	....	....
1112.0	0.2470	....	....

[a]Compiled by W. H. Severns, based on data given in the *International Critical Tables*.

TABLE 4. WEIGHTS OF SATURATED AND PARTLY SATURATED AIR FOR VARIOUS BAROMETRIC AND HYGROMETRIC CONDITIONS[a, b]

Pounds per Cubic Foot

Dry-Bulb Temperature Deg F	Barometric Pressure—Inches: 26			27			28			29			30			Approx Average Increase in Weight per Deg Wet-Bulb Depression
	Wt per Cu Ft Saturated Air	Decr's Wt per Deg Inc Dry-Bulb	Incr's Wt. per 0.1" Rise in Bar	Wt per Cu Ft Saturated Air	Decr's Wt per Deg Inc. Dry-Bulb	Incr's Wt per 0.1" Rise in Bar.	Wt per Cu Ft Saturated Air	Decr's Wt per Deg Inc Dry-Bulb	Incr's Wt. per 0.1" Rise in Bar	Wt per Cu Ft Saturated Air	Decr's Wt per Deg Inc Dry-Bulb	Incr's Wt per 0.1" Rise in Bar.	Wt per Cu Ft Saturated Air	Decr's Wt per Deg Inc Dry-Bulb	Incr's Wt per 0.1" Rise in Bar	
0	0.07500	0.00016	0.00029	0.07788	0.00016	0.00029	0.08077	0.00017	0.00029	0.08365	0.00018	0.00029	0.08654	0.00019	0.00029	0.000015
10	0.07338	0.00016	0.00028	0.07620	0.00016	0.00028	0.07903	0.00017	0.00028	0.08185	0.00018	0.00028	0.08468	0.00018	0.00028	0.000016
20	0.07180	0.00016	0.00028	0.07456	0.00016	0.00028	0.07733	0.00017	0.00028	0.08009	0.00018	0.00028	0.08286	0.00018	0.00028	0.000017
30	0.07027	0.00015	0.00027	0.07297	0.00016	0.00027	0.07569	0.00016	0.00027	0.07839	0.00017	0.00027	0.08110	0.00017	0.00027	0.000019
40	0.06879	0.00015	0.00026	0.07143	0.00015	0.00027	0.07409	0.00016	0.00027	0.07675	0.00016	0.00027	0.07942	0.00017	0.00027	0.000021
50	0.06732	0.00015	0.00026	0.06992	0.00015	0.00026	0.07252	0.00016	0.00026	0.07512	0.00016	0.00026	0.07773	0.00016	0.00026	0.000023
60	0.06588	0.00015	0.00026	0.06843	0.00015	0.00026	0.07098	0.00015	0.00026	0.07353	0.00016	0.00026	0.07609	0.00016	0.00026	0.000026
70	0.06442	0.00015	0.00025	0.06692	0.00015	0.00025	0.06943	0.00015	0.00025	0.07193	0.00016	0.00025	0.07440	0.00016	0.00025	0.000029
80	0.06297	0.00015	0.00025	0.06542	0.00015	0.00025	0.06789	0.00015	0.00025	0.07034	0.00015	0.00025	0.07280	0.00016	0.00025	0.000034
90	0.06146	0.00015	0.00024	0.06388	0.00016	0.00024	0.06629	0.00016	0.00024	0.06870	0.00016	0.00024	0.07112	0.00017	0.00024	0.000039
100	0.05991	0.00016	0.00024	0.06228	0.00016	0.00024	0.06465	0.00016	0.00024	0.06703	0.00017	0.00024	0.06939	0.00018	0.00024	0.000044
110	0.05828	0.00016	0.00023	0.06060	0.00017	0.00023	0.06293	0.00017	0.00023	0.06526	0.00018	0.00023	0.06759	0.00019	0.00023	0.000051
120	0.05653	0.00018	0.00023	0.05882	0.00018	0.00023	0.06111	0.00018	0.00023	0.06339	0.00019	0.00023	0.06569	0.00020	0.00023	0.000059
130	0.05467	0.00019	0.00023	0.05692	0.00019	0.00023	0.05917	0.00019	0.00023	0.06142	0.00020	0.00023	0.06367	0.00022	0.00023	0.000068
140	0.05262	0.00021	0.00022	0.05483	0.00021	0.00022	0.05704	0.00021	0.00022	0.05925	0.00022	0.00022	0.06147	0.00024	0.00022	0.000078
150	0.05036	0.00023	0.00022	0.05253	0.00023	0.00022	0.05471	0.00023	0.00022	0.05689	0.00024	0.00022	0.05906	0.00026	0.00022	0.000090
160	0.04788	0.00025	0.00022	0.05001	0.00025	0.00022	0.05216	0.00026	0.00021	0.05430	0.00026	0.00021	0.05644	0.00029	0.00021	0.000103
170	0.04509	0.00028	0.00021	0.04720	0.00028	0.00021	0.04931	0.00029	0.00021	0.05141	0.00031	0.00021	0.05352	0.00033	0.00021	0.000118
180	0.04197	0.00031	0.00021	0.04404	0.00031	0.00021	0.04611	0.00032	0.00021	0.04818	0.00034	0.00021	0.05026	0.00036	0.00021	0.000134
190	0.03845	0.00035	0.00021	0.04049	0.00036	0.00021	0.04253	0.00036	0.00021	0.04457	0.00037	0.00021	0.04662	0.00038	0.00021	0.000153
200	0.03449	0.00040	0.00020	0.03650	0.00040	0.00020	0.03851	0.00040	0.00020	0.04052	0.00041	0.00020	0.04254	0.00041	0.00020	0.000173

[a]From *Fan Engineering*.

[b]A convenient and accurate chart for quickly determining the weight of air under any condition of dry-bulb, wet-bulb, and pressure is *A Chart for Determining the Weight of Moist Air in Pounds per Cubic Foot*, by John E Younger. Published in *Mechanical Engineering*, June, 1926.

Table 4 is intended to aid in determining the density of moist air, taking into account its temperature, pressure, and moisture content.

Example 1 To show the use of Table 4 Given air at 83 F dry-bulb and 68 F wet-bulb (or a depression of 15 deg) with a barometric pressure of 29 40 in of mercury. What will be the weight of this air in pounds per cubic foot?

Solution. From Table 4 the weight of saturated air at 80 F and 29 00 in barometer is found to be 0 07034 lb per cubic foot There is a decrease of 0.00015 lb per degree dry-bulb temperature above 80 F. There is an increase of 0 00025 lb for each 0.1 in above 29.00 in From the last column of Table 4 it is found that there is an increase of approximately 0.000035 lb per degree wet-bulb depression when the dry-bulb is 83 F. Tabulating the items:

0.07034	= weight of saturated air at 80 F and 29 00 bar
− 0.00045	= decrement for 3 deg dry-bulb, 3 × 0 00015.
+ 0.00100	= increment for 0.4 in. bar, 4 × 0.00025.
+ 0.00053	= increment for 15 deg wet-bulb depression, 15 × 0.000035.
0 07142	= weight in pounds per cubic foot of air at 83 F dry-bulb, 68 F wet-bulb, 29.40 in bar.

It is usual to assume that dry air, moist air, and the water vapor in the air follow the laws of perfect gases. This assumption while not absolutely true, especially with saturated vapor at temperatures much above 140 F, is sufficiently accurate for practical purposes and it greatly simplifies computations.

Boyle's Law refers to the relation between the pressure and volume of a gas, and may be stated as follows: *With temperature constant, the volume of a given weight of gas varies inversely as its absolute pressure.* Hence, if P_1 and P_2 represent the initial and final absolute pressures, and V_1 and V_2 represent corresponding volumes of the same mass, say one pound of gas, then $\frac{V_1}{V_2} = \frac{P_2}{P_1}$, or $P_1 V_1 = P_2 V_2$, but since $P_1 V_1$ for any given case is a definite constant quantity, it follows that the product of the absolute pressure and volume of a gas is a constant, or $PV = C$, when T is kept constant. Any change in the pressure and volume of a gas at constant temperature is called an *isothermal change.*

Charles' Law refers to the relation among pressure, volume, and temperature of a gas and may be stated as follows *The volume of a given weight of gas varies directly as the absolute temperature at constant pressure, and the pressure varies directly as the absolute temperature at constant volume.* Hence, when heat is added at constant volume, V_c, the resulting equation is $\frac{P_2}{P_1} = \frac{T_2}{T_1}$, or, for the same temperature range at constant pressure, P_c, the relation is $\frac{V_2}{V_1} = \frac{T_2}{T_1}$.

In general, for any weight of gas, W, since volume is proportional to weight, the relation among P, V, and T is

$$PV = WRT \tag{3}$$

where

P = the absolute pressure of the gas, pounds per square foot.

V = the volume of the weight W, cubic feet.

W = the weight of the gas, pounds
R = a constant depending on the nature of the gas. The average value of R for air is 53.34.
T = the absolute temperature, degrees Fahrenheit.

This is the characteristic equation for a perfect gas, and while no gases are perfect in this sense, they conform so nearly that Equation 3 will apply to most engineering computations.

HUMIDITY

Humidity is the water vapor mixed with dry air in the atmosphere. *Absolute humidity* has a multiplicity of meanings, but usually the term refers to the weight of water vapor per unit volume of space occupied, expressed in grains or pounds per cubic foot. With this meaning, absolute humidity is nothing but the actual density of the water vapor in the mixture and might better be so called. A study of Keenan's Steam Tables[2] indicates that water vapor, either saturated or super-heated, at partial pressures lower than 4 in. of mercury may be treated as a gas with a gas constant R of 1.21 in the characteristic equation of the gas $pV = wR\ (t + 460)$. Within such limits, the density (d) of water vapor is

$$d = \frac{w}{V} = \frac{e}{1.21\ (t + 460)} \text{ (pounds per cubic foot)} \tag{4a}$$

$$= \frac{5785\ e}{t + 460} \text{ (grains per cubic foot)} \tag{4b}$$

where

e = actual partial pressure of vapor, inches of mercury
t = dry-bulb temperature, degrees Fahrenheit.

Specific Humidity

It simplifies many problems which deal with mixtures of dry air and water vapor to express the weight or the mass of the vapor in terms of the weight or the mass of dry air. If the weight of the water vapor in a mixture be divided by the weight of the dry air, and the weight of dry air be made unity, we have an expression of the weight of water vapor carried by a unit weight of dry air. This relation has no generally accepted name. It has been variously called: mixing ratio, proportionate humidity, mass or density ratio, absolute humidity, and specific humidity. Of all these terms *specific humidity* is the most suggestive of the meaning which it is desired to express and it has found considerable use in this sense even though it is defined in *International Critical Tables* as the ratio of the mass of vapor to the total mass. It will be understood here that *specific humidity* refers to the weight of water vapor carried by one pound of dry air.

The gas constant for dry air, when the partial pressure of the air is expressed in inches of Hg, is 0.753; so that the specific humidity, if represented by W, is

[2]Published by *American Society of Mechanical Engineers*, see abstract in Table 8.

$$W = \frac{e}{1.21\ (t + 460)} \div \frac{B\text{-}e}{0.753\ (t + 460)}$$

$$= 0.622 \left(\frac{e}{B\text{-}e}\right) \text{ (pounds)} \qquad (5a)$$

$$= 4354 \left(\frac{e}{B\text{-}e}\right) \text{ (grains)} \qquad (5b)$$

where

e = actual partial pressure of vapor, inches of mercury.
B = total pressure of mixture (barometric pressure), inches of mercury.

Relative Humidity

Relative humidity (Φ) is either the ratio of the actual partial pressure, e, of the water vapor in the air to the saturation pressure, e_t, at the dry-bulb temperature, or the ratio of the actual density, d, of the vapor to the density of saturated vapor, d_t, at the dry-bulb temperature. That is:

$$\Phi = \frac{e}{e_t} = \frac{d}{d_t} \qquad (6)$$

The relative humidity of a given mixture at a given temperature is not the same as the specific humidity, W, of the mixture divided by the specific humidity, W_t, of saturated vapor at the same temperature, for from Equations 5a and 6

$$\frac{W}{W_t} = 0.622 \left(\frac{\Phi\ e_t}{B - \Phi\ e_t}\right) \div 0.622 \left(\frac{e_t}{B\text{-}e_t}\right) = \frac{\Phi\ (B\text{-}e_t)}{B - \Phi e_t} \qquad (7)$$

The specific humidity of an unsaturated air-vapor mixture cannot, therefore, be accurately found by multiplying the specific humidity of saturated vapor by its relative humidity; although the error is usually small especially when the relative humidity is high.

With a relative humidity of 100 per cent, the dry-bulb, wet-bulb, and dew-point temperatures are equal. With a relative humidity less than 100 per cent, the dry-bulb exceeds the wet-bulb, and the wet-bulb exceeds the dew-point temperature.

RELATION OF DEW POINT TO RELATIVE HUMIDITY

A peculiar relationship exists between the dew point and the relative humidity and this is found most useful in air conditioning work. This relationship is, that for a fixed relative humidity there is substantially a constant difference between the dew point and the dry-bulb temperature over a considerable temperature range. Table 5, giving the dry-bulb and dew-point temperatures and the dew-point differentials for 50 per cent relative humidity, illustrates this relationship clearly.

TABLE 5. TEMPERATURES FOR 50 PER CENT RELATIVE HUMIDITY

Dry-bulb temperature	65.0	70.0	75.0	80 0	85.0	90.0
Dew-point temperature	45.8	50.5	55 25	59.75	64.25	68 75
Difference between dew-point and dry-bulb temperature	19.2	19.5	19.75	20.25	20 75	21.25

It will be seen from an inspection of this table that the difference between the dew-point temperature and the room temperature is approximately 20 deg throughout this range of dry-bulb temperatures or, to be more exact, the differential increases only 10 per cent for a range of practically 25 deg.

This principle holds true for other humidities and is due to the fact that the pressure of the water vapor practically doubles for every 20 deg through this range.

The approximate relative humidity for any difference between dew-point and dry-bulb temperature may be expressed in per cent as:

$$\frac{100}{2^{\frac{t-t_1}{20}}} \tag{8}$$

where

t_1 = dew-point temperature.

This principle is very useful in determining the available cooling effect obtainable with saturated air when a desired relative humidity is to be maintained in a room, even though there may be a wide variation in room temperature. This problem is one which applies to certain industrial conditions, such as those in cotton mills and tobacco factories, where relatively high humidities are carried and where one of the principal problems is to remove the heat generated by the machinery. It also permits the use of a differential thermostat, responsive to both the room temperature and the dew-point temperature, to control the relative humidity in the room.

Table 6 gives, for different temperatures, the density of saturated vapor, d_t, the weight of saturated vapor mixed with 1 lb of dry air, W_t, (at a relative humidity of 100 per cent and a barometric pressure, B, of 29.92 in. of mercury), the specific volume of dry air, and the volume of an air-vapor mixture containing 1 lb of dry air (at a relative humidity of 100 per cent and a pressure of 29.92 in. of mercury). The preceding equations or the data from Table 6 may be conveniently used in solving the following typical problems:

Example 2. Humidifying and Heating. Air is to be maintained at 70 F with a relative humidity of 40 per cent (Φ = 0.4) when the outside air is at 0 F and 70 per cent relative humidity (Φ = 0.7) and a barometric pressure, B, of 29.92 in. of mercury. Find the weight of water vapor added to each pound of dry air and the dew-point temperature of the humidified air.

Solution. From Equation 5a and Table 6,

$$W_1 = 0.622 \left(\frac{0.7 \times 0.03773}{29.92 - 0.0264}\right) = 0.000548 \text{ lb per pound of dry air.}$$

$$W_2 = 0.622 \left(\frac{0.4 \times 0.7386}{29.92 - 0.295}\right) = 0.00618 \text{ lb per pound of dry air}$$

The water vapor added per pound of dry air must be ($W_2 - W_1$) or 0.005632 lb. By inspection of Table 6, W_t = 0.00618 at 44.5 F, so this is the dew-point temperature of the humidified air.

An approximation of the same result from Table 6 is

W_1 = 0.7 × 0.0007852 = 0 00054964 lb per pound of dry air.
W_2 = 0.4 × 0.01574 = 0.006296 lb per pound of dry air.

The water vapor added per pound of dry air is approximately 0.00574636 lb and the dew-point temperature is approximately 45 F. The degree of approximation is evident.

Example 3. Dehumidifying and Cooling. Air with a dry-bulb temperature of 84 F, a wet-bulb of 70 F, or a relative humidity of 50 per cent ($\Phi = 0.5$), and a barometric pressure, B, of 29 92 in. of mercury is to be cooled to 54 F. Find the dew-point temperature of the entering air and the weight of vapor condensed per pound of dry air.

Solution. From Equation 5a and Table 6,

$$W_1 = 0.622 \left(\frac{0\,5 \times 1.1752}{29\,92 - 0\,5876}\right) = 0\,01248 \text{ lb per pound of dry air.}$$

$$W_2 = 0.622 \left(\frac{0\,42003}{29\,92 - 0\,42003}\right) = 0\,00887 \text{ lb per pound of dry air.}$$

Since $W_1 = W_t$ when $t = 63\,4$ F, this is the dew-point temperature of the entering air. The weight of vapor condensed is $(W_1 - W_2)$ or 0.00361 lb per pound of dry air.

An approximate result is

$W_1 = 0.5 \times 0.02543 = 0.012715$ lb per pound of dry air.
$W_2 = 1 \times 0.008856 = 0\,008856$ lb per pound of dry air, since the exit air is saturated.

Since $W_1 = W_t$ at $t = 64$ F, this is the dew-point temperature of the entering air. The weight of vapor condensed is 0.003859 lb per pound of dry air. The degree of approximation is again evident.

ADIABATIC SATURATION OF AIR

The process of adiabatic saturation of air is of considerable importance in air conditioning. Suppose that 1 lb of dry air, initially unsaturated but carrying W lb of water vapor with a dry-bulb temperature, t, and a wet-bulb temperature, t', be made to pass through a tunnel containing an exposed water surface. Further assume the tunnel to be completely insulated, thermally, so that the only heat transfer possible is that between the air and water As the air passes over the water surface, it will gradually pick up water vapor and will approach saturation *at the initial wet-bulb temperature of the air*, if the water be supplied at this wet-bulb temperature. During the process of adiabatic saturation, then, the dry-bulb temperature of the air drops to the wet-bulb temperature as a limit, the wet-bulb temperature remains substantially constant, and the weight of water vapor associated with each pound of dry air increases to $W_{t'}$, as a limit, *where* $W_{t'}$ is the weight of saturated vapor per pound of dry air for saturation at the wet-bulb temperature.

Example 4. If air with a dry-bulb of 85 F and a wet-bulb of 70 F be saturated adiabatically by spraying with recirculated water, what will be the final temperature and the vapor content of the air?

Solution. The final temperature will be equal to the initial wet-bulb temperature or 70 F, and since the air is saturated at this temperature, from Table 6, $W = 0.01574$ lb per pound of dry air.

In the adiabatic saturation process, since the heat given up by the dry air and associated vapor in cooling to the wet-bulb temperature is utilized in evaporation of water at the wet-bulb temperature, W. H. Carrier has pointed out[3] that the equation for the process of *adiabatic saturation*, and hence for a process of *constant wet-bulb temperature*, is:

[3] *A.S M E Transactions*, Vol 33, 1911, p 1005.

TABLE 6. PROPERTIES OF SATURATED WATER VAPOR WITH AIR AT LOW TEMPERATURES[a] (PART I)

Temp., F	Pressure of Saturated Vapor × 10⁻⁵		Weight of Saturated Vapor				Volume in Cu Ft Barometer, 29.92 In. Hg		Heat Content per Lb		
	In. of Hg	Lb per Sq In	per Cu Ft: Pounds × 10⁻⁵	per Cu Ft: Grains	per lb of Dry Air: Pounds × 10⁻⁵	per lb of Dry Air: Grains	of 1 lb of Dry Air	of 1 lb of Dry Air + Vapor to Saturate it	Dry Air 0 F Datum	Vapor 32 F Datum	Dry Air with Vapor to Saturate it
−130	0.276	0.1356	0.000693	0.000049	0.005738	0.00040	8.31	8.31	−31.71	1000.7	−31.71
−129	.306	.1503	.000766	.000054	.006362	.00045	8.33	8.33	−31.46	1001.2	−31.46
−128	.338	.1660	.000843	.000059	.007027	.00049	8.36	8.36	−31.21	1001.6	−31.21
−127	.373	.1832	.000928	.000065	.007755	.00054	8.38	8.38	−30.96	1002.1	−30.96
−126	.411	.2019	.001019	.000071	.008545	.00060	8.41	8.41	−30.71	1002.5	−30.71
−125	0.455	0.2235	0.001125	0.000079	0.009459	0.00066	8.43	8.43	−30.46	1003.0	−30.46
−124	.499	.2451	.001230	.000086	.01037	.00073	8.46	8.46	−30.21	1003.4	−30.21
−123	.542	.2682	.001332	.000093	.01127	.00079	8.48	8.48	−29.96	1003.9	−29.96
−122	.604	.2967	.001480	.00010	.01256	.00088	8.51	8.51	−29.72	1004.3	−29.72
−121	.669	.3286	.001635	.00012	.01391	.00097	8.53	8.53	−29.47	1004.8	−29.47
−120	0.735	0.3610	0.001791	0.00013	0.01528	0.00107	8.56	8.56	−29.22	1005.2	−29.22
−119	.805	.3954	.001956	.00014	.01674	.00117	8.58	8.58	−28.97	1005.7	−28.97
−118	.892	.4382	.002161	.00015	.01854	.00130	8.61	8.61	−28.72	1006.1	−28.72
−117	.989	.4858	.002388	.00017	.02056	.00144	8.63	8.63	−28.47	1006.6	−28.47
−116	1.098	.5393	.002644	.00019	.02283	.00160	8.66	8.66	−28.23	1007.0	−28.23
−115	1.208	0.5934	0.002900	0.00020	0.02511	0.00176	8.68	8.68	−27.98	1007.5	−27.98
−114	1.317	.6469	.003153	.00022	.02738	.00192	8.71	8.71	−27.73	1007.9	−27.73
−113	1.444	.7093	.003447	.00024	.03002	.00210	8.73	8.73	−27.48	1008.4	−27.48
−112	1.575	.7736	.003749	.00026	.03274	.00229	8.76	8.76	−27.23	1008.8	−27.23
−111	1.728	.8488	.004101	.00029	.03593	.00252	8.78	8.78	−26.99	1009.3	−26.99
−110	1.889	0.9279	0.004471	0.00031	0.03927	0.00275	8.81	8.81	−26.74	1009.7	−26.74
−109	2.087	1.0251	.004925	.00035	.04339	.00304	8.83	8.83	−26.49	1010.2	−26.49
−108	2.292	1.1258	.005393	.00038	.04765	.00334	8.86	8.86	−26.24	1010.6	−26.24
−107	2.511	1.2334	.005892	.00041	.05220	.00365	8.89	8.89	−26.00	1011.1	−26.00
−106	2.742	1.3469	.006415	.00045	.05701	.00399	8.91	8.91	−25.75	1011.5	−25.75
−105	2.983	1.4652	0.006960	0.00049	0.06202	0.00434	8.94	8.94	−25.50	1012.0	−25.50
−104	3.258	1.6003	.007580	.00053	.06773	.00474	8.96	8.96	−25.26	1012.4	−25.26
−103	3.543	1.7403	.008219	.00058	.07366	.00516	8.99	8.99	−25.01	1012.9	−25.01
−102	3.872	1.9019	.008958	.00063	.08050	.00564	9.01	9.01	−24.76	1013.3	−24.76
−101	4.213	2.0694	.009719	.00068	.08759	.00613	9.04	9.04	−24.51	1013.8	−24.51
−100	4.607	2.2630	0.010599	0.00074	0.09578	0.00705	9.06	9.06	−24.27	1014.2	−24.27
−99	5.018	2.4637	.011512	.00081	.1043	.00730	9.09	9.09	−24.02	1014.7	−24.02
−98	5.455	2.6784	.012480	.00087	.1134	.00794	9.11	9.11	−23.78	1015.1	−23.78
−97	5.946	2.9196	.013566	.00095	.1236	.00865	9.14	9.14	−23.53	1015.6	−23.53
−96	6.470	3.1781	.014721	.00103	.1345	.00942	9.16	9.16	−23.28	1016.0	−23.28

[a]Compiled by W. M. Sawdon vapor pressures converted from *International Critical Tables*.

TABLE 6. PROPERTIES OF SATURATED WATER VAPOR WITH AIR AT LOW TEMPERATURES[a] (PART I, CONTINUED)

Temp, F	Pressure of Saturated Vapor × 10⁻⁵		Weight of Saturated Vapor				Volume in Cu Ft Barometer, 29 92 In H		Heat Content per Lb		
	In of Hg	Lb per Sq In	per Cu Ft		per lb of Dry Air		of 1 lb of Dry Air	of 1 lb of Dry Air + Vapor to Saturate it	Dry Air 0 F Datum	Vapor 32 F Datum	Dry Air with Vapor to Saturate it
			Pounds × 10⁻⁵	Grains	Pounds × 10⁻⁵	Grains					
−95	7.047	3 4604	0 015990	0 00112	0 1465	0 01026	9 19	9.19	−23 04	1016 5	−23 04
−94	7 638	3 7507	.017284	00121	1588	01112	9 21	9 21	−22 79	1016 9	−22 79
−93	8 316	4 0837	018767	00131	1729	01210	9 23	9 23	−22 55	1017 4	−22 55
−92	9 017	4 4281	020292	00142	1875	.01312	9 26	9 26	−22 30	1017 8	−22 30
−91	9 806	4 8156	.022009	00154	2039	01427	9.29	9 29	−22 05	1018 3	−22 05
−90	10 64	5 2264	0 023817	0 00167	0 2212	0 01548	9 31	9 31	−21 81	1018 7	−21 81
−89	11 53	5 6635	025738	.00180	.2397	01678	9.34	9 34	−21 56	1019 2	−21 56
−88	12 51	6 1449	027851	00195	.2601	.01821	9.36	9 36	−21 32	1019 6	−21 32
−87	13 53	6 6459	030041	.00210	2813	01969	9 39	9 39	−21 07	1020 1	−21 07
−86	14 69	7 2157	032530	00328	3054	02138	9 41	9 41	−20 83	1020 5	−20 83
−85	15 87	7 7953	0 035049	0 00245	0 3299	0 02309	9 44	9 44	−20 58	1021 0	−20 58
−84	17 20	8 4486	037885	.00265	3576	02503	9 46	9 46	−20 34	1021 4	−20 34
−83	18 58	9 1265	040817	00286	3863	02704	9 49	9 49	−20 09	1021 9	−20 09
−82	20 10	9 8731	044037	00308	4179	.02925	9 51	9 51	−19 84	1022 3	−19 84
−81	21 72	10 669	047463	00332	4516	03161	9 54	9 54	−19 60	1022 8	−19 60
−80	23 47	11 517	0 051151	0 00358	0 4879	0 03415	9 56	9 56	−19 36	1023 2	−19 36
−79	25 34	12 436	055082	00386	5268	03688	9 59	9 59	−19 11	1023 7	−19 10
−78	27 29	13 394	059165	00414	5674	.03972	9 61	9 61	−18 87	1024 1	−18 86
−77	29 52	14 489	063831	.00447	6137	04296	9 64	9 64	−18 62	1024 6	−18 61
−76	31 81	15.614	.068605	00480	6613	04629	9 66	9 66	−18 38	1025 0	−18 37
−75	34 37	16 883	0 073933	0 00518	0 7146	0.05002	9 69	9 69	−18 13	1025 5	−18 12
−74	37.01	18 179	079405	00556	.7694	05386	9 72	9 72	−17 89	1025 9	−17 88
−73	39 96	19 628	085510	00599	8308	05816	9 74	9 74	−17 64	1026 4	−17 63
−72	43 04	21 141	.091865	00643	8948	06264	9 77	9 77	−17 40	1026 8	−17 39
−71	46 33	22 757	.098632	.00690	9632	.06742	9 79	9 79	−17 16	1027 3	−17 15
−70	49 87	24.496	0 10590	0 00741	1 037	0 07259	9 82	9.82	−16 91	1027 7	−16 90
−69	53.59	26 323	.11350	.00795	1 114	.07798	9 84	9 84	−16 67	1028 2	−16 66
−68	57 65	28 318	.12179	00853	1.199	08393	9 87	9 87	−16 42	1028 6	−16 41
−67	61.81	30 361	13024	.00911	1 285	08995	9 80	9 89	−16 18	1029 1	−16 17
−66	66 41	32 621	13959	00977	1.381	.09667	9 92	9 92	−15 94	1029 5	−15 93
−65	71 17	34 959	0 14922	0 01044	1.480	0 10360	9 94	9.94	−15.69	1030 0	−15 67
−64	76.64	37 646	16028	.01122	1.593	11151	9.97	9.97	−15 45	1030.4	−15 43
−63	82 28	40 416	.17164	.01201	1 711	.11977	9 99	9 99	−15 21	1030.9	−15 19
−62	88 19	43 319	.18350	.01285	1.833	12831	10 02	10 02	−14 96	1031 3	−14 94
−61	94.62	46 477	.19638	.01375	1 967	.13769	10 04	10.04	−14 72	1031 8	−14.70

[a]Compiled by W M Sawdon, vapor pressures converted from *International Critical Tables*

TABLE 6. PROPERTIES OF SATURATED WATER VAPOR WITH AIR AT LOW TEMPERATURES[a] (PART I, CONTINUED)

TEMP, F	Pressure of Saturated Vapor × 10⁻⁵		Weight of Saturated Vapor				Volume in Cu Ft Barometer, 29 92 In. Hg		Heat Content per Lb		
	In. of Hg	Lb per Sq In.	per Cu Ft: Pounds × 10⁻⁵	per Cu Ft: Grains	per lb of Dry Air: Pounds × 10⁻⁵	per lb of Dry Air: Grains	of 1 lb of Dry Air	of 1 lb of Dry Air + Vapor to Saturate it	Dry Air 0 F Datum	Vapor 32 F Datum	Dry Air with Vapor to Saturate it
−60	101 4	49.808	0 20993	0 01470	2 108	0 14756	10 07	10 07	−14.48	1032 2	−14.46
−59	108 8	53 443	22469	01573	2 262	15834	10 09	10 09	−14 23	1032 7	−14 21
−58	116.3	57 127	.23958	.01677	2 418	16926	10.12	10 12	−13 99	1033 1	−13 97
−57	124.8	61.302	25645	01795	2.595	.18165	10.14	10 14	−13 75	1033 6	−13 72
−56	133 4	65.526	27344	01914	2 773	.19411	10 17	10 17	−13 50	1034 0	−13 47
−55	143.0	70 242	0 29239	0 02047	2 973	0 20811	10 19	10 19	−13 26	1034 5	−13.23
−54	153 0	75 154	31207	.02184	3 181	22267	10 22	10 22	−13 02	1034 9	−12 99
−53	163.5	80.311	.33267	.02329	3 399	23793	10.24	10 24	−12 78	1035 4	−12 74
−52	174 9	85.911	35499	02485	3 636	25452	10 27	10 27	−12 53	1035 8	−12 49
−51	187.0	91 854	.37862	02650	3.888	.27216	10 29	10 29	−12 29	1036 3	−12 25
−50	199 9	98.191	0 40376	0 02826	4 156	0 29092	10 32	10 32	−12 05	1036 7	−12 01
−49	213.0	104.63	42917	03004	4.428	30996	10 34	10 34	−11 81	1037 2	−11 76
−48	227.9	111 94	.45808	03207	4 738	.33166	10 37	10 37	−11 57	1037 6	−11 52
−47	243 1	119.41	48744	03412	5 054	35378	10.40	10 40	−11 32	1038 1	−11 27
−46	259.5	127.47	51905	.03633	5 395	37765	10 42	10 42	−11 08	1038 5	−11 02
−45	276.7	135 92	0 55213	0.03865	5.753	0.40271	10 45	10 45	−10 84	1039 0	−10 78
−44	295.0	144.90	.58722	.04111	6 133	42931	10 47	10 47	−10 60	1039 4	−10 54
−43	314.7	154 58	.62493	.04375	6 543	.45801	10 50	10 50	−10 35	1039 9	−10 28
−42	335.3	164.70	.66426	.04650	6.971	.48797	10 52	10 52	−10 11	1040 3	−10 04
−41	357.6	175 65	70672	.04947	7 435	52045	10 55	10.55	−9.872	1040 8	−9 795
−40	380 3	186.80	0 7498	0 05249	7 907	0 55349	10 57	10 57	−9 629	1041 2	−9 547
−39	405.5	199.18	7976	05583	8 431	.59017	10 60	10 60	−9 388	1041 7	−9 300
−38	431.2	211.81	.8461	.05922	8.965	.62755	10.62	10 62	−9 146	1042 1	−9 053
−37	459.2	225.56	.8989	.06292	9 548	.66836	10 65	10 65	−8.905	1042 6	−8 805
−36	488.4	239.90	9538	06677	10 16	.71120	10 67	10.67	−8 663	1043 0	−8 557
−35	519.5	255.18	1.0122	0.07085	10.80	0.75600	10 69	10.69	−8 422	1043 5	−8 309
−34	552.4	271 34	1.0738	.07517	11.49	.80430	10.72	10.72	−8 180	1043.9	−8 060
−33	586.5	288.09	1.1374	.07962	12.30	85400	10.75	10 75	−7 939	1044 4	−7 812
−32	623.7	306.36	1.2067	.08447	12.97	.90790	10.77	10.77	−7 698	1044 8	−7 562
−31	661.8	325.08	1 2774	.08942	13.76	96320	10 80	10 80	−7 457	1045 3	−7 313
−30	701.0	344.33	1.3499	0.09449	14.58	1.0206	10.82	10.82	−7.216	1045 7	−7.064
−29	742.2	364.57	1.4260	.09982	15.43	1.0801	10.85	10.85	−6.975	1046 2	−6.814
−28	791.2	388.64	1.5166	.10616	16.45	1.1515	10.87	10.87	−6.734	1046 6	−6 562
−27	841.0	413.10	1.6083	.11258	17.49	1.2243	10 90	10 90	−6.493	1047 1	−6 310
−26	892.1	438.20	1.7021	.11914	18.55	1.2985	10.92	10.92	−6.251	1047.5	−6 057

[a]Compiled by W. M. Sawdon, vapor pressures converted from *International Critical Tables.*

TABLE 6. PROPERTIES OF SATURATED WATER VAPOR WITH AIR AT LOW TEMPERATURES[a] (PART I, CONCLUDED)

Temp., F	Pressure of Saturated Vapor × 10⁻⁵: In. of Hg	Pressure of Saturated Vapor × 10⁻⁵: Lb per Sq In	Weight of Saturated Vapor, per Cu Ft: Pounds × 10⁻⁵	Weight of Saturated Vapor, per Cu Ft: Grains	Weight of Saturated Vapor, per lb of Dry Air: Pounds × 10⁻⁵	Weight of Saturated Vapor, per lb of Dry Air: Grains	Volume in Cu Ft Barometer, 29 92 In Hg: of 1 lb of Dry Air	Volume in Cu Ft Barometer, 29 92 In Hg: of 1 lb of Dry Air + Vapor to Saturate it	Heat Content per Lb: Dry Air 0 F Datum	Heat Content per Lb: Vapor 32 F Datum	Heat Content per Lb: Dry Air with Vapor to Saturate it
−25	946.4	464 87	1 8016	0.12611	19 68	1 3776	10 95	10 95	−6 011	1048 0	−5 805
−24	1003.	492 67	1 9049	.13334	20 86	1 4602	10 97	10 97	−5 770	1048 4	−5 551
−23	1064	522 64	2 0162	14113	22 13	1 5491	11 00	11 00	−5 529	1048 9	−5 297
−22	1126	553 09	2 1287	.14901	23.42	1 6394	11 02	11 02	−5 288	1049 3	−5 042
−21	1192	585 51	2 2484	.15739	24 79	1 7353	11 05	11 05	−5 047	1049 8	−4 787
−20	1262 0	619 89	2 3750	0 16625	26 25	1 8375	11 07	11 07	−4 807	1050 2	−4 531
−19	1337.	656 73	2 5105	.17574	27 81	1 9467	11 10	11 10	−4 566	1050 7	−4 274
−18	1416	695 54	2 6527	.18569	29 45	2 0615	11 13	11 13	−4 325	1051 1	−4 015
−17	1496	734 84	2 7963	19574	31 12	2 1784	11 15	11 15	−4 085	1051 6	−3 758
−16	1584.	778 06	2 9542	20679	32 95	2 3065	11 18	11 18	−3 844	1052 0	−3 497
−15	1675.0	822 76	3 1168	0 21818	34 84	2 4388	11 20	11 21	−3 604	1052 5	−3 237
−14	1772	870 41	3 2899	.23029	36 86	2 5802	11 23	11 24	−3 363	1052 9	−2 975
−13	1874	920 51	3 4714	24300	38 98	2 7286	11 25	11 26	−3 123	1053 4	−2 712
−12	1980.	972 58	3 6596	25617	41.19	2 8833	11 28	11 29	−2 883	1053 8	−2 449
−11	2093	1028 1	3 8599	27019	43 54	3 0478	11 30	11 31	−2 642	1054 3	−2 183
−10	2210 0	1085.6	4 0666	0 28466	45 98	3 2186	11 33	11 34	−2 402	1054 7	−1 917
−9	2335.	1147 0	4 2871	.30009	48 58	3 4006	11 35	11 36	−2 162	1055 2	−1 649
−8	2463.	1209 8	4 5120	31584	51 25	3 5875	11 38	11 39	−1 921	1055 6	−1 380
−7	2502	1229 0	4 5734	32014	52 06	3 6442	11 40	11 41	−1 681	1056 1	−1 131
−6	2745.	1348 3	5.0066	35046	57.12	3 9984	11 43	11 44	−1 441	1056 5	−0 8375
−5	2898 0	1423 5	5 2738	0 36917	60 30	4 2210	11 45	11 46	−1 201	1057 0	−0 5636
−4	3055	1500 6	5 5473	38831	63 57	4 4499	11 48	11 49	−0 9604	1057 4	−0 2882
−3	3222.	1582 6	5 8379	40865	67 05	4 6935	11 50	11 51	−0 7203	1057 9	−0 01098
−2	3397.	1668 6	6 1414	42990	70 69	4 9483	11 53	11 54	−0 4802	1058 3	+0 2679
−1	3580.	1758 5	6 4583	45208	74.50	5.2150	11 55	11 57	−0 2401	1058 8	+0 5487
0	3773 0	1853.3	6 7914	0 47500	78 52	5 5000	11 58	11 59	0	1059 2	+0 8317

[a]Compiled by W. M. Sawdon, vapor pressures converted from *International Critical Tables*

TABLE 6 PROPERTIES OF SATURATED WATER VAPOR WITH AIR, 0 F TO 200 F[a] (PART II)

Temp., F	Pressure of Saturated Vapor		Weight of Saturated Vapor				Volume in Cu Ft Barometer, 29.92 In Hg		Heat Content per Lb		
	In of Hg	Lb per Sq In	per Cu Ft Pounds	per Cu Ft Grains	per lb of Dry Air Pounds	per lb of Dry Air Grains	of 1 lb of Dry Air	of 1 lb of Dry Air + Vapor to Saturate it	Dry Air 0 F Datum	Vapor 32 F Datum	Dry Air with Vapor to Saturate it
0	0.03773	0.01853	0.000067914	0.475	0.0007852	5.50	11.58	11.59	0.0000	1059.2	0.8317
1	.03975	.01963	.000071395	.500	.0008275	5.79	11.60	11.62	.2401	1059.7	1.117
2	.04186	.02056	.000075021	.525	.0008714	6.10	11.63	11.64	.4801	1060.1	1.404
3	.04409	.02166	.000078851	.552	.0009179	6.43	11.65	11.67	.7201	1060.6	1.694
4	.04645	.02282	.000082890	.580	.0009671	6.77	11.68	11.70	.9601	1061.0	1.986
5	0.04886	0.02400	0.000087005	0.609	0.001017	7.12	11.70	11.72	1.200	1061.5	2.280
6	.05144	.02527	.000091399	.640	.001071	7.50	11.73	11.75	1.440	1061.9	2.577
7	.05412	.02658	.000095955	.672	.001127	7.89	11.75	11.77	1.680	1062.4	2.877
8	.05692	.02796	.00010070	.705	.001186	8.30	11.78	11.80	1.920	1062.8	3.180
9	.05988	.02941	.00010572	.740	.001247	8.73	11.80	11.83	2.160	1063.3	3.486
10	0.06295	0.03092	0.00011090	0.776	0.001311	9.18	11.83	11.85	2.400	1063.7	3.795
11	.06618	.03251	.00011634	.814	.001379	9.65	11.86	11.88	2.640	1064.2	4.108
12	.06958	.03418	.00012206	.854	.001450	10.15	11.88	11.91	2.880	1064.6	4.424
13	.07309	.03590	.00012794	.896	.001523	10.66	11.91	11.93	3.120	1065.1	4.742
14	.07677	.03771	.00013410	.939	.001600	11.20	11.93	11.96	3.359	1065.5	5.064
15	0.08067	0.03963	0.00014062	0.984	0.001682	11.77	11.96	11.99	3.599	1066.0	5.392
16	.08469	.04160	.00014732	1.031	.001766	12.36	11.98	12.01	3.839	1066.4	5.722
17	.08895	.04369	.00015440	1.081	.001855	12.99	12.00	12.04	4.079	1066.9	6.058
18	.09337	.04586	.00016174	1.132	.001947	13.63	12.03	12.07	4.319	1067.3	6.397
19	.09797	.04812	.00016935	1.185	.002043	14.30	12.06	12.09	4.559	1067.8	6.741
20	0.1028	0.05050	0.00017747	1.242	0.002144	15.01	12.08	12.12	4.798	1068.2	7.088
21	.1078	.05295	.00018584	1.299	.002250	15.75	12.11	12.15	5.038	1068.7	7.443
22	.1132	.05560	.00019439	1.361	.002361	16.53	12.13	12.18	5.278	1069.1	7.802
23	.1186	.05826	.00020335	1.423	.002476	17.33	12.16	12.20	5.518	1069.6	8.166
24	.1244	.06111	.00021276	1.489	.002596	18.17	12.18	12.23	5.758	1070.0	8.536
25	0.1304	0.06405	0.00022255	1.558	0.002722	19.05	12.21	12.26	5.998	1070.5	8.912
26	.1366	.06710	.00023278	1.629	.002853	19.97	12.23	12.29	6.237	1070.9	9.292
27	.1432	.07034	.00024342	1.704	.002991	20.94	12.26	12.32	6.477	1071.4	9.682
28	.1500	.07368	.00025445	1.781	.003133	21.93	12.28	12.34	6.717	1071.8	10.075
29	.1571	.07717	.00026597	1.862	.003283	22.99	12.31	12.37	6.957	1072.3	10.477
30	0.1645	0.08050	0.00027797	1.946	0.003439	24.07	12.33	12.40	7.197	1072.7	10.886
31	.1722	.08458	.00029043	2.033	.003601	25.21	12.36	12.43	7.437	1073.2	11.302
32	.1803	.08856	.00030343	2.124	.003771	26.40	12.38	12.46	7.677	1073.6	11.726
33	.1879	.09230	.00031471	2.203	.003931	27.52	12.41	12.49	7.917	1074.1	12.139
34	.1957	.09610	.00032690	2.283	.004034	28.66	12.43	12.51	8.157	1074.5	12.556

[a]Compiled by W. M. Sawdon, vapor pressures converted from *International Critical Tables*

Table 6 Properties of Saturated Water Vapor with Air, 0 F to 200 F[a] (Part II, Continued)

Temp., F	Pressure of Saturated Vapor		Weight of Saturated Vapor				Volume in Cu Ft Barometer, 29 92 In Hg		Heat Content per Lb		
			per Cu Ft		per lb of Dry Air						
	In of Hg	Lb per Sq In	Pounds	Grains	Pounds	Grains	of 1 lb of Dry Air	of 1 lb of Dry Air + Vapor to Saturate it	Dry Air 0 F Datum	Vapor 32 F Datum	Dry Air with Vapor to Saturate it
35	0 20360	0.1000	0 0003394	2 376	0.004262	29 83	12 46	12 54	8 397	1075 0	12 979
36	.21195	.1041	0003527	2.469	004438	31 07	12 48	12 57	8 636	1075 4	13 409
37	22050	1083	0003662	2 563	004618	32 33	12 51	12 60	8 876	1075 9	13 845
38	22925	1126	0003799	2 660	004803	33 62	12 53	12 63	9 116	1076 3	14 285
39	.23842	.1171	.0003943	2.760	.004996	34 97	12 56	12 66	9 356	1076 8	14 736
40	0 24778	0 1217	0 0004090	2 863	0 005194	36 36	12 59	12 69	9 596	1077 2	15 191
41	.25755	1265	0004243	2 970	005401	37 80	12 61	12 72	9 836	1077 7	15 657
42	.26773	.1315	0004401	3 081	005616	39 31	12 64	12 75	10 08	1078 1	16 13
43	27832	1367	0004566	3 196	005840	40 88	12 66	12 78	10 32	1078 6	16 62
44	.28911	1420	0004735	3 315	006069	42 48	12 69	12 81	10 56	1079 0	17 11
45	0 30031	0 1475	0 0004909	3 436	0 006306	44 14	12 71	12 84	10 80	1079 5	17 61
46	31191	1532	0005088	3 562	006553	45 87	12 74	12 87	11 04	1079 9	18 12
47	32393	1591	0005274	3 692	006808	47 66	12 76	12 90	11 28	1080 4	18 64
48	33635	1652	0005465	3 826	007072	49 50	12 79	12 93	11 52	1080 8	19 16
49	34917	1715	0005663	3 964	007345	51 42	12 81	12 96	11 76	1081 3	19 70
50	0 36241	0.1780	0 0005866	4 106	0 007626	53 38	12 84	12 99	12 00	1081 7	20 25
51	.37625	1848	.0006078	4 255	007921	55 45	12 86	13 02	12 23	1082 2	20 80
52	.39051	1918	.0006296	4 407	008226	57 58	12 89	13 06	12 47	1082 6	21 38
53	40496	1989	.0006516	4 561	008534	59 74	12 91	13 09	12 71	1083 1	21 95
54	42003	.2063	0006746	4 722	.008856	61 99	12 94	13.12	12 95	1083 5	22 55
55	0 43570	0 2140	0 0006984	4 889	0 009192	64 34	12 96	13 15	13 19	1084 0	23 15
56	.45179	2219	0007228	5 060	009536	66 75	12 99	13 19	13 43	1084 4	23 77
57	.46828	2300	0007477	5.234	009890	69 23	13 01	13 22	13 67	1084 9	24 40
58	.48538	2384	0007735	5 415	01026	71 82	13 04	13 25	13 91	1085 3	25 05
59	50310	.2471	.0008003	5 602	01064	74 48	13 06	13 29	14 15	1085 8	25 70
60	0.52142	0 2561	0 0008278	5 795	0 01103	77 21	13 09	13 32	14 39	1086 2	26 37
61	.54035	2654	.0008562	5 993	.01144	80 08	13 11	13 35	14 63	1086 7	27 08
62	55970	2749	.0008852	6.196	01186	83 02	13 14	13 39	14 87	1087 1	27 76
63	.57985	.2848	0009153	6 407	01229	86 03	13 16	13 42	15 11	1087 6	28 48
64	.60042	.2949	0009460	6 622	01274	89 18	13 19	13 46	15 35	1088 0	29 21
65	0 62179	0 3054	0 0009778	6 845	0.01320	92 40	13 21	13 49	15 59	1088 5	29.96
66	.64378	3162	0010105	7 074	.01368	95 76	13 24	13 53	15 83	1088 9	30 73
67	66638	.3273	0010440	7 308	.01417	99 19	13 26	13 57	16 07	1089 4	31 51
68	.68980	3388	.0010816	7 571	.01468	102 8	13 29	13 60	16 31	1089.8	32 31
69	.71382	.3506	.0011140	7 798	.01520	106 4	13 31	13 64	16 55	1090 3	33 12

[a] Compiled by W. M. Sawdon, vapor pressures converted from *International Critical Tables*

TABLE 6. PROPERTIES OF SATURATED WATER VAPOR WITH AIR, 0 F TO 200 F[a] (PART II, CONTINUED)

Temp., F	Pressure of Saturated Vapor		Weight of Saturated Vapor				Volume in Cu Ft Barometer, 29.92 In Hg		Heat Content per Lb		
	In. of Hg	Lb per Sq In.	per Cu Ft		per lb of Dry Air		of 1 lb of Dry Air	of 1 lb of Dry Air + Vapor to Saturate it	Dry Air 0 F Datum	Vapor 32 F Datum	Dry Air with Vapor to Saturate it
			Pounds	Grains	Pounds	Grains					
70	0.73866	0.3628	0.0011507	8.055	0.01574	110.2	13.34	13.68	16.79	1090.7	33.96
71	.76431	.3754	.0011884	8.319	.01631	114.2	13.37	13.71	17.03	1091.2	34.83
72	.79058	.3883	.0012269	8.588	.01688	118.2	13.40	13.75	17.27	1091.6	35.70
73	.81766	.4016	.0012667	8.867	.01748	122.4	13.42	13.79	17.51	1092.1	36.60
74	.84555	.4153	.0013075	9.153	.01809	126.6	13.44	13.83	17.75	1092.5	37.51
75	0.87448	0.4295	0.0013497	9.448	0.01873	131.1	13.47	13.87	17.99	1093.0	38.46
76	.90398	.4440	.0013927	9.749	.01938	135.7	13.49	13.91	18.23	1093.4	39.42
77	.93452	.4590	.0014371	10.06	.02005	140.4	13.52	13.95	18.47	1093.9	40.40
78	.96588	.4744	.0014825	10.38	.02075	145.3	13.54	13.99	18.71	1094.3	41.42
79	.99825	.4903	.0015295	10.71	.02147	150.3	13.57	14.03	18.95	1094.8	42.46
80	1.0318	0.5067	0.0015777	11.04	0.02221	155.5	13.59	14.08	19.19	1095.2	43.51
81	1.0661	.5236	.0016273	11.39	.02298	160.9	13.62	14.12	19.43	1095.7	44.61
82	1.1013	.5409	.0016781	11.75	.02377	166.4	13.64	14.16	19.67	1096.1	45.72
83	1.1377	.5588	.0017304	12.11	.02459	172.1	13.67	14.21	19.91	1096.6	46.88
84	1.1752	.5772	.0017841	12.49	.02543	178.0	13.69	14.26	20.15	1097.0	48.05
85	1.2135	0.5960	0.0018389	12.87	0.02629	184.0	13.72	14.30	20.39	1097.5	49.24
86	1.2527	.6153	.0018950	13.27	.02718	190.3	13.74	14.34	20.63	1097.9	50.47
87	1.2933	.6352	.0019531	13.67	.02810	196.7	13.77	14.39	20.87	1098.4	51.74
88	1.3346	.6555	.0020116	14.08	.02904	203.3	13.79	14.44	21.11	1098.8	53.02
89	1.3774	.6765	.0020725	14.51	.03002	210.1	13.82	14.48	21.35	1099.3	54.35
90	1.4211	0.6980	0.0021344	14.94	0.03102	217.1	13.84	14.53	21.59	1099.7	55.70
91	1.4661	.7201	.0021982	15.39	.03205	224.4	13.87	14.58	21.83	1100.2	57.09
92	1.5125	.7429	.0022634	15.84	.03312	231.8	13.89	14.63	22.07	1100.6	58.52
93	1.5600	.7662	.0023304	16.31	.03421	239.5	13.92	14.69	22.32	1101.1	59.99
94	1.6088	.7902	.0023992	16.79	.03535	247.5	13.94	14.73	22.56	1101.5	61.50
95	1.6591	0.8140	0.0024697	17.28	0.03652	255.6	13.97	14.79	22.80	1102.0	63.05
96	1.7108	.8403	.0025425	17.80	.03772	264.0	13.99	14.84	23.04	1102.4	64.62
97	1.7638	.8663	.0026164	18.31	.03896	272.7	14.02	14.90	23.28	1102.9	66.25
98	1.8181	.8930	.0026925	18.85	.04024	281.7	14.04	14.95	23.52	1103.3	67.92
99	1.8741	.9205	.0027700	19.39	.04156	290.9	14.07	15.01	23.76	1103.8	69.63
100	1.9316	0.9487	0.0028506	19.95	0.04293	300.5	14.10	15.07	24.00	1104.2	71.40
101	1.9904	.9776	.0029316	20.52	.04433	310.3	14.12	15.12	24.24	1104.7	73.21
102	2.0507	1.0072	.0030156	21.11	.04577	320.4	14.15	15.18	24.48	1105.1	75.06
103	2.1128	1.0377	.0031017	21.71	.04726	330.8	14.17	15.25	24.72	1105.6	76.97
104	2.1763	1.0689	.0031887	22.32	.04879	341.5	14.20	15.31	24.96	1106.0	78.92

[a]Compiled by W. M. Sawdon, vapor pressures converted from *International Critical Tables*.

Table 6. Properties of Saturated Water Vapor with Air, 0 F to 200 F[a] (Part II, Continued)

Temp., F	Pressure of Saturated Vapor		Weight of Saturated Vapor				Volume in Cu Ft Barometer, 29 92 In Hg		Heat Content per Lb		
	In of Hg	Lb per Sq In.	per Cu Ft Pounds	per Cu Ft Grains	per lb of Dry Air Pounds	per lb of Dry Air Grains	of 1 lb of Dry Air	of 1 lb of Dry Air + Vapor to Saturate it	Dry Air 0 F Datum	Vapor 32 F Datum	Dry Air with Vapor to Saturate it
105	2.2414	1.1009	0 0032786	22 95	0 05037	352 6	14 22	15.37	25 20	1106 5	80 93
106	2 3084	1.1338	0033715	23.60	.05200	364 0	14 25	15 44	25 44	1106 9	83 00
107	2 3770	1.1675	0034650	24 26	.05368	375 8	14.27	15 50	25 68	1107 4	85 13
108	2 4473	1 2020	.0035612	24.93	05541	387 9	14.30	15.57	25 92	1107 8	87 30
109	2.5196	1.2375	.0036603	25.62	.05719	400 3	14.32	15.64	26.16	1108.3	89.54
110	2.5939	1.274	0 0037622	26 34	0 05904	413 3	14 35	15.71	26 40	1108 7	91 86
111	2 6692	1.311	0038669	27 07	06092	426 4	14 37	15 78	26 64	1109 2	94 21
112	2.7486	1.350	0039729	27 81	06292	440 4	14 39	15 85	26 88	1109 6	96 70
113	2 8280	1.389	0040816	28 57	06493	454 5	14.42	15 93	27 12	1110 1	99 20
114	2.9094	1.429	0041911	29 34	06700	469 0	14 45	16 00	27 36	1110 5	101 76
115	2 9929	1 470	0 0043047	30 13	0 06913	483 9	14 47	16 08	27 60	1111 0	104 40
116	3.0784	1 512	0044208	30 95	07134	499 4	14.50	16 16	27 84	1111 4	107 13
117	3.1660	1.555	.0045372	31.76	.07361	515.3	14.52	16 24	28 08	1111 9	109 92
118	3 2576	1 600	0046620	32 63	07600	532 0	14 55	16.32	28 32	1112 3	112 85
119	3.3492	1 645	0047846	33 49	07840	548 8	14.57	16 41	28 56	1112 8	115 80
120	3 4449	1 692	0 0049115	34 38	0 08093	566 5	14 60	16 50	28 80	1113 2	118 89
121	3.5406	1.739	.005040	35 28	.08348	584 4	14.62	16.58	29 04	1113.7	122 01
122	3 6404	1 788	005173	36 21	.08616	603 1	14.65	16 68	29 28	1114 1	125 27
123	3.7422	1 838	005311	37.18	08892	622.4	14 67	16 77	29 52	1114 6	128 63
124	3 8460	1 889	.005450	38 15	.09175	642.3	14.70	16.87	29.76	1115.0	132.06
125	3 9519	1.941	0 005590	39.13	0 09466	662 6	14.72	16.96	30 00	1115 5	135 59
126	4 0618	1 995	005734	40 14	09770	683 9	14.75	17.06	30 24	1115.9	139 26
127	4 1718	2 049	.005882	41.17	.1008	705.6	14 77	17.17	30 48	1116 4	143 01
128	4 2858	2 105	.006031	42.22	.1040	728.0	14.80	17 27	30.72	1116 8	146 87
129	4 4039	2.163	.006188	43.32	.1074	751 8	14.83	17.38	30 96	1117 3	150 96
130	4.5220	2.221	0.006344	44.41	0.1107	774 9	14 85	17.49	31.20	1117.7	154.93
131	4.6441	2.281	.006504	45.53	.1143	800 1	14 88	17.61	31 45	1118.2	159.26
132	4.7703	2 343	.006671	46 70	.1180	826 0	14 90	17 73	31 69	1118 6	163 68
133	4.8986	2 406	.006839	47 87	.1218	852 6	14 93	17 85	31 93	1119 1	168 24
134	5 0289	2 470	.007010	49 07	.1257	879 9	14 95	17 97	32.17	1119 5	172 89
135	5 1633	2.536	0 007185	50 30	0.1297	907 9	14.98	18 10	32.41	1120 0	177 67
136	5.2997	2 603	.007364	51.55	1339	937 3	15 00	18.23	32 65	1120.4	182 67
137	5.4402	2 672	.007547	52.83	1382	967 4	15 03	18 36	32.89	1120 9	187 80
138	5.5827	2.742	.007732	54.12	1427	998 9	15 05	18 50	33.13	1121.3	193 14
139	5.7293	2.814	.007923	55 46	.1473	1031 1	15 08	18 65	33 37	1121.8	198 61

[a]Compiled by W. M Sawdon, vapor pressures converted from *International Critical Tables*

TABLE 6. PROPERTIES OF SATURATED WATER VAPOR WITH AIR, 0 F TO 200 F[a] (PART II, CONTINUED)

Temp., F	Pressure of Saturated Vapor: In of Hg	Pressure of Saturated Vapor: Lb per Sq In	Weight of Saturated Vapor per Cu Ft: Pounds	Weight of Saturated Vapor per Cu Ft: Grains	Weight of Saturated Vapor per lb of Dry Air: Pounds	Weight of Saturated Vapor per lb of Dry Air: Grains	Volume in Cu Ft Barometer, 29.92 In Hg: of 1 lb of Dry Air	Volume in Cu Ft Barometer, 29.92 In Hg: of 1 lb of Dry Air + Vapor to Saturate it	Heat Content per Lb: Dry Air 0 F Datum	Heat Content per Lb: Vapor 32 F Datum	Heat Content per Lb: Dry Air with Vapor to Saturate it
140	5.8779	2.887	0.008116	56.81	0.1521	1064.7	15.10	18.79	33.61	1122.2	204.30
141	6.0306	2.962	.008313	58.19	.1570	1099.0	15.13	18.94	33.85	1122.7	210.11
142	6.1874	3.039	.008516	59.61	.1622	1135.4	15.15	19.10	34.09	1123.1	216.26
143	6.3482	3.118	.008724	61.07	.1675	1172.5	15.18	19.26	34.33	1123.6	222.53
144	6.5111	3.198	.008933	62.53	.1730	1211.0	15.20	19.43	34.57	1124.0	229.02
145	6.6781	3.280	0.009148	64.04	0.1787	1250.9	15.23	19.60	34.81	1124.5	235.76
146	6.8471	3.363	.009366	65.56	.1846	1292.2	15.25	19.78	35.05	1124.9	242.71
147	7.0222	3.449	.009590	67.13	.1908	1335.6	15.28	19.96	35.29	1125.4	250.02
148	7.1993	3.536	.009817	68.72	.1971	1379.7	15.30	20.15	35.53	1125.8	257.43
149	7.3805	3.625	.010040	70.28	.2037	1425.9	15.33	20.35	35.77	1126.3	265.20
150	7.5658	3.716	0.010284	71.99	0.2105	1473.5	15.35	20.55	36.02	1126.7	273.19
151	7.7551	3.809	.010526	73.68	.2176	1523.2	15.38	20.76	36.26	1127.2	281.54
152	7.9485	3.904	.010772	75.40	.2250	1575.0	15.40	20.97	36.50	1127.6	290.21
153	8.1460	4.001	.011022	77.15	.2327	1628.9	15.43	21.20	36.74	1128.1	299.25
154	8.3476	4.100	.011279	78.95	.2407	1684.9	15.45	21.43	36.98	1128.5	308.61
155	8.5532	4.201	0.011539	80.77	0.2490	1743.0	15.48	21.67	37.22	1129.0	318.34
156	8.7650	4.305	.011807	82.65	.2577	1803.9	15.50	21.93	37.46	1129.4	328.51
157	8.9788	4.410	.012077	84.54	.2667	1866.9	15.53	22.19	37.70	1129.9	339.04
158	9.1986	4.518	.012354	86.48	.2761	1932.7	15.56	22.46	37.94	1130.3	350.02
159	9.4206	4.627	.012634	88.44	.2858	2000.6	15.58	22.74	38.18	1130.8	361.36
160	9.6486	4.739	0.012919	90.43	0.2961	2072.7	15.61	23.03	38.43	1131.2	373.38
161	9.8807	4.853	.013211	92.48	.3067	2146.9	15.63	23.33	38.67	1131.7	385.76
162	10.119	4.970	.013509	94.56	.3179	2225.3	15.66	23.65	38.91	1132.1	398.80
163	10.361	5.089	.013812	96.68	.3295	2306.5	15.68	23.98	39.15	1132.5	412.34
164	10.608	5.210	.014120	98.84	.3416	2391.2	15.71	24.33	39.39	1133.0	426.42
165	10.860	5.334	0.014434	101.0	0.3544	2480.8	15.73	24.69	39.63	1133.5	441.34
166	11.117	5.460	.014753	103.3	.3677	2573.9	15.76	25.07	39.87	1133.9	456.81
167	11.379	5.589	.015080	105.6	.3817	2671.9	15.78	25.46	40.11	1134.4	473.11
168	11.646	5.720	.015410	107.9	.3964	2774.8	15.81	25.88	40.35	1134.8	490.18
169	11.919	5.854	.015750	110.3	.4118	2882.6	15.83	26.31	40.59	1135.3	508.11
170	12.196	5.990	0.016092	112.6	0.4280	2996.0	15.86	26.77	40.83	1135.7	526.91
171	12.480	6.130	.016444	115.1	.4451	3115.7	15.88	27.24	41.07	1136.2	546.79
172	12.770	6.272	.016801	117.6	.4631	3241.7	15.91	27.74	41.32	1136.6	567.68
173	13.065	6.417	.017164	120.1	.4821	3374.7	15.93	28.28	41.56	1137.1	589.76
174	13.366	6.565	.017534	122.7	.5022	3515.4	15.96	28.84	41.80	1137.5	613.05

[a]Compiled by W. M. Sawdon, vapor pressures converted from *International Critical Tables*.

TABLE 6 PROPERTIES OF SATURATED WATER VAPOR WITH AIR, 0 F TO 200 F[a] (PART II, CONCLUDED)

Temp F	Pressure of Saturated Vapor		Weight of Saturated Vapor				Volume in Cu Ft Barometer, 29 92 In Hg		Heat Content per Lb		
	In of Hg	Lb per Sq In.	per Cu Ft		per lb of Dry Air		of 1 lb of Dry Air	of 1 lb of Dry Air + Vapor to Saturate it	Dry Air 0 F Datum	Vapor 32 F Datum	Dry Air with Vapor to Saturate it
			Pounds	Grains	Pounds	Grains					
175	13.674	6.716	0 017914	125 4	0 5235	3664.5	15 98	29 43	42 04	1138 0	637 78
176	13 985	6 869	018294	128 1	5459	3821 3	16 01	30 05	42 28	1138 4	663 73
177	14.303	7.025	018684	130 8	.5697	3987 9	16 03	30 71	42 52	1138 9	691 35
178	14.627	7.184	.019080	133 6	5949	4164 3	16 06	31 41	42 76	1139 3	720 53
170	14.354	7.345	.019477	136 3	6215	4350 5	16 08	32 15	43 00	1139 8	751 39
180	15.290	7.510	0 019888	139 2	0 6501	4550 7	16 11	32 94	43 24	1140 2	784.48
181	15.632	7 678	020304	142.1	6805	4763 5	16 13	33 78	43 49	1140 7	819 74
182	15.981	7.849	020729	145.1	7131	4991 7	16 16	34 68	43 73	1141 1	857 45
183	16.337	8.024	021159	148 1	7481	5236 7	16 18	35 65	43 97	1141 6	898 00
184	16.697	8.201	021598	151 2	7854	5497 8	16 21	36 67	44 21	1142 0	941 14
185	17.066	8.382	0 022045	154 3	0 8258	5780 6	16 23	37 78	44 45	1142 5	987 93
186	17.440	8 566	022497	157 5	8693	6085 1	16 26	38 98	44 69	1142 9	1038 21
187	17.821	8.753	022956	160.7	9162	6413 4	16 28	40 27	44 93	1143 4	1092 51
188	18.210	8.944	023424	164 0	9673	6771 1	16 31	41 67	45 18	1143 8	1151 58
189	18.605	9.138	023900	167.3	1 0227	7158 9	16 34	43 04	45 42	1144 3	1216 04
190	19.008	9 336	0 024384	170 7	1 083	7581 0	16 36	44 85	45.66	1144 7	1285 37
191	19 419	9.538	024881	174 2	1 150	8050 0	16 39	46 68	45 90	1145 2	1362 88
192	19.839	9.744	025380	177.7	1 224	8568 0	16 41	48 70	46 14	1145 6	1448 35
193	20 266	9 954	025893	181 3	1 306	9142 0	16 44	50 93	46 38	1146 1	1543 19
194	20.702	10.168	.026413	184 9	1 397	9779 0	16 46	53 42	46 62	1146.5	1648 28
195	21.144	10.385	0 026939	188 6	1 499	10493 0	16 49	56 20	46 86	1147 0	1766 21
196	21.592	10 605	027472	192 3	1 613	11291 0	16 51	59 31	47 10	1147 4	1897 86
197	22.048	10 829	.028019	196.1	1.742	12194 0	16 54	62 85	47.34	1147 9	2046 98
198	22.512	11 057	028571	200.0	1 890	13230 0	16 56	66 88	47 59	1148 3	2217 88
199	22.984	11.289	.029129	203.9	2 061	14427 0	16 59	71.54	47.83	1148 8	2415.51
200	23.465	11.525	0.029700	207.9	2 261	15827 0	16 61	76 99	48 07	1149 2	2646 41

[a]Compiled by W. M Sawdon vapor pressures converted from *International Critical Tables*

$$h'_{fg}\,(W_{t'} - W) = c_{p_a}\,(t - t') + c_{p_s}W\,(t - t') \tag{9a}$$

and using $c_{p_a} = 0.24$ and $c_{p_s} = 0.45$

$$h'_{fg}\,(W_{t'} - W) = (0.24 + 0.45W)\,(t - t') \tag{9b}$$

where

h'_{fg} = latent heat of vaporization at t', Btu per pound.

$(W_{t'} - W)$ = increase in vapor associated with 1 lb of dry air when it is saturated adiabatically from an initial dry-bulb temperature, t, and an initial vapor content, W, pounds.

Knowing any two of the three primary variables, t, t', or W, the third may be found from this equation for any process of adiabatic saturation.

TOTAL HEAT AND HEAT CONTENT

The total heat of a mixture of dry air and water vapor was originally defined by W. H. Carrier as

$$\Sigma = c_{p_a}\,(t - 0) + W\,[h'_{fg} + c_{p_s}\,(t - t')] \tag{10}$$

where

Σ = total heat of the mixture, Btu per pound of dry air.
c_{p_a} = mean specific heat at constant pressure of dry air.
c_{p_s} = mean specific heat at constant pressure of water vapor.
t = dry-bulb temperature, degrees Fahrenheit.
t' = wet-bulb temperature, degrees Fahrenheit.
W = weight of water vapor mixed with each pound of dry air, pounds.
h'_{fg} = latent heat of vaporization at t', Btu per pound.

Since this definition holds for any mixture of dry air and water vapor, the total heat of a mixture with a relative humidity of 100 per cent and at a temperature equal to the wet-bulb temperature (t') is

$$\Sigma' = c_{p_a}\,(t' - 0) + W_{t'}\,h'_{fg} \tag{11}$$

By equating Equation 10 to Equation 11, the equation for the adiabatic saturation process, Equation 9a, follows. This demonstrates that the adiabatic saturation process at constant wet-bulb temperature is also a process of constant total heat. In short, the total heat of a mixture of dry air and water vapor is the same for any two states of the mixture at the same wet-bulb temperature. This fact furnishes a convenient means of finding the total heat of an air-vapor mixture in any state.

By considering the temperatures in Table 6 to be wet-bulb readings, the total heat of any air-vapor mixture may be obtained from the last column in the table.

Enthalpy

This total heat of an air-vapor mixture is not exactly equal to the true heat content or enthalpy of the mixture since the heat content of the liquid is not included in Equation 10. With the meaning of heat content in agreement with present practise in other branches of thermodynamics,

the true heat content of a mixture of dry air and water vapor (with 0 F as the datum for dry air, and the saturated liquid at 32 F as the datum for the water vapor) is

$$h = c_{p_a}(t - 0) + W h_s = 0.24 (t - 0) + W h_s \tag{12}$$

where

h = the heat content of the mixture, Btu per pound of dry air.
t = the dry-bulb temperature, degrees Fahrenheit.
W = the weight of vapor per pound of dry air, pounds.
h_s = the heat content of the vapor in the mixture, Btu per pound.

The heat content of the water vapor in the mixture may be found in steam charts or tables when the dry-bulb temperature and the partial pressure of the vapor are known. Or, since the heat content of steam at low partial pressures, whether super-heated or saturated, depends only upon temperature, the following empirical equation, derived from Properties of Saturated Steam by J. H. Keenan, Table 8, may be used:

$$h_s = 1059.2 + 0.45\, t \tag{13}$$

Substituting this value of h_s in Equation 12, the heat content of the mixture is

$$h = 0.24 (t - 0) + W (1059\ 2 + 0.45\, t) \tag{14}$$

Example 5. Find the total heat of an air-vapor mixture having a dry-bulb temperature of 85 F and a wet-bulb temperature of 70 F.

Solution. From Table 6, for saturation at the wet-bulb temperature, $W_t{}^1 = 0.01574$, and from Equation 14,

$$h = 0.24\,(70-0) + 0.01574\,(1059.2 + 0.45 \times 70) = 33.96 \text{ Btu per pound dry air.}$$

An energy equation can be written that applies, in general, to various air-conditioning processes, and this equation can be used to determine the quantity of heat transferred during such processes. In the most general form, this equation may be explained with the aid of Fig. 1 as follows:

The rectangle may represent any apparatus, *e.g.*, a drier, humidifier, dehumidifier, cooling tower, or the like, by proper choice of the direction of the arrows.

In general, a mixture of air and water vapor, such as atmospheric air, enters the apparatus at 1 and leaves at 3. Water is supplied at some temperature, t_2. For the flow of 1 lb of dry air (with accompanying vapor) through the apparatus, provided there is no appreciable change in the elevation or velocity of the fluids and no mechanical energy delivered to or by the apparatus,

$$h_1 + E_h + (W_3 - W_1)\, h_2 = h_3 + R_c$$

or

$$E_h - R_c = h_3 - h_1 - (W_3 - W_1)\, h_2 \tag{15}$$

where

E_h = the quantity of heat supplied per pound of dry air, Btu.
R_c = the quantity of heat lost externally by heat transfer from the apparatus, Btu per pound of dry air.
W_1 = the weight of water vapor entering, per pound of dry air.
W_3 = the weight of water vapor leaving, per pound of dry air.
h_2 = the heat content of the water supplied at t_2, Btu per pound.

$h_3 - h_1$ = the increase in the heat content of the air-water vapor mixture in passing through the apparatus, Btu per pound of dry air

$$= 0.24\,(t_3 - t_1) + W_3\,(1059.2 + 0.45\,t_3) - W_1\,(1059.2 + 0.45t_1)$$

The net quantity of heat added to or removed from air-water vapor mixtures in air conditioning work is frequently *approximated* by taking the differences in total heat at exit and entrance.

For example, in Fig. 1, an *approximate* result is

$$E_h - R_c = \Sigma_3 - \Sigma_1 \qquad (16)$$

where

Σ_3 = the total heat of the air-vapor mixture at exit, Btu per pound of dry air.
Σ_1 = the total heat of the air-vapor mixture at entrance, Btu per pound of dry air

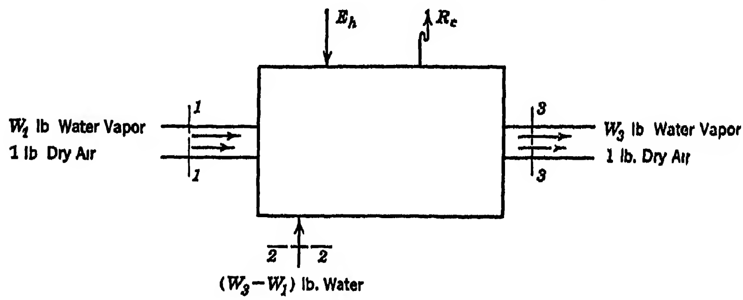


FIG. 1. DIAGRAM ILLUSTRATING ENERGY EQUATION 15

From the definitions of *total heat* and *heat content*, it may be demonstrated that Equation 16 is exactly equivalent to Equation 15, when, and only when, $t'_3 = t'_1 = t_2$; *i.e.*, when the initial and final wet-bulb temperatures and the temperature of the water supplied are equal. The one process that meets these conditions is adiabatic saturation, and either equation will give a result of zero; for other conditions, Equation 16 is approximate but satisfactory for many calculations.

The following problems illustrate the application of these principles:

Example 6. Heating (data from Example 2). Assuming the water to be supplied at 50 F, the net quantity of heat supplied is, from Equation 15,

$$E_h - R_c = 0.24\ (70 - 0) + 0.000548 \times 0.45\ (70 - 0) + 0.005032\ [1059.2 + 0.45 \times 70 - (50 - 32)] = 22.90 \text{ Btu per pound of dry air.}$$

Example 7. Cooling (data from Example 3) If the condensate is removed at 54 F the quantity of heat removed is found from Equation 15, by proper regard to the arrow direction in Fig. 1,

$$E_h + R_c = 0.24\ (84 - 54) + 0.00887 \times 0.45\ (84 - 54) + 0.00361\ [1059.2 + 0.45 \times 84 - (54 - 32)] = 11.24 \text{ Btu per pound of dry air.}$$

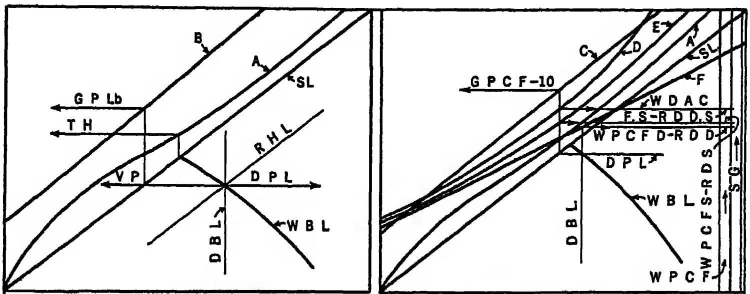
Using Table 6, the initial total heat of the air-vapor mixture, since the wet-bulb temperature is 70 F, is 33.96 Btu per pound of dry air.

The final total heat is, from Table 6, since the exit air is saturated, 22.55 Btu per pound. Hence, using Equation 16, the quantity of heat removed is, approximately, (33.96 − 22.55) or 11.41 Btu per pound of dry air. The degree of approximation to the correct result is evident in this example.

PSYCHROMETRIC CHART

The revised Bulkeley Psychrometric Chart[4], will be found attached to the inside back cover. It shows graphically the relationships expressed in Equations 9a and 9b. It also gives the grains of moisture per pound of dry air for saturation, the grains of moisture per cubic foot of saturated air, the total heat in Btu per pound of dry air saturated with moisture, and the weight of the dry air in pounds per cubic foot. Fig. 2 shows the procedure to follow in using the Bulkeley Chart. The directrix curves above the saturation line are as follows:

A is the total heat in Btu contained in the mixture above 0 F, and is to be referred to the column of figures at the left side of the chart. Heat of the liquid is not included



A, B, C, D, E, F=Directrix lines
D B L=Dry bulb line
D P. L.=Dew point line

R H L =Relative humidity line
W B L=Wet bulb line
S L=Saturation line

G P Lb=Grains moisture sat mixture per lb dry air
T. H =Total heat sat mixture per lb dry air
V P=Vapor pressure in mm mercury
G P. C F. S=Grains moisture per cu ft sat air
W D A C F S=Weight dry air per cu ft in lbs saturated

R D D S=Relative density dry air per cu ft saturated
W P C F S=Weight per cu ft in lbs saturated
R D S=Relative density per cu ft saturated
W P C F D=Weight per cu ft in lbs dry
R D D=Relative density per cu ft dry

$$\text{G. P C F.} \times \frac{\text{Abs temp at D P}}{\text{Abs temp at D B}} = \text{G P C F at partial saturation}$$

$$\text{W. P C F} \times \frac{\text{Abs temp at D P}}{\text{Abs temp at D B}} = \text{W P C F at partial saturation}$$

$$\text{R D} \times \frac{\text{Abs temp at D P}}{\text{Abs temp at D B}} = \text{R D at partial saturation}$$

FIG. 2. DIAGRAMS SHOWING PROCEDURE TO FOLLOW IN USING BULKELEY CHART

B is the grains of moisture of water vapor contained in each pound of the saturated mixture and is to be referred to the figures at the left side of the chart.

C is the grains of moisture of water vapor per cubic foot of saturated mixture, and is to be referred to the figures at the left side of the chart which are to be divided by 10

D is the weight in decimal fractions of a pound of dry air in one cubic foot of the saturated mixture, and is referred to the first column of figures to the right of the saturation line between the vertical dry-bulb temperature lines 170 and 180 F. The relative density of the mixture is read in a similar manner from the same curve by the column of figures between the vertical dry-bulb temperature lines 180 and 190 F

E is similar to *D* but is for one cubic foot of the saturated mixture.

[4]The Bulkeley Psychrometric Chart was presented to the Society in 1926 (See A S H V E TRANSACTIONS, Vol 32, 1926) Single copy of the chart can be furnished at a cost of $ 25

F is similar to *D* but is for dry air, devoid of all moisture or water vapor For convenience, the approximate absolute temperature of 500 F is given at 40 F on the saturation line for the purpose of calculating volume, weight per cubic foot, and relative density at partial saturation.

METHOD OF USING THE CHART

Example 8. Relative Humidity: At the intersection of the 78 F wet-bulb line and the 95 F dry-bulb line, the relative humidity is read directly on the straight diagonal lines as 46 per cent.

Example 9. Dew Point: At the intersection of the 78 F wet-bulb line, the dew-point temperature is read directly on the horizontal temperature lines as 70 9 F.

Example 10. Vapor Pressure: At the intersection of the 78 F wet-bulb line and the 95 F dry-bulb line, pass in a horizontal direction to the left of the chart and on the logarithmic scale read the vapor pressure as 19 4 millimeters of mercury. (Divide by 25 4 for inches.)

Example 11. Total Heat Above 0 F in Mixture per Pound of Dry Air Saturated with Moisture: From where the wet-bulb line joins the saturation line, pass in a vertical direction on the 78 F dry-bulb line to its intersection with curve *A* and on the logarithmic scale at the left of the chart read 40.6 Btu per pound of mixture. The use of this curve to obtain the total heat in the mixture at any wet-bulb temperature is a great convenience, as the number of Btu required to heat the mixture and humidify it, as well as the refrigeration required to cool and dehumidify the mixture, can be obtained by taking the difference in total heat before and after treatment of the mixture.

Example 12. Grains of Moisture per Pound of Dry Air Saturated with Moisture: From 70.9 F dew-point temperature on the saturation line, pass vertically to the intersection with curve *B* and on the logarithmic scale at the left read 114 grains of moisture per pound

Example 13. Grains of Moisture per Cubic Foot of Mixture, Partially Saturated: From 70 9 F dew-point temperature on the saturation line proceed in a vertical direction to curve *C*, and on the logarithmic scale to the left read 83.3 which, divided by 10, gives 8 33 grains. A temperature of 70.9 F is equal to an absolute temperature of 530.9, and 95 F equals 555, absolute temperature. Therefore, $\frac{530.9}{555} \times 8.33 = 7.97$ grains per cubic foot of partially saturated mixture

Example 14 Grains of Moisture per Cubic Foot of Saturated Air: Starting at the saturation line at the desired temperature, pass in a vertical direction to curve *C* and on the logarithmic scale at the left, read a number which, divided by 10, will give the answer.

Example 15. Weight per Cubic Foot of Dry Air and Relative Density: From the point where, for example, the 70 F vertical dry-bulb line intersects curve *E*, pass to right side and read 0.075 lb; if cubic feet per pound are desired, divide 1 by this amount. The relative density is read immediately to the right as 1.00.

Example 16. Weight of Dry Air per Cubic Foot of Saturated Mixture and Relative Density: From the point where, for example, the 70 F vertical line intersects the curve *D*, pass to the right and read weight per cubic foot as 0.07316 with a relative density of 0.9755 for saturated air at 70 F.

Example 17. Weight of Dry Air per Cubic Foot and Relative Density of Partially Saturated Air: Air at 50 F and a wet-bulb temperature of 46 F is to be heated to 130 F. The wet- and dry-bulb lines intersect at a dew-point temperature of 42 F. Pass to the left where this dew-point line intersects the saturation line and then pass in a vertical direction to where the 42 F dry-bulb line intersects with curve *D*. Then pass directly to the right and read the weight per cubic foot of saturated air at 42 F as 0.07844 and the relative density as 1.046. The absolute temperature at 42 F is 502, and at 130 F is 590. Therefore, $\frac{502}{590} = 0\ 851$ The weight of 1 cu ft of air at 50 F dry-bulb and 46 F wet-bulb when heated to 130 F is $0.07844 \times 0.851 = 0\ 06675$, and the relative density is $1.046 \times 0.851 = 0.89$.

PROPERTIES OF WATER

Composition of Water. Water is a chemical compound (H_2O) formed by the union of two volumes of hydrogen and one volume of oxygen, or two parts by weight of hydrogen and 16 parts by weight of oxygen.

Density of Water. Water has its greatest density at 39.2 F, and it expands when heated or cooled from this temperature. At 62 F a U. S. gallon of 231 cu in. of water weighs approximately 8⅓ lb, and a cubic foot of water is equal to 7.48 gal. The specific volume of water depends on the temperature and it is always the reciprocal of its density. (See Table 7.)

Table 7. Thermal Properties of Water

Temperature Deg F	Sat Press Lb per Sq In	Volume Cu Ft per Lb	Weight Lb per Cu Ft	Specific Heat
32	0 0887	0 01602	62.42	1.0093
40	0 1217	0.01602	62.42	1.0048
50	0.1780	0 01602	62.42	1.0015
60	0 2561	0.01603	62.38	0.9995
70	0.3628	0 01605	62.31	0.9982
80	0.5067	0.01607	62.23	0.9975
90	0 6980	0.01610	62.11	0.9971
100	0.9487	0.01613	62.00	0.9970
110	1.274	0 01616	61.88	0.9971
120	1.692	0 01620	61.73	0.9974
130	2.221	0 01625	61.54	0 9978
140	2.887	0.01629	61.39	0 9984
150	3 716	0.01634	61.20	0 9990
160	4 739	0.01639	61.01	0 9998
170	5 990	0.01645	60.79	1.0007
180	7.510	0 01650	60.61	1 0017
190	9.336	0 01656	60.39	1.0028
200	11.525	0.01663	60 13	1.0039
210	14.123	0.01669	59.92	1 0052
212	14.696	0 01670	59.88	1.0055
220	17.188	0.01676	59.66	1.0068
240	24.97	0.01690	59.17	1.0104
260	35.43	0 01706	58.62	1.0148
280	49.20	0.01723	58.04	1.0200
300	67.01	0.01742	57 41	1.0260
350	134.62	0 01797	55 65	1.0440
400	247 25	0.01865	53.62	1 0670
450	422.61	0.01950	51.30	1 0950
500	681.09	0.02050	48.80	1.1300
550	1045.4	0.02190	45.70	1.2000
600	1544.6	0 02410	41.50	1.3620
700	3096.4	0.03940	25.40	----------

Water Pressures. Pressures are often stated in feet or inches of water column. At 62 F, with h equal to the head in feet, the pressure of a column of water is $62.383h$ lb per square foot, or $0.433h$ lb per square inch. A column of water 2.309 ft (27.71 in.) high exerts a pressure of one pound per square inch at 62 F.

Boiling Point of Water. The boiling point of water varies with the pressure; it is lower at higher altitudes. A change in pressure will always be accompanied by a change in the boiling point, and there will be a cor-

TABLE 8. PROPERTIES OF SATURATED STEAM: PRESSURE TABLE[a]

Abs Press. Lb /Sq In	Temp. Deg F.	Specific Volume: Sat Liquid	Specific Volume: Evap.	Specific Volume: Sat. Vapor	Total Heat: Sat Liquid	Total Heat: Evap	Total Heat: Sat. Vapor	Entropy: Sat Liquid	Entropy: Evap	Entropy: Sat. Vapor	Abs Press. Lb /Sq In.
p	t	v_f	v_{fg}	v_g	h_f	h_{fg}	h_g	s_f	s_{fg}	s_g	p
½" Hg	58.83	0.01603	1256.9	1256.9	26.88	1058.8	1085.7	0.0533	2.0422	2.0955	½" Hg
¾" Hg	70.44	0.01605	856.5	856.5	38.47	1052.5	1091.0	0.0754	1.9856	2.0609	¾" Hg
1" Hg	79.06	0.01607	652.7	652.7	47.06	1047.8	1094.9	0.0914	1.9451	2.0365	1" Hg
1½" Hg	91.75	0.01610	445.3	445.3	59.72	1040.8	1100.6	0.1147	1.8877	2.0024	1½" Hg
2" Hg	101.17	0.01613	339.5	339.5	69.10	1035.7	1104.8	0.1316	1.8468	1.9784	2" Hg
2½" Hg	108.73	0.01616	275.2	275.2	76.63	1031.5	1108.1	0.1450	1.8148	1.9598	2½" Hg
3" Hg	115.08	0.01618	231.8	231.8	82.96	1027.9	1110.8	0.1561	1.7885	1.9446	3" Hg
1.0	101.76	0.01614	333.8	333.9	69.69	1035.3	1105.0	0.1326	1.8442	1.9769	1.0
2.0	126.10	0.01623	173.94	173.96	93.97	1021.6	1115.6	0.1750	1.7442	1.9192	2.0
3.0	141.49	0.01630	118.84	118.86	109.33	1012.7	1122.0	0.2009	1.6847	1.8856	3.0
4.0	152.99	0.01636	90.72	90.74	120.83	1005.9	1126.8	0.2198	1.6420	1.8618	4.0
5.0	162.25	0.01641	73.59	73.61	130.10	1000.4	1130.6	0.2348	1.6088	1.8435	5.0
6.0	170.07	0.01645	62.03	62.05	137.92	995.8	1133.7	0.2473	1.5814	1.8287	6.0
7.0	176.85	0.01649	53.68	53.70	144.71	991.7	1136.4	0.2580	1.5582	1.8162	7.0
8.0	182.87	0.01652	47.38	47.39	150.75	988.1	1138.9	0.2674	1.5379	1.8053	8.0
9.0	188.28	0.01656	42.42	42.44	156.19	984.8	1141.0	0.2758	1.5200	1.7958	9.0
10.0	193.21	0.01658	38.44	38.45	161.13	981.8	1143.0	0.2834	1.5040	1.7874	10.0
11.0	197.75	0.01661	35.15	35.17	165.68	979.1	1144.8	0.2903	1.4894	1.7797	11.0
12.0	201.96	0.01664	32.40	32.42	169.91	976.5	1146.4	0.2968	1.4760	1.7727	12.0
13.0	205.88	0.01666	30.06	30.08	173.85	974.1	1147.9	0.3027	1.4636	1.7663	13.0
14.0	209.56	0.01669	28.05	28.06	177.55	971.8	1149.3	0.3082	1.4521	1.7604	14.0
14.696	212.00	0.01670	26.80	26.82	180.00	970.2	1150.2	0.3119	1.4446	1.7564	14.696
16.0	216.32	0.01673	24.75	24.76	184.35	967.4	1151.8	0.3184	1.4312	1.7496	16.0
18.0	222.40	0.01678	22.16	22.18	190.48	963.5	1154.0	0.3274	1.4127	1.7402	18.0
20.0	227.96	0.01682	20.078	20.095	196.09	959.9	1156.0	0.3356	1.3960	1.7317	20.0
22.0	233.07	0.01685	18.363	18.380	201.25	956.6	1157.8	0.3431	1.3809	1.7240	22.0
24.0	237.82	0.01689	16.924	16.941	206.05	953.4	1159.5	0.3500	1.3670	1.7170	24.0
26.0	242.25	0.01692	15.701	15.718	210.54	950.4	1161.0	0.3564	1.3542	1.7106	26.0
28.0	246.41	0.01695	14.647	14.664	214.75	947.7	1162.4	0.3624	1.3422	1.7046	28.0
30.0	250.34	0.01698	13.728	13.745	218.73	945.0	1163.7	0.3680	1.3310	1.6990	30.0
32.0	254.05	0.01701	12.923	12.940	222.50	942.5	1165.0	0.3732	1.3206	1.6938	32.0
34.0	257.58	0.01704	12.209	12.226	226.09	940.0	1166.1	0.3783	1.3107	1.6890	34.0
36.0	260.94	0.01707	11.570	11.587	229.51	937.7	1167.2	0.3830	1.3014	1.6844	36.0
38.0	264.16	0.01710	10.998	11.015	232.79	935.5	1168.3	0.3876	1.2925	1.6800	38.0
40.0	267.24	0.01712	10.480	10.497	235.93	933.3	1169.2	0.3919	1.2840	1.6759	40.0
42.0	270.21	0.01715	10.010	10.027	238.95	931.2	1170.2	0.3961	1.2759	1.6720	42.0
44.0	273.06	0.01717	9.582	9.599	241.86	929.2	1171.1	0.4000	1.2682	1.6683	44.0
46.0	275.81	0.01719	9.189	9.207	244.67	927.2	1171.9	0.4039	1.2608	1.6647	46.0
48.0	278.45	0.01722	8.829	8.846	247.37	925.4	1172.7	0.4076	1.2537	1.6613	48.0
50.0	281.01	0.01724	8.496	8.514	249.98	923.5	1173.5	0.4111	1.2469	1.6580	50.0
52.0	283.49	0.01726	8.189	8.206	252.52	921.7	1174.3	0.4145	1.2404	1.6549	52.0
54.0	285.90	0.01728	7.902	7.919	254.99	920.0	1175.0	0.4178	1.2340	1.6518	54.0
56.0	288.23	0.01730	7.636	7.653	257.38	918.3	1175.7	0.4210	1.2279	1.6489	56.0
58.0	290.50	0.01732	7.388	7.405	259.71	916.6	1176.4	0.4241	1.2220	1.6461	58.0
60.0	292.71	0.01735	7.155	7.172	261.98	915.0	1177.0	0.4271	1.2162	1.6434	60.0
62.0	294.85	0.01737	6.937	6.955	264.18	913.4	1177.6	0.4300	1.2107	1.6407	62.0
64.0	296.94	0.01739	6.732	6.749	266.33	911.9	1178.2	0.4329	1.2053	1.6382	64.0
66.0	298.98	0.01741	6.539	6.556	268.43	910.4	1178.8	0.4356	1.2001	1.6357	66.0
68.0	300.98	0.01743	6.357	6.375	270.49	908.9	1179.4	0.4384	1.1950	1.6333	68.0
70.0	302.92	0.01744	6.186	6.203	272.49	907.4	1179.9	0.4410	1.1900	1.6310	70.0
72.0	304.82	0.01746	6.024	6.041	274.45	906.0	1180.5	0.4435	1.1852	1.6287	72.0
74.0	306.68	0.01748	5.870	5.887	276.37	904.6	1181.0	0.4460	1.1805	1.6265	74.0
76.0	308.50	0.01750	5.723	5.741	278.25	903.2	1181.5	0.4485	1.1759	1.6244	76.0
78.0	310.28	0.01752	5.584	5.602	280.09	901.9	1182.0	0.4509	1.1714	1.6223	78.0
80.0	312.03	0.01754	5.452	5.470	281.90	900.5	1182.4	0.4532	1.1670	1.6202	80.0
82.0	313.74	0.01756	5.325	5.343	283.67	899.2	1182.9	0.4555	1.1627	1.6182	82.0
84.0	315.42	0.01757	5.204	5.222	285.42	897.9	1183.4	0.4578	1.1586	1.6163	84.0
86.0	317.06	0.01759	5.089	5.107	287.13	896.7	1183.8	0.4599	1.1545	1.6144	86.0
88.0	318.68	0.01761	4.979	4.997	288.80	895.4	1184.2	0.4621	1.1505	1.6126	88.0
90.0	320.27	0.01763	4.874	4.892	290.45	894.2	1184.6	0.4642	1.1465	1.6107	90.0
92.0	321.83	0.01764	4.773	4.791	292.07	893.0	1185.0	0.4663	1.1427	1.6090	92.0
94.0	323.37	0.01766	4.676	4.694	293.67	891.8	1185.4	0.4683	1.1389	1.6072	94.0
96.0	324.88	0.01768	4.584	4.602	295.25	890.6	1185.8	0.4703	1.1352	1.6055	96.0
98.0	326.37	0.01769	4.494	4.512	296.80	889.4	1186.2	0.4723	1.1316	1.6038	98.0

[a]Abstracted from *Steam Tables and Mollier Diagram*, by Prof. J. H. Keenan, 1930 edition, by permission the publisher, *The American Society of Mechanical Engineers*.

TABLE 8. PROPERTIES OF SATURATED STEAM: PRESSURE TABLE—(Continued)

Abs Press. Lb/Sq In	Temp Deg F	Specific Volume: Sat Liquid	Specific Volume: Evap.	Specific Volume: Sat. Vapor	Total Heat: Sat. Liquid	Total Heat: Evap.	Total Heat: Sat. Vapor	Entropy: Sat. Liquid	Entropy: Evap.	Entropy: Sat. Vapor	Abs Press. Lb/Sq. In.
p	t	v_f	v_{fg}	v_g	h_f	h_{fg}	h_g	s_f	s_{fg}	s_g	p
100.0	327.83	0.01771	4.408	4.426	298.33	888.2	1186.6	0.4742	1.1280	1.6022	100.0
102.0	329.27	0.01773	4.326	4.344	299.83	887.1	1186.9	0.4761	1.1245	1.6006	102.0
104.0	330.68	0.01774	4.247	4.265	301.30	886.0	1187.3	0.4779	1.1211	1.5990	104.0
106.0	332.08	0.01776	4.171	4.189	302.76	884.9	1187.6	0.4798	1.1177	1.5974	106.0
108.0	333.44	0.01777	4.097	4.115	304.19	883.8	1188.0	0.4816	1.1144	1.5959	108.0
110.0	334.79	0.01779	4.026	4.044	305.61	882.7	1188.3	0.4834	1.1111	1.5944	110.0
112.0	336.12	0.01780	3.958	3.976	307.00	881.6	1188.6	0.4851	1.1079	1.5930	112.0
114.0	337.43	0.01782	3.892	3.910	308.36	880.6	1188.9	0.4868	1.1048	1.5915	114.0
116.0	338.72	0.01783	3.828	3.846	309.71	879.5	1189.2	0.4885	1.1017	1.5901	116.0
118.0	340.01	0.01785	3.766	3.784	311.05	878.5	1189.5	0.4901	1.0986	1.5887	118.0
120.0	341.26	0.01786	3.707	3.725	312.37	877.4	1189.8	0.4918	1.0956	1.5874	120.0
122.0	342.50	0.01788	3.652	3.670	313.67	876.4	1190.1	0.4934	1.0926	1.5860	122.0
124.0	343.73	0.01789	3.597	3.615	314.96	875.4	1190.4	0.4950	1.0897	1.5847	124.0
126.0	344.94	0.01791	3.542	3.560	316.23	874.4	1190.6	0.4965	1.0868	1.5834	126.0
128.0	346.14	0.01792	3.487	3.505	317.49	873.4	1190.9	0.4981	1.0840	1.5821	128.0
130.0	347.31	0.01794	3.433	3.451	318.73	872.4	1191.2	0.4996	1.0812	1.5808	130.0
132.0	348.48	0.01795	3.383	3.401	319.95	871.5	1191.4	0.5011	1.0784	1.5796	132.0
134.0	349.64	0.01796	3.335	3.353	321.17	870.5	1191.7	0.5026	1.0757	1.5783	134.0
136.0	350.78	0.01798	3.288	3.306	322.37	869.6	1191.9	0.5041	1.0730	1.5771	136.0
138.0	351.91	0.01799	3.242	3.260	323.56	868.6	1192.2	0.5056	1.0703	1.5759	138.0
140.0	353.03	0.01801	3.198	3.216	324.74	867.7	1192.4	0.5070	1.0677	1.5747	140.0
142.0	354.14	0.01802	3.155	3.173	325.91	866.7	1192.6	0.5084	1.0651	1.5735	142.0
144.0	355.22	0.01804	3.112	3.130	327.06	865.8	1192.9	0.5098	1.0625	1.5724	144.0
146.0	356.31	0.01805	3.071	3.089	328.20	864.9	1193.1	0.5112	1.0600	1.5712	146.0
148.0	357.37	0.01806	3.031	3.049	329.32	864.0	1193.3	0.5126	1.0575	1.5701	148.0
150.0	358.43	0.01808	2.992	3.010	330.44	863.1	1193.5	0.5140	1.0550	1.5690	150.0
152.0	359.47	0.01809	2.954	2.972	331.54	862.2	1193.7	0.5153	1.0526	1.5679	152.0
154.0	360.51	0.01810	2.917	2.935	332.64	861.3	1193.9	0.5166	1.0502	1.5668	154.0
156.0	361.53	0.01812	2.882	2.900	333.72	860.4	1194.1	0.5180	1.0478	1.5658	156.0
158.0	362.54	0.01813	2.846	2.864	334.80	859.5	1194.3	0.5193	1.0454	1.5647	158.0
160.0	363.55	0.01814	2.812	2.830	335.86	858.7	1194.5	0.5205	1.0431	1.5636	160.0
162.0	364.54	0.01816	2.779	2.797	336.91	857.8	1194.7	0.5218	1.0408	1.5626	162.0
164.0	365.52	0.01817	2.746	2.764	337.95	857.0	1194.9	0.5230	1.0385	1.5616	164.0
166.0	366.50	0.01818	2.715	2.733	338.99	856.1	1195.1	0.5243	1.0363	1.5606	166.0
168.0	367.46	0.01819	2.683	2.701	340.01	855.2	1195.3	0.5255	1.0340	1.5596	168.0
170.0	368.42	0.01821	2.653	2.671	341.03	854.4	1195.4	0.5268	1.0318	1.5586	170.0
172.0	369.37	0.01822	2.623	2.641	342.04	853.6	1195.6	0.5280	1.0296	1.5576	172.0
174.0	370.31	0.01823	2.594	2.612	343.04	852.7	1195.8	0.5292	1.0275	1.5566	174.0
176.0	371.24	0.01825	2.566	2.584	344.03	851.9	1196.0	0.5304	1.0253	1.5557	176.0
178.0	372.16	0.01826	2.538	2.556	345.01	851.1	1196.1	0.5315	1.0232	1.5548	178.0
180.0	373.08	0.01827	2.511	2.529	345.99	850.3	1196.3	0.5327	1.0211	1.5538	180.0
182.0	374.00	0.01828	2.484	2.502	346.97	849.5	1196.4	0.5339	1.0190	1.5529	182.0
184.0	374.90	0.01829	2.458	2.476	347.94	848.6	1196.6	0.5350	1.0169	1.5520	184.0
186.0	375.78	0.01831	2.433	2.451	348.89	847.9	1196.8	0.5362	1.0149	1.5511	186.0
188.0	376.67	0.01832	2.407	2.425	349.83	847.1	1196.9	0.5373	1.0129	1.5502	188.0
190.0	377.55	0.01833	2.383	2.401	350.77	846.3	1197.0	0.5384	1.0109	1.5493	190.0
192.0	378.42	0.01834	2.359	2.377	351.70	845.5	1197.2	0.5395	1.0089	1.5484	192.0
194.0	379.27	0.01835	2.335	2.353	352.61	844.7	1197.3	0.5406	1.0070	1.5475	194.0
196.0	380.13	0.01837	2.312	2.330	353.53	844.0	1197.5	0.5417	1.0050	1.5467	196.0
198.0	380.97	0.01838	2.289	2.307	354.43	843.2	1197.6	0.5427	1.0031	1.5458	198.0
200.0	381.82	0.01839	2.267	2.285	355.33	842.4	1197.8	0.5438	1.0012	1.5450	200.0
205.0	383.89	0.01842	2.213	2.231	357.56	840.5	1198.1	0.5465	0.9964	1.5429	205.0
210.0	385.93	0.01844	2.162	2.180	359.76	838.6	1198.4	0.5491	0.9918	1.5409	210.0
215.0	387.93	0.01847	2.113	2.131	361.91	836.8	1198.7	0.5516	0.9873	1.5389	215.0
220.0	389.89	0.01850	2.066	2.084	364.02	835.0	1199.0	0.5540	0.9829	1.5369	220.0
225.0	391.81	0.01853	2.0208	2.0393	366.10	833.2	1199.3	0.5565	0.9786	1.5350	225.0
230.0	393.70	0.01856	1.9778	1.9964	368.14	831.4	1199.6	0.5588	0.9743	1.5332	230.0
235.0	395.56	0.01859	1.9367	1.9553	370.15	829.7	1199.8	0.5612	0.9702	1.5313	235.0
240.0	397.40	0.01861	1.8970	1.9156	372.13	827.9	1200.1	0.5635	0.9661	1.5295	240.0
245.0	399.20	0.01864	1.8589	1.8775	374.09	826.2	1200.3	0.5658	0.9620	1.5278	245.0

TABLE 8. PROPERTIES OF SATURATED STEAM: PRESSURE TABLE—(Concluded)

Abs Press. Lb /Sq In.	Temp Deg F	Specific Volume			Total Heat			Entropy			Abs Press. Lb /Sq In.
		Sat Liquid	Evap	Sat Vapor	Sat Liquid	Evap	Sat Vapor	Sat. Liquid	Evap	Sat. Vapor	
p	t	v_f	v_{fg}	v_g	h_f	h_{fg}	h_g	s_f	s_{fg}	s_g	p
250.0	400.97	0.01867	1.8223	1.8410	376.02	824.5	1200.5	0.5680	0.9581	1.5261	250.0
260.0	404.43	0.01872	1.7536	1.7723	379.78	821.2	1201.0	0.5723	0.9504	1.5227	260.0
270.0	407.79	0.01877	1.6895	1.7083	383.44	818.0	1201.4	0.5765	0.9430	1.5194	270.0
280.0	411.06	0.01882	1.6302	1.6490	387.02	814.7	1201.8	0.5805	0.9357	1.5163	280.0
290.0	414.24	0.01887	1.5745	1.5934	390.50	811.6	1202.1	0.5845	0.9287	1.5132	290.0
300.0	417.33	0.01892	1.5225	1.5414	393.90	808.5	1202.4	0.5883	0.9220	1.5102	300.0
320.0	423.29	0.01901	1.4279	1.4469	400.47	802.5	1203.0	0.5957	0.9089	1.5046	320.0
340.0	428.96	0.01910	1.3439	1.3630	406.75	796.6	1203.4	0.6027	0.8965	1.4992	340.0
360.0	434.39	0.01918	1.2689	1.2881	412.80	790.9	1203.7	0.6094	0.8846	1.4940	360.0
380.0	439.59	0.01927	1.2015	1.2208	418.61	785.3	1203.9	0.6157	0.8733	1.4891	380.0
400.0	444.58	0.0194	1.1407	1.1601	424.2	779.8	1204.1	0.6218	0.8625	1.4843	400.0
420.0	449.38	0.0194	1.0853	1.1047	429.6	774.5	1204.1	0.6277	0.8520	1.4798	420.0
440.0	454.01	0.0195	1.0345	1.0540	434.8	769.3	1204.1	0.6334	0.8420	1.4753	440.0
460.0	458.48	0.0196	0.9881	1.0077	439.9	764.1	1204.0	0.6388	0.8322	1.4711	460.0
480.0	462.80	0.0197	0.9456	0.9633	444.9	759.0	1203.9	0.6441	0.8228	1.4670	480.0
500.0	466.99	0.0198	0.9063	0.9261	449.7	754.0	1203.7	0.6493	0.8137	1.4630	500.0
520.0	471.05	0.0198	0.8701	0.8899	454.4	749.0	1203.5	0.6543	0.8048	1.4591	520.0
540.0	474.99	0.0199	0.8363	0.8562	459.0	744.1	1203.2	0.6592	0.7962	1.4554	540.0
560.0	478.82	0.0200	0.8047	0.8247	463.6	739.3	1202.9	0.6639	0.7878	1.4517	560.0
580.0	482.55	0.0201	0.7751	0.7952	468.0	734.5	1202.5	0.6686	0.7796	1.4482	580.0
600.0	486.17	0.0202	0.7475	0.7677	472.3	729.8	1202.1	0.6731	0.7716	1.4447	600.0
620.0	489.71	0.0202	0.7217	0.7419	476.6	725.1	1201.7	0.6775	0.7638	1.4413	620.0
640.0	493.16	0.0203	0.6972	0.7175	480.8	720.5	1201.2	0.6818	0.7562	1.4380	640.0
660.0	496.53	0.0204	0.6744	0.6948	484.9	715.9	1200.8	0.6861	0.7487	1.4348	660.0
680.0	499.82	0.0205	0.6527	0.6732	488.9	711.3	1200.2	0.6902	0.7414	1.4316	680.0
700.0	503.04	0.0206	0.6321	0.6527	492.9	706.8	1199.7	0.6943	0.7342	1.4285	700.0
720.0	506.19	0.0206	0.6128	0.6334	496.8	702.4	1199.2	0.6983	0.7272	1.4255	720.0
740.0	509.28	0.0207	0.5944	0.6151	500.6	697.9	1198.6	0.7022	0.7203	1.4225	740.0
760.0	512.30	0.0208	0.5769	0.5977	504.4	693.5	1198.0	0.7060	0.7136	1.4196	760.0
780.0	515.27	0.0209	0.5602	0.5811	508.2	689.2	1197.4	0.7098	0.7069	1.4167	780.0
800.0	518.18	0.0209	0.5444	0.5653	511.8	684.9	1196.7	0.7135	0.7004	1.4139	800.0
820.0	521.03	0.0210	0.5293	0.5503	515.5	680.6	1196.0	0.7171	0.6940	1.4111	820.0
840.0	523.83	0.0211	0.5149	0.5360	519.0	676.4	1195.4	0.7207	0.6877	1.4084	840.0
860.0	526.58	0.0212	0.5013	0.5225	522.6	672.1	1194.7	0.7242	0.6815	1.4057	860.0
880.0	529.29	0.0213	0.4881	0.5094	526.0	667.9	1194.0	0.7277	0.6754	1.4031	880.0
900.0	531.95	0.0213	0.4756	0.4969	529.5	663.8	1193.3	0.7311	0.6694	1.4005	900.0
920.0	534.56	0.0214	0.4635	0.4849	532.9	659.7	1192.6	0.7344	0.6635	1.3980	920.0
940.0	537.13	0.0215	0.4520	0.4735	536.2	655.6	1191.8	0.7377	0.6577	1.3954	940.0
960.0	539.66	0.0216	0.4409	0.4625	539.6	651.5	1191.1	0.7410	0.6520	1.3930	960.0
980.0	542.14	0.0217	0.4303	0.4520	542.8	647.5	1190.3	0.7442	0.6464	1.3905	980.0
1000.0	544.58	0.0217	0.4202	0.4419	546.0	643.5	1189.6	0.7473	0.6408	1.3881	1000.0
1050.0	550.53	0.0219	0.3960	0.4179	554.0	633.6	1187.6	0.7550	0.6273	1.3822	1050.0
1100.0	556.28	0.0222	0.3738	0.3960	561.7	623.9	1185.6	0.7624	0.6141	1.3765	1100.0
1150.0	561.81	0.0224	0.3540	0.3764	569.2	614.3	1183.5	0.7695	0.6014	1.3709	1150.0
1200.0	567.14	0.0226	0.3356	0.3582	576.5	604.9	1181.4	0.7764	0.5891	1.3656	1200.0
1250.0	572.30	0.0228	0.3187	0.3415	583.6	595.6	1179.2	0.7831	0.5772	1.3603	1250.0
1300.0	577.32	0.0230	0.3029	0.3259	590.6	586.3	1177.0	0.7897	0.5654	1.3552	1300.0
1350.0	582.21	0.0232	0.2884	0.3116	597.5	577.2	1174.7	0.7962	0.5540	1.3501	1350.0
1400.0	586.96	0.0235	0.2748	0.2983	604.3	568.1	1172.4	0.8024	0.5428	1.3452	1400.0
1450.0	591.58	0.0237	0.2621	0.2858	611.0	559.1	1170.0	0.8086	0.5318	1.3404	1450.0
1500.0	596.08	0.0239	0.2502	0.2741	617.5	550.2	1167.6	0.8146	0.5212	1.3357	1500.0
1600.0	604.74	0.0244	0.2284	0.2528	630.2	532.6	1162.7	0.8262	0.5003	1.3265	1600.0
1700.0	612.98	0.0249	0.2089	0.2338	642.5	515.0	1157.5	0.8373	0.4801	1.3174	1700.0
1800.0	620.86	0.0254	0.1913	0.2167	654.7	497.2	1151.8	0.8482	0.4601	1.3083	1800.0
1900.0	628.39	0.0260	0.1754	0.2014	666.8	478.9	1145.7	0.8589	0.4402	1.2990	1900.0
2000.0	635.6	0.0265	0.1610	0.1875	679.0	460.0	1139.0	0.8696	0.4200	1.2896	2000.0
2200.0	649.2	0.0277	0.1346	0.1623	703.7	420.0	1123.8	0.8912	0.3788	1.2700	2200.0
2400.0	661.9	0.0292	0.1112	0.1404	729.4	376.4	1105.8	0.9133	0.3356	1.2488	2400.0
2600.0	673.8	0.0310	0.0895	0.1205	756.7	327.8	1084.5	0.9364	0.2892	1.2257	2600.0
2800.0	684.9	0.0333	0.0688	0.1021	786.7	272.3	1058.9	0.9618	0.2379	1.1996	2800.0
3000.0	695.2	0.0367	0.0477	0.0844	823.1	202.5	1025.6	0.9922	0.1754	1.1676	3000.0
3200.0	704.9	0.0459	0.0142	0.0601	887.0	75.9	962.9	1.0461	0.0651	1.1112	3200.0
3226.0	706.1	0.0522	0	0.0522	925.0	0	925.0	1.0785	0	1.0785	3226.0

responding change in the latent heat of evaporation. These values are given in Table 8.

Specific Heat. The specific heat of water, or the amount of heat (Btu) required to raise the temperature of one pound of water one degree Fahrenheit, varies with the temperature, but it is commonly assumed to be unity at all temperatures. Steam tables are based on exact values, however. The specific heat of ice at 32 F is 0.492 Btu per pound. The amount of heat required to raise one pound of water at 32 F through a known temperature interval depends on the average specific heat for the temperature range.

Sensible and Latent Heat. The heat necessary to raise the temperature of one pound of water from 32 F to the boiling point is known as the *heat of the liquid* or *sensible heat.* When more heat is added, the water begins to evaporate and expand at constant temperature until the water is entirely changed into steam. The heat thus added is known as the *latent heat of evaporation.*

PROPERTIES OF STEAM

Steam is water vapor which exists in the vaporous condition because sufficient heat has been added to the water to supply the latent heat of evaporation and change the liquid into vapor. This change in state takes place at a definite and constant temperature which is determined solely by the pressure of the steam. The volume of a pound of steam is the *specific volume* which decreases as the pressure increases. The reciprocal of this, or the weight of steam per cubic foot, is the *density.* (See Table 8.)

Steam which is in contact with the water from which it was generated is known as *saturated steam.* If it contains no actual water in the form of mist or priming, it is called *dry saturated steam.* If this be heated and the pressure maintained the same as when it was vaporized, its temperature will increase and it will become *superheated,* that is, its temperature will be higher than that of saturated steam at the same pressure.

REFERENCES

Rational Psychrometric Formulae, by W. H. Carrier (*A.S M E Transactions*, Vol 33, 1911).

Temperature of Evaporation, by W. H. Carrier (A.S.H.V.E. TRANSACTIONS, Vol. 24, 1918).

The Evaporation of a Liquid into a Gas, by W. K. Lewis (*A S M.E Transactions*, Vol. 44, 1922).

The Evaporation of a Liquid into a Gas—a Correction, by W. K. Lewis (*Mechanical Engineering*, September, 1933).

Temperature of Evaporation of Water into Air, by W H Carrier and D. C. Lindsay (*A.S.M.E. Transactions*, Vol. 46, 1924).

A New Psychrometric Chart, by C A. Bulkeley (A.S H V.E. TRANSACTIONS, Vol 32, 1926).

A Review of Psychrometric Charts, by C. O. Mackey (*Heating and Ventilating*, June, July, 1931).

Air Conditioning Applied to Cold Storage and a New Psychrometric Chart, by C. A. Bulkeley (*Refrigerating Engineering*, February, 1932).

The Psychrometric Chart, by E V. Hill (*Aerologist*, April, May, June, 1932).

Air Conditioning Theory, by John A Goff (*Refrigerating Engineering*, January, 1933)

Mixtures of Air and Water Vapor, by C A Bulkeley (*Refrigerating Engineering*, January, 1933)

Basic Theory of Air Conditioning, by Lawrence Washington (Western Conference on Air Conditioning, San Francisco, Calif, February 9-10, 1933)

The Theory of the Psychrometer, by J H Arnold (*Physics*, July, September, 1933)

Heat Transmission in Cooling Air with Extended Surfaces, by W L. Knaus (*Refrigerating Engineering*, January, February, 1935)

A New Psychrometric Chart, by F O Urban (*Refrigerating Engineering*, November 1935)

Principles of Engineering Thermodynamics, by Kiefer and Stuart

Chemical Engineering, by Lewis, Walker and McAdams

Fan Engineering, Buffalo Forge Co

Heat Transmission, by W H McAdams

International Critical Tables, 1928.

Experimental Mechanical Engineering, by H. Diederichs and W C. Andrae.

Psychrometric Charts, by Donald B. Brooks, *Bureau of Standards Miscellaneous Publication* No 143.

The Deviation of the Actual Wet-Bulb Temperature from the Temperature of Adiabatic Saturation, by David Dropkin (*Cornell University Engr Exp Station Bul.* No 23, July, 1936).

PROBLEMS IN PRACTICE

1 ● Given air at 70 F dry-bulb and 50 per cent relative humidity with a barometric pressure of 29.00 in. Hg, find the weight of vapor per pound of dry air.

Pressure of saturated vapor = e_t = 0.7387 in. Hg (Table 6).
From Equation 5a,

$$W = 0.622 \left(\frac{0.5 \times 0.7387}{29.00 - (0.5)(0.7387)}\right)$$

W = 0.008024 lb of vapor per pound of dry air at 70 F dry-bulb and 50 per cent relative humidity.

Approximate Method

Weight of saturated vapor per pound of dry air = W_t = 0.01574 lb (Table 6). 0.01574 $\times$ 0.5 = 0.0787 lb of vapor per pound of dry air at 70 F dry-bulb and 50 per cent relative humidity.

2 ● Given air with a dry-bulb temperature of 80 F, relative humidity of 55 per cent, and a barometric pressure of 28.85 in. Hg, calculate the weight of a cubic foot of mixture.

Pressure of saturated vapor at 80 F = e_t = 1.0316 in. Hg (Table 6)

Pressure of the vapor in the mixture = 1.0316 $\times$ 0 55 = 0.5676 in. Hg.

Pressure of the dry air in the mixture = 28.85 − 0.5676 = 28.282 in. Hg.

$pV = wR\ (t + 460)$ (R = 0.753 when partial pressure of air is expressed in in. Hg).
28.282 $\times$ 1 = $d_a \times$ 0.753 $\times$ (80 + 460)

$$d_a = \frac{28.282}{0.753 \times 540} = 0.06955 \text{ lb} = \text{weight of dry air in 1 cu ft of the mixture.}$$

Likewise from Equation 4a,

$$d_v = \frac{0.5676}{1.21 \times 540} = 0.000868 \text{ lb} = \text{weight of vapor per cubic feet at 55 per cent}$$
relative humidity.

Weight of 1 cu ft of the mixture = 0.06955 + 0.000868 = 0.070418 lb.

3 ● Given air with a dry-bulb temperature of 75 F, a relative humidity of 60 per cent, and a barometric pressure of 28.80 in. Hg, calculate the volume of 1 lb of the mixture.

Pressure of saturated vapor at 75 F $= e_t = 0.8745$ in Hg

Pressure of vapor in the mixture $= 0.8745 \times 0.6 = 0.525$ in Hg

Pressure of dry air in the mixture $= 28.80 - 0.525 = 28.275$ in. Hg

$$d_a = \frac{28.275}{0.753 \times 535} = 0.07018 \text{ lb} = \text{weight of dry air in 1 cu ft of the mixture}$$

From Equation 4a,

$$d_v = \frac{0.525}{1.21 \times 535} = 0.000811 \text{ lb} = \text{weight of vapor per cubic feet at 55 per cent}$$

relative humidity

Weight of 1 cu ft of the mixture $= 0.07018 + 0.000811 = 0.070991$ lb

Volume of 1 lb of the mixture $= \dfrac{1}{0.070991} = 14.08$ cu ft

4 ● It is desired to maintain a temperature of 80 F and a relative humidity of 50 per cent in a factory where the equipment gives off 6,000 Btu per hour. If the entering air is at 70 F with an average barometric pressure of 29.92 in. Hg; determine the relative humidity, and the pounds of air required per hour if there is no heat interchange between the walls, windows, or floors of the building.

Pressure of saturated vapor at 80 F $= 1.0316$ in. Hg (Table 6).

Pressure of vapor in the mixture $= 1.0316 \times 0.5 = 0.5158$ in. Hg.

$$W = 0.622\left(\frac{0.5158}{29.92 - 0.5158}\right) = 0.01091 \text{ lb.}$$

Pressure of saturated vapor at 70 F $= 0.7387$ in Hg.

With the same specific humidity

$$0.01091 = 0.622\left(\frac{0.7387 \times \Phi}{29.92 - (0.7387 \times \Phi)}\right)$$

$\Phi = 69.8$ per cent relative humidity at 70 F.

$h = 0.24 \times 80 + 0.01091\,(1059.2 + 0.45 \times 80) = 31.15$ Btu per pound, the heat content of the mixture at 80 F and 50 per cent relative humidity

$h = 0.24 \times 70 + 0.01091\,(1059.2 + 0.45 \times 70) = 28.70$ Btu per pound, the heat content of the mixture at 70 F and the same specific humidity.

$31.15 - 28.70 = 2.45$ Btu to be removed per pound of air

6000 Btu = heat given off by equipment per hour.

$\dfrac{6000}{2.45} = 2449$ lb of air required per hour.

5 ● Given 1 lb of dry air at 78 F and a barometric pressure of 29.92 in. Hg; calculate the volume. If the temperature is raised to 96 F and the volume remains constant, what will be the new pressure, P_2, in in. Hg?

$PV = wR\,(t + 460)$

R (for air) $= 53.34$.

$W = 1$ lb.

$P =$ absolute pressure, pounds per square foot

$$V = \frac{1 \times 53.34 \times (78 + 460)}{29.92 \times 0.491 \times 144}$$

$V = 13.57$ cu ft $=$ volume of 1 lb

$$\frac{P_2}{P_1} = \frac{T_2}{T_1}; \quad P_2 = \frac{T_2 P_1}{T_1}$$

$$P_2 = \frac{(96 + 460)\ (29\ 92 \times 0\ 491 \times 144)}{(78 + 460)\ (0\ 491 \times 144)} = 30.90 \text{ in. Hg.}$$

6 ● Given saturated air at a temperature of 75 F and a barometric pressure of 29.92 in. Hg; determine the heat content of the mixture per pound of dry air, including the heat content of the liquid above 32 F.

From Equation 12,

$h = 0.24\ (t - 0) + Wh_s$.

where

h_s = 1059.2 + 0.45t (Empirical equation derived from Keenan's Steam Tables).
t = 75 F.
W = 0.01873 lb of water vapor (Table 6)
h = 0 24 (75 − 0) + 0.01873 (1059.2 + 0.45 × 75).
h = 38 46 Btu per pound of dry air.

7 ● A building requires 50,000 cu ft of air per hour to be raised from −10 F dry-bulb and 75 per cent relative humidity to 72 F dry-bulb and 30 per cent relative humidity. Determine the amount of heat and the weight of water which it is necessary to supply per hour if the temperature of the supply water is 50 F and the barometric pressure is 28.75 in. Hg.

Assume air volume to be dry air at 70 F.

Weight of air = 0 075 × 50,000 = 3750 lb per hour.

From Table 6,

Pressure of vapor in the mixture, outside air = 0.75 × 0.0221 = 0 0166 in. Hg.

Specific humidity, outside air $= 0\ 622 \left(\frac{0.0166}{28\ 75 - 0\ 0166}\right) = 0\ 0003589$ lb.

From Table 6,

Pressure of vapor in the mixture, inside air = 0.30 × 0.7906 = 0.2372 in. Hg.

Specific humidity, inside air $= 0.622 \left(\frac{0.2372}{28.75 - 0.23718}\right) = 0\ 005174$ lb.

Water to be added = 3750 (0 005174 − 0.0003589) = 18.06 lb per hour.

Heat content, inside air = 0 24 × 72 + 0.005174 (1059.2 + 0.45 × 72) = 22.925 Btu per pound.

Heat content, outside air = 0.24 × (−10) + 0.0003589 (1059.2 + (0.45 × −10) = −2.021 Btu per pound.

Btu added incident to the water per pound of dry air.

(0 005174 − 0.0003589) (50 − 32) = 0 0867 Btu per pound.

Heat requirement per hour = (22.925 − (−2 021 + 0.0867) × 3750 = 93,221 Btu.

8 ● Determine the amount of heat and water that must be extracted to cool 3750 lb of air (weighed dry) from 95 F and 60 per cent relative humidity to 50 F and 100 per cent relative humidity with a barometric pressure of 28.75 in. Hg.

Pressure of vapor in the mixture, outside air = 0.6 × 1.659 = 0.995 in. Hg.

Specific humidity, outside air $= 0.622 \left(\frac{0\ 995}{28.75 - 0.995}\right) = 0.0223$ lb.

Specific humidity, inside air = 0.007626 lb.

Weight of water to be extracted per hour = (0 0223 − 0.007626) × 3750 = 55.03 lb.

Heat content, outside air = 0 24 × 95 + 0.0223 (1059.2 + 0.45 × 95) = 47.37 Btu per pound

Heat content, inside air = 0.24 × 50 + 0.00764 (1059.2 + 0.45 × 50) = 20.26 Btu per pound.

Heat to be extracted = (47.37 − 20.26) × 3750 = 101,662 Btu.

Chapter 2

REFRIGERATION

Classification of Systems, Refrigerants, Mechanical Compression Systems, Theoretical Cycle, Refrigerating Effect per Pound, Coefficient of Performance, Steam Ejector System, System Characteristics, Closed Absorption Systems, Open Adsorption Systems, Reverse Cycle

THE various types of refrigeration systems most commonly used for air conditioning purposes may be classified fundamentally as follows:

1. Compression Systems
   a Mechanical—Reciprocating, Rotary, and Centrifugal
   b Ejector
2. Absorption Systems
   a. Closed.
   b. Open.

Of these, the mechanical systems are the most extensively used at the present time and will be given complete consideration in the following discussion.

REFRIGERANTS

The common refrigerants are volatile liquids which produce refrigeration by their evaporation under reduced pressure. Factors usually influencing the choice of refrigerant are safety, chemical stability, operating pressures and adaptability for the type of system to be used.

Of the six refrigerants whose properties are listed in Tables 1 to 6, ammonia, carbon dioxide, dichlorodifluoromethane (F_{12}) and methyl chloride are used in reciprocating and rotary mechanical compression systems. Monofluorotrichloromethane (F_{11}) and water are used in centrifugal compression systems. Water is used almost to the exclusion of other refrigerants in ejector systems. Closed absorption systems may use ammonia, methyl chloride, water, or others as the refrigerant.

MECHANICAL REFRIGERATION SYSTEMS

While the mechanical refrigeration systems differ in the methods used for compression of the refrigerant vapor, they are all fundamentally similar. Refrigerant vapor, usually saturated or slightly superheated, is drawn into a compressor (see Fig. 1). It is then compressed and discharged at a higher pressure to a condenser. The vapor is condensed as it contacts a heat transfer surface over which is flowing a cooling medium such as water, air or a combination of the two. The liquid refrigerant

Table 1. Properties of Ammonia

Sat Temp F	Abs Press Lb per Sq In	Volume		Heat Content and Entropy Taken From −40 F							
				Heat Content		Entropy		100 F Superheat		200 F Superheat	
		Liquid	Vapor	Liquid	Vapor	Liquid	Vapor	Ht Ct	Entropy	Ht Ct	Entropy
0	30.42	0 02419	9 116	42 9	611.8	0 0975	1 3352	666 8	1.4439	720 3	1 5317
5	34.27	0.02432	8.150	48.3	613 3	0.1092	1 3253	668 9	1 4339	722 7	1 5215
10	38.51	0 02446	7 304	53 8	614 9	0 1208	1.3157	670 9	1.4242	725 0	1 5115
15	43.14	0.02460	6.562	59 2	616 3	0 1323	1.3062	673 0	1 4148	727.3	1 5018
20	48 21	0 02474	5 910	64 7	617 8	0.1437	1 2969	675 0	1.4056	729 6	1 4925
25	53 73	0 02488	5.334	70 2	619.1	0 1551	1 2879	677 0	1 3965	731.9	1 4833
30	59.74	0 02503	4 825	75 7	620.5	0.1663	1.2790	678 9	1 3879	734.2	1 4744
35	66 26	0.02518	4 373	81 2	621.7	0.1775	1.2704	680 8	1.3794	736.5	1.4658
40	73.32	0.02533	3 971	86 8	623.0	0.1885	1.2618	682 7	1 3712	738 6	1 4575
45	80.96	0 02548	3 614	92.3	624.1	0.1996	1 2535	684 6	1.3630	740 9	1.4493
50	89.19	0.02564	3 294	97 9	625.2	0.2105	1.2453	686 4	1 3552	743.1	1.4412
55	98 06	0 02581	3 008	103.5	626 3	0.2214	1 2373	688 1	1.3474	745.3	1 4335
60	107.60	0 02597	2 751	109 2	627.3	0.2322	1.2294	689 9	1.3399	747.4	1.4260
65	117 80	0.02614	2.520	114.8	628 2	0.2430	1 2216	691 7	1.3326	749.5	1.4186
70	128.80	0 02632	2 312	120 5	629.1	0 2537	1 2140	693 3	1.3254	751 6	1.4114
75	140.50	0 02650	2.125	126 2	629 9	0.2643	1.2065	695 0	1 3184	753.7	1 4044
80	153.00	0 02668	1 955	132 0	630.7	0.2749	1.1991	696.6	1.3116	755 8	1.3976
85	166.40	0.02687	1 801	137 8	631.4	0.2854	1.1918	698.2	1 3048	757 9	1.3909
90	180.60	0 02707	1.661	143 5	632.0	0.2958	1 1846	699 7	1 2983	759.9	1.3843
95	195.80	0.02727	1 534	149.4	632 6	0.3062	1 1775	701 2	1.2919	761 9	1.3783
100	211 90	0.02747	1 419	155 2	633 0	0.3166	1 1705	702 7	1 2855	763.8	1 3718
105	228 90	0 02769	1 313	161.1	633.4	0.3269	1 1635	704 2	1 2793	765.7	1.3655

$n = \frac{C_p}{C_v} = 1\ 3172$

Table 2. Properties of Carbon Dioxide

Sat Temp. F	Abs Press Lb per Sq In	Volume		Heat Content and Entropy Taken From 40 F							
				Heat Content		Entropy		50 F Superheat		100 F Superheat	
		Liquid	Vapor	Liquid	Vapor	Liquid	Vapor	Ht. Ct	Entropy	Ht. Ct.	Entropy
0	305.5	0 01570	0.29040	18.8	138.9	0 0418	0.3024	153.7	0.3342	167.5	0.3612
5	332 0	0 01592	0.26610	20.3	138 8	0.0472	0 3000	153.7	0.3312	167.8	0.3582
10	360.2	0.01614	0.24370	24 0	138.7	0.0526	0.2970	153.7	0.3281	168.0	0.3550
15	390.0	0 01637	0.22360	26 7	138.5	0 0581	0 2939	153.7	0.3257	168.3	0.3517
20	421.8	0 01663	0 20490	29 4	138.3	0.0638	0.2909	153 7	0.3227	168.6	0.3489
25	455.3	0 01690	0 18790	32.3	138.1	0.0697	0.2879	153.7	0.3196	168.8	0.3464
30	490.8	0.01719	0.17220	35.4	137.8	0.0758	0.2849	153.7	0 3164	169 1	0.3441
35	529.3	0.01752	0.15770	38.5	137.3	0.0817	0 2813	153.7	0.3147	169.3	0.3415
40	567.8	0 01787	0.14440	41.7	136.7	0 0874	0.2776	153.7	0.3132	169.6	0.3391
45	609 6	0 01826	0.13210	45.0	135 9	0.0935	0 2739	153.7	0.3112	169.9	0.3365
50	653.6	0 01868	0.12050	48 4	135.0	0.1000	0 2699	153 7	0.3081	170.1	0.3342
55	700.0	0.01917	0 10960	51.9	133.7	0.1062	0 2656	153.7	0.3051	170.4	0.3320
60	748.6	0.01970	0.09940	55.5	132.1	0.1135	0.2608	153.7	0.3022	170.7	0.3297
65	799.8	0.02034	0.08990	59 4	130.2	0.1206	0.2554	153.7	0.2995	170.9	0.3277
70	853 4	0 02112	0.08040	63 7	127.5	0.1282	0.2487	153 7	0.2971	171.2	0.3257
75	909 7	0 02217	0.07072	68 4	123.7	0.1370	0.2404	153.7	0 2947	171.4	0.3237
80	968.7	0.02370	0.06064	73 9	118.7	0.1476	0 2304	153.7	0.2927	171.7	0.3220
85	1030 3	0 02620	0.05006	81.4	112.2	0.1668	0 2169	153.7	0.2909	180.0	0.3204
87.8	1069.9	0 03454	0 03454	97.0	97.0	0.1880	0 1880	153 7	0.2901	180.1	0.3199

$n = \frac{C_p}{C_v} = 1\ 28$

flows to the evaporator through an expansion valve which reduces its pressure and regulates its flow. The evaporator absorbs heat from a medium which is to be cooled. When this medium is water or brine, the evaporator is known as a water or brine cooler and the refrigeration system, if used for air cooling, is known as an indirect system. When the medium cooled is air, the evaporator is known as a direct expansion cooler and the system is known as a direct expansion system.

Fundamentally, the function of the system is to absorb heat at one temperature and *pump* it to a higher temperature, where it may be removed by an available cooling medium. In order to conserve refrigerant,

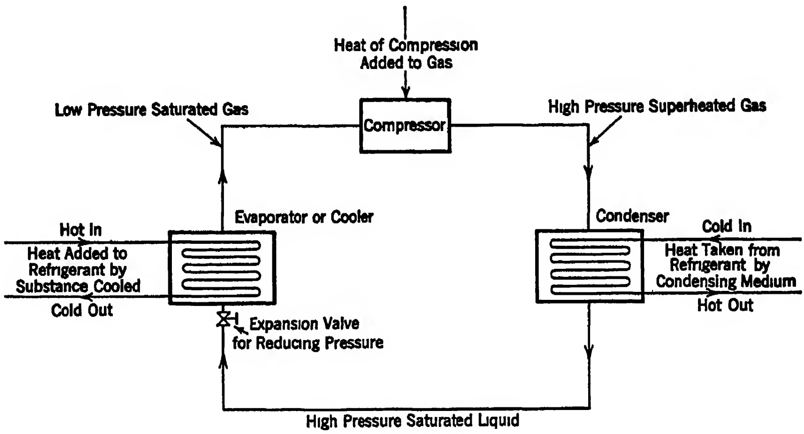


FIG 1 MECHANICAL REFRIGERATION SYSTEM

virtually all refrigeration systems are completely closed and the same refrigerant is recirculated.

Theoretical Mechanical Refrigeration Cycle

The complete mechanical refrigeration cycle may be illustrated on the temperature-entropy diagram, and also on the pressure-volume diagram both of which are shown in Fig. 2.

Considering the theoretical cycle, saturated vapor is drawn into the compressor at *a* and compressed at constant entropy (adiabatically) and then delivered to the condenser at *b*. Condensation occurs at constant temperature T_2 from *b* to *c* with a contraction from the vapor to the liquid volume. The line *cd* represents cooling from the temperature of the condenser to that of the evaporator by an external cooling means. At the same time, the pressure is lowered to P_1. Evaporation then occurs from *d* to *a* at temperature T_1, completing the work cycle *abcda*. Since no external means of cooling the refrigerant liquid is normally available, the cooling is generally accomplished by evaporation of a portion of the refrigerant. Since the work of expansion is usually used up as friction in the expansion valve, this process is carried on at constant total heat, as represented by the line *ce* on the temperature-entropy diagram. Thus the refrigerating effect is represented by an area *eagfe*. While the normal

Table 3. Properties of Dichlorodifluoromethane (F_{12})

Sat Temp F	Abs Press Lb per Sq In	Volume Liquid	Volume Vapor	Heat Content and Entropy Taken From −40 F: Heat Content Liquid	Heat Content Vapor	Entropy Liquid	Entropy Vapor	25 F Superheat Ht. Ct	25 F Superheat Entropy	50 F Superheat Ht Ct	50 F Superheat Entropy
0	23 87	0 0110	1 637	8.25	78 21	0 01869	0.17091	81.71	0.17829	85 26	0 18547
5	26 51	0 0111	1 485	9 32	78 79	0 02097	0 17052	82 29	0.17786	85.89	0.18502
10	29 35	0 0112	1.351	10 39	79 36	0 02328	0 17015	82.90	0.17747	86 51	0.18460
15	32 44	0 0112	1 230	11 48	79 94	0 02556	0 16981	83 49	0 17710	87 13	0 18420
20	35 75	0 0113	1 121	12 55	80.49	0.02783	0 16949	84 09	0.17679	87.76	0 18382
25	39 33	0 0114	1 025	13 66	81 06	0 03008	0 16920	84.67	0 17643	88.37	0.18349
30	43 16	0 0115	0 939	14 76	81 61	0 03233	0 16887	85 25	0.17612	88.97	0.18315
35	47.28	0 0116	0 863	15 88	82 16	0 03458	0.16860	85.83	0.17582	89 56	0.18285
40	51 68	0 0116	0 792	17 00	82 71	0 03680	0 16833	86.41	0 17554	90.16	0 18256
45	56.38	0.0117	0 730	18 14	83.26	0 03904	0.16808	86.96	0.17528	90.76	0.18227
50	61 39	0 0118	0 673	19 27	83 78	0 04126	0.16785	87.54	0 17505	91.38	0.18203
55	66 74	0 0119	0 622	20.41	84.31	0.04348	0 16763	88.09	0.17482	91.93	0.18181
60	72 41	0 0119	0 575	21.57	84 82	0 04568	0 16741	88.64	0.17458	92 51	0.18155
65	78 44	0 0120	0 532	22.72	85.32	0.04789	0 16721	89 18	0.17436	93 11	0.18132
70	84.82	0 0121	0 493	23.90	85 82	0.05009	0.16701	89.72	0.17417	93.66	0.18114
75	91 60	0.0122	0 458	25.08	86.32	0.05229	0 16681	90.25	0 17397	94.23	0.18092
80	98 76	0.0123	0.425	26.28	86.80	0.05446	0.16662	90.78	0.17379	94.80	0.18075
85	106 40	0.0124	0 395	27.48	87.28	0 05665	0 16644	91.27	0.17361	95.33	0.18056
90	114 30	0 0125	0 368	28.70	87.74	0.05882	0.16624	91.77	0.17344	95.86	0.18040
95	122 80	0 0126	0.343	29 93	88 19	0 06100	0.16604	92.27	0.17323	96.39	0.18020
100	131 60	0.0127	0 319	31.16	88.62	0 06316	0 16584	92.75	0.17308	96.92	0.18004
105	140.90	0 0128	0.298	32.40	89 03	0.06534	0 16564	93.24	0.17291	97.46	0.17991
110	150.70	0 0129	0 277	33 65	89 43	0 06749	0 16542	93.66	0.17274	97.93	0.17976
115	161 00	0.0130	0.258	34.90	89 80	0.06965	0.16520	94.07	0.17253	98.40	0.17955
120	171.80	0 0132	0.240	36.16	90.15	0 07180	0 16495	94.47	0.17233	98.84	0.17939

$\kappa = \dfrac{C_p}{C_v} = 1\ 138$ at 25 C

Table 4 Properties of Water

Sat Temp F	Abs Press. Lb per Sq In	Volume Liquid	Volume Vapor	Heat Content and Entropy Taken From 32 F: Heat Content Liquid	Heat Content Vapor	Entropy Liquid	Entropy Vapor	50 F Superheat Ht. Ct.	50 F Superheat Entropy	100 F Superheat Ht. Ct.	100 F Superheat Entropy
32	0.0887	0 01602	3296.0	0 00	1073.0	0.0000	2.1826	1096.9	2.2277	1120.8	2.2688
35	0.1000	0 01602	2941 0	3.02	1074.4	0.0062	2 1724	1098.3	2.2172	1122.2	2.2581
40	0 1217	0.01602	2441 0	8.05	1076.8	0.0163	2.1555	1100.6	2 2000	1124.5	2.2406
45	0.1475	0.01602	2034.0	13.07	1079 2	0 0262	2.1390	1102.9	2.1832	1126.7	2.2234
50	0.1780	0 01602	1702.0	18.08	1081.5	0.0361	2 1230	1105.2	2.1667	1129.0	2.2066
55	0.2140	0.01603	1430.0	23.08	1083 9	0.0459	2.1073	1107.5	2.1506	1131.3	2.1902
60	0.2561	0.01603	1206.0	28.08	1086.2	0.0556	2.0920	1109.8	2 1349	1133.5	2.1742
65	0.3054	0.01604	1021.0	33.08	1088.6	0.0652	2.0771	1112.2	2.1196	1135.8	2.1585
70	0.3628	0.01605	868.0	38.07	1090.9	0.0746	2.0625	1114.5	2.1046	1138.1	2.1432
75	0.4295	0.01606	740.0	43 06	1093.2	0 0840	2.0483	1116.7	2.0900	1140.3	2.1283
80	0.507	0.01607	632.9	48.05	1095.5	0.0933	2.0344	1119.0	2.0758	1142.5	2.1138
85	0.596	0.01609	543.3	53.04	1097.8	0.1025	2 0208	1121.2	2.0619	1144.7	2.0996
90	0.698	0.01610	467.9	58.03	1100.0	0.1116	2.0075	1123.4	2.0483	1146.8	2.0857
95	0.815	0.01612	404.2	63.01	1102.3	0.1206	1 9946	1125.6	2.0350	1148.9	2.0721
100	0 949	0.01613	350.3	68.00	1104.6	0.1296	1.9819	1127.9	2.0220	1151.1	2.0588
105	1 101	0.01615	304.4	72.98	1106 8	0.1384	1.9695	1130.2	2.0093	1153.2	2.0458

For properties of steam at high temperatures, see Page 28.

$\kappa = \dfrac{C_p}{C_v} = 1\ 33$

theoretical cycle starts with saturated vapor, operation is common at a condition of superheated vapor (as at a_1). Moreover, expansion may start either with a mixture of liquid and vapor or with a sub-cooled liquid, as at c_1, with expansion to e_1. It is obvious that this latter is desirable as it increases the refrigerating effect. Area $a_1b_1cdaa_1$ represents the work of such a superheated cycle, while the area $e_1a_1g_1f_1e_1$ represents the refrigerating effect of the cycle with superheated vapor and sub-cooled refrigerant liquid.

It will be noted on the pressure-volume diagram the volume of the saturated liquid is indicated by a dotted line close to and parallel to the ordinate.

In the discussions in this chapter a slight error is introduced by not including all of the work of pumping the liquid from the low to the high pressure. This occurs because the liquid line is not a line of equal pressure but of saturation pressures. The error in work per pound of refrigerant

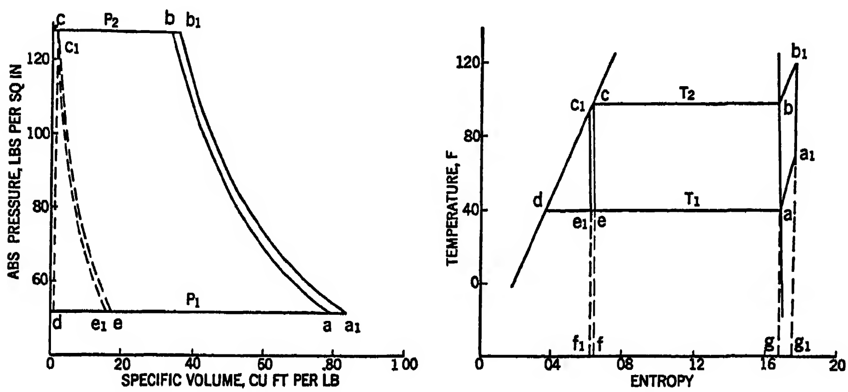


FIG. 2. THEORETICAL DICHLORODIFLUOROMETHANE (F_{12}) CYCLES

figured from total heats, which should be added to the indicated figures is roughly the specific volume of the liquid at the lower pressure and temperature multiplied by the pressure difference in appropriate units. This error may become of some importance in calculations involving carbon dioxide or in problems involving the liquid of any of the refrigerants, as in figuring expansion valve orifices.

Theoretical Work per Pound

The temperature-entropy and pressure-volume diagrams are based on one pound of the refrigerant. Likewise, the theoretical work and the refrigerating effects are conveniently based on a pound of refrigerant. The compression work per pound may be found by several methods.

The temperature-entropy method starts with state point a. Since the quality of a is known, the heat content of the vapor H_a is known, and also the entropy S_a. Since $S_a = S_b$ and with T_2 given, H_b can be determined. If W = work in foot-pounds per pound of refrigerant, then

$$W = (H_b - H_a) \times 778 \tag{1}$$

The pressure-volume method starts with state point a, whose pressure and specific volume are known. The work of compression is the adiabatic work of compression from P_1 to P_2, plus the work of expelling the vapor at constant pressure P_2 minus the external work of evaporation of the vapor to volume V_1 at pressure P_1.

$$W = \frac{n}{n-1} \times P_1 V_1 \left[\left(\frac{P_2}{P_1} \right)^{\frac{n-1}{n}} - 1 \right] \tag{2}$$

It is frequently helpful to think of the compression of the vapor in terms of head. The head may be likened to a vertical column of vapor in which is located the vapor to be compressed. The compression occurs when the vapor is moved down from a level corresponding to P_1 to a new level corresponding to P_2, in equilibrium with the surrounding vapor. If this process is carried on isentropically, the result will be the same as indicated previously. Then if h is the head in feet,

$$W = h \tag{3}$$

This relationship may easily be seen from the fact that a small difference of head dh divided by the specific volume of the vapor V is equal to the increment of pressure difference dP.

Table 5. Properties of Methyl Chloride

Sat Temp F	Abs Press Lb per Sq In	Volume		Heat Content and Entropy Taken From 40 F							
				Heat Content		Entropy		100 F Superheat		200 F Superheat	
		Liquid	Vapor	Liquid	Vapor	Liquid	Vapor	Ht Ct.	Entropy	Ht. Ct.	Entropy
0	18.73	0 0162	5.0520	14.4	192.4	0 0328	0.4197	215.6	0.467	237.2	0.507
5	20.91	0 0163	4 5630	16.2	194.1	0.0368	0.4195	217.0	0.464	238.5	0.503
10	23.30	0 0164	4.1290	18.1	195.8	0 0407	0 4192	218.5	0.463	240.0	0.500
15	25.89	0 0165	3 7430	19.9	196 9	0 0446	0 4177	219.8	0.461	241.0	0.498
20	28.71	0.0166	3.4030	21.8	198.1	0.0486	0.4152	221.0	0.460	242.5	0.496
25	31 77	0 0167	3 0980	23.6	199 1	0.0525	0.4137	222.1	0.459	244.0	0.494
30	35 07	0 0168	2.8260	25.5	200.1	0 0562	0.4130	223.4	0.457	245.2	0.493
35	38 63	0 0169	2 5830	27.3	201.1	0.0600	0.4110	224.5	0.454	246.5	0.492
40	42 47	0 0170	2 3650	29.2	202.2	0.0637	0.4099	225.7	0.453	247.7	0.490
45	46.50	0 0171	2 1700	31.0	203 1	0.0675	0.4086	226.8	0.451	249.1	0.488
50	51 03	0 0172	1 9920	32 9	204.1	0.0709	0.4069	227.9	0.449	250.5	0.487
55	55 80	0 0173	1 8320	34.8	204.9	0 0746	0.4052	228.8	0.448	251.7	0.486
60	60 88	0 0174	1.6880	36 7	205.7	0.0784	0.4037	229.6	0.446	253.0	0.484
65	66.32	0 0175	1.5580	38.5	206 4	0 0821	0.4022	230.5	0.443	254.3	0.483
70	72.11	0.0176	1 4390	40.4	207.2	0.0855	0.4004	231.3	0.441	255.5	0.481
75	78 29	0 0177	1.3310	42.2	207 8	0.0891	0 3989	232.2	0.439	256.8	0.479
80	84.86	0 0178	1 2330	44.1	208.5	0.0926	0.3973	233.0	0.437	257.9	0.478
85	91.86	0 0179	1.1440	46.0	209.2	0.0962	0 3959	233.8	0.435	259.2	0.477
90	99 26	0 0180	1 0620	47.8	209 7	0.0994	0.3941	234.5	0 433	260.4	0.476
95	107.10	0 0181	0 9877	49.7	210.3	0.1029	0.3926	235.2	0.432	261.5	0.475
100	115.40	0.0182	0 9193	51.6	210.9	0.1063	0.3910	236.0	0.431	262.4	0.474
105	124.20	0.0183	0.8567	53.5	211.4	0.1097	0.3894	237.0	0.430	263.3	0.473
110	133.40	0 0185	0.7990	55 3	211.8	0.1130	0.3877	237.9	0.428	264.3	0.472
115	143 20	0 0186	0.7461	57 2	212.3	0 1164	0.3861	238.5	0.427	265.0	0.470
120	153 50	0.0187	0.6972	59.1	212.8	0.1198	0.3845	239.0	0.425	265.6	0.468

$n = \frac{C_p}{C_v} = 1\ 20$

TABLE 6. PROPERTIES OF MONOFLUOROTRICHLOROMETHANE (F_{11})

SAT TEMP F	ABS PRESS LB PER SQ IN	VOLUME		HEAT CONTENT AND ENTROPY TAKEN FROM −40 F							
				Heat Content		Entropy		25 F Superheat		50 F Superheat	
		Liquid	Vapor	Liquid	Vapor	Liquid	Vapor	Ht Ct	Entropy	Ht Ct	Entropy
0	2 59	0 01020	13 700	7 81	90 4	0 0178	0 1975	93 9	0.2049	97.4	0.2120
5	2.96	0.01024	12.100	8.81	91 2	0 0200	0 1974	94.7	0.2047	98 2	0 2117
10	3.38	0 01028	10.700	9.82	92.0	0.0222	0.1973	95.5	0.2045	99.0	0.2114
15	3 85	0.01032	9.530	10 80	92 8	0 0243	0.1971	96.3	0.2043	99 8	0.2111
20	4.36	0.01036	8.490	11.90	93.7	0.0264	0 1970	97.2	0.2041	100.7	0.2109
25	4.94	0.01040	7.580	12 90	94.5	0.0286	0.1969	98.0	0.2039	101 5	0.2107
30	5.57	0.01045	6.770	13 90	95 3	0 0307	0.1969	98 8	0.2038	102.3	0.2105
35	6.27	0 01049	6 080	14.90	96 1	0.0328	0 1968	99 6	0.2037	103.1	0.2103
40	7 03	0 01053	5.460	16.00	96.8	0.0349	0.1968	100 3	0.2036	103 8	0.2101
45	7.88	0.01057	4.920	17.00	97.6	0.0370	0.1967	101 1	0 2035	104 6	0.2099
50	8 79	0.01062	4.440	18 10	98.4	0 0391	0.1967	101.9	0.2034	105.4	0.2098
55	9 80	0 01066	4 020	19.10	99.2	0.0412	0.1967	102 7	0 2033	106.2	0.2097
60	10 90	0.01071	3.640	20.20	100.0	0.0432	0.1967	103.5	0.2033	107.0	0.2096
65	12.10	0.01076	3.300	21 30	100 8	0 0453	0.1967	104.3	0.2032	107.8	0 2094
70	13.40	0 01081	3.000	22.40	101 5	0.0473	0.1967	105.0	0 2032	108.5	0.2093
75	14.80	0.01086	2.740	23.50	102.2	0.0493	0.1967	105.7	0 2031	109.2	0.2092
80	16.30	0.01091	2.500	24 50	102.9	0.0513	0.1966	106.4	0 2030	109.9	0.2090
85	17.90	0.01096	2.280	25.60	103.6	0.0533	0.1966	107.1	0.2029	110.6	0.2089
90	19.70	0.01101	2 090	26 70	104.4	0.0553	0.1966	107.9	0.2028	111.4	0.2088
95	21.60	0 01106	1.918	27.80	105.1	0.0573	0.1966	108.6	0.2028	112.1	0.2087
100	23 60	0 01111	1.761	28.90	105.7	0.0593	0.1965	109.2	0.2027	112.7	0.2085
105	25.90	0.01116	1.620	30 10	106.4	0.0613	0.1965	109 9	0.2026	113.4	0.2084

$n = \frac{C_p}{C_v} = 1\ 135$

Head is very useful in considering the performance of centrifugal compressors, which merely substitute a centrifugal for the gravity head. It is also useful in considering problems of fluid flow. In these problems, the head per degree can be obtained either by direct calculation or approximately by dividing the total head by the temperature difference $T_2 - T_1$. The velocity head loss can then be calculated in degrees, using the customary formula $V^2 = 2gh$.

Refrigerating Effect per Pound

The refrigerating effect per pound is computed by the same method, regardless of the type of refrigeration system. The solution is indicated on the temperature-entropy diagram of Fig. 2. Assuming that the vapor leaving the evaporator is saturated, the refrigerating effect in Btu per pound is obtained by subtracting from the heat content of the vapor at temperature T_1, the heat content of the liquid at T_2, or if the liquid is sub-cooled, the liquid temperature.

Thus, the refrigerating effect in Btu per pound is equal to

$$H_a - H_c = H_a - H_e \tag{4}$$

If the vapor entering the compressor is superheated or supersaturated, a correction in the heat of the vapor is made accordingly.

The unit of refrigeration is the *ton*, based on the latent heat of fusion of one ton of ice in 24 hours.

Thus one ton = 200 Btu per minute = 12,000 Btu per hour.

Coefficient of Performance

The coefficient of performance of a refrigeration system is the ratio of the refrigerating effect to the work of compression, both expressed in the same units.

The ideal or Carnot coefficient of performance depends upon the temperatures T_1 and T_2 in much the same way as the ideal efficiency of a steam engine depends upon its working temperature, with an inverse relationship.

$$\text{Ideal C. of P.} = \frac{T_1}{T_2 - T_1} \tag{5}$$

Evidently the smaller the compression range, the less power will be required to produce a given refrigerating effect.

Table 7. Theoretical Comparison of Various Refrigerants[a]

Refrigerant	Work per Lb Ft Lb	Equivalent Head Ft	Gas Temp Leaving Comp. F	Refrig Effect per Lb	Lb per Min per Ton	Coeff of Perfor	Cycle Eff per Cent
Ammonia	53,900	53,900	209	489.8	0.408	7.06	84.8
Carbon Dioxide[b]	9,940	9,940	146	64.0	3.125	5.00	46.9
Dichlorodifluoromethane (F_{12})	6,280	6,280	134	55.74	3 638	6.90	82.9
Methyl Chloride	16,500	16,500	183	156.9	1 275	7.38	88.8
Monofluorotrichloromethane (F_{11})	7,050	7,050	101	74.0	2.605	7.53	90.4
Water[c]	114,300	114,300	360	1010.8	0 181	6.85	82.3
Ideal (Carnot)						8 33	100.0

[a]Evap Refrig. Temp = 40 F Suction Vapor Temp. = 65 F.
Cond Refrig. Temp = 100 F. Liquid Temp = 98 F.

[b]Based on 1057 lb per sq in (87 F) Condenser Pressure, and 85 F Liquid Temperature.

[c]Based on 40 F Temperature

The theoretical coefficient of performance of actual refrigerants is always less than the ideal due to the tendency of most refrigerants to superheat when compressed, and due to the heat of the liquid which must be removed. The cycle efficiency is the theoretical C. of P. divided by the ideal for the same temperatures. The cycle efficiency usually changes as the compression temperatures change.

Comparative results of modified theoretical cycles of the refrigerants, are given in Table 7.

Practical Cycle

Fig. 3 illustrates the pressure-volume and temperature-entropy diagrams for an actual cycle. These diagrams are based upon the compressor receiving vapor superheated and upon sub-cooling of the liquid going to the evaporator. The theoretical cycle is $a_1b_1cc_1e_1a_1$. However, the vapor during compression actually follows line a_1b_2 due to superheating as a result of the inefficient work of compression. The theoretical work of

compression is $a_1b_1cda_1$. Added to this is the area $b_2b_1g_1h_1b_2$ on the temperature-entropy diagram which represents the inefficient work of compression (assuming no compressor heat losses). The sum of these areas represents the total work of the compressor per pound of refrigerant, and the ratio of theoretical cycle work to the actual work represents the overall efficiency. It should be noted that area $a_1b_2b_1a_1$ is considered as part of the inefficient work and is commonly termed the superheat loss. The refrigerating effect per pound is the same for the practical as for the theoretical cycle, working with the same sub-cooling of liquid and superheating of vapor, that is, area $e_1a_1g_1f_1e_1$.

Sources of loss which are usually recognized as reflected by the overall

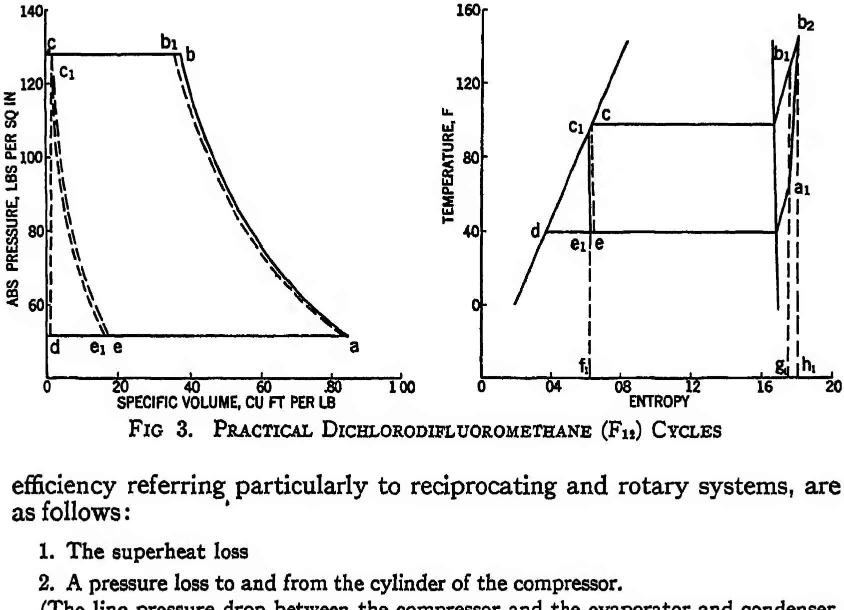


Fig 3. Practical Dichlorodifluoromethane (F_{12}) Cycles

efficiency referring particularly to reciprocating and rotary systems, are as follows:

1. The superheat loss
2. A pressure loss to and from the cylinder of the compressor.

(The line pressure drop between the compressor and the evaporator and condenser, respectively, is usually taken into account separately in the design of the refrigeration system)

3. Leakage loss through valves and past pistons is quite small in most compressors.
4. With an oil soluble refrigerant, there may be an absorption loss due to absorption and re-evaporation of refrigerant in the oil of the cylinder.
5. Mechanical losses are always present and are usually a large part of the total

Reciprocating and rotary compressors always take in less vapor than that which corresponds to the displacement. The overall volumetric efficiency is the ratio of the suction vapor volume to the piston displacement. Part of this is the re-expansion volumetric efficiency which is the volume, at suction pressure, of the usefully re-expanded vapor which was in the clearance volume. This is expressed by the following equation:

$$\text{Volumetric Efficiency} = 1 - \frac{v_c}{v_d} \times \left[\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}} - 1\right] \tag{6}$$

where

v_c = clearance volume.
v_d = cylinder displacement volume.

The balance of the overall volumetric efficiency is known as the superheat volumetric efficiency even though it includes some other sources of capacity loss.

The mechanical efficiency of a reciprocating and rotary compressor must be multiplied by the superheat volumetric efficiency to give the overall efficiency of the compressor.

$$\text{Eff.}_{\text{overall}} = \frac{\text{Vol Eff}_{\text{overall}}}{\text{Vol. Eff.}_{\text{reexp.}}} \times \text{Mech. Eff.} = \text{Super Vol Eff.} \times \text{Mech. Eff.} \qquad (7)$$

Normally, the volumetric efficiency of a compressor varies with the

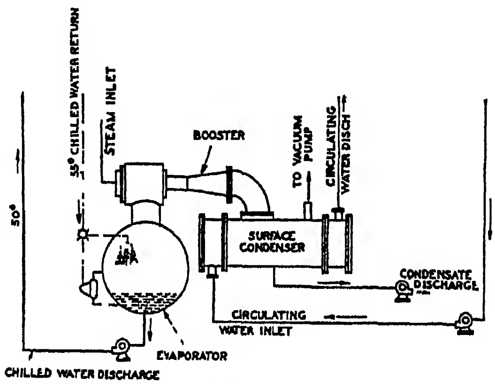


Fig. 4. Steam Ejector Compression Refrigeration System

ratio of compression, while the mechanical efficiency remains virtually fixed. Good standard practices for F_{12} compressors are:

Low comp. ratio = 2.5 to 1		High comp. ratio = 5 to 1
Vol eff.$_{\text{reexp.}}$	94 to 96 per cent	88 to 92 per cent
Vol. eff.$_{\text{super.}}$	75 to 85 per cent	73 to 77 per cent
Vol. eff.$_{\text{overall}}$	70 to 81 per cent	64 to 71 per cent
Mech. eff.	75 to 85 per cent	75 to 85 per cent

These values are for one ton or larger compressors. Part of the difference expresses the change with capacity. With other refrigerants and other types of compressors there will be some further variation.

Steam Ejector Refrigeration System

Fig. 4 is a diagrammatic representation of a typical steam ejector refrigeration system. Live saturated steam is supplied to a nozzle which discharges at a high vacuum into the steam ejector. The expansion of the steam imparts a tremendous velocity to it, which entrains vapor from a flash evaporator and compresses it with a conversion from velocity to static pressure. The mixture of vapor is then delivered to a condenser. The condensate is removed by suitable means, some of it being used in some cases to supply make-up water to the evaporator circuit. The con-

denser is maintained free of air by means of a suitable purge. This may take the form of a water ejector if only a small amount of air is to be removed, as in a completely closed system. If a larger capacity is required, a two-stage purge is usually used, sometimes a combination steam and water and occasionally a two-stage steam purge. The vapor which is drawn from the evaporator is the result of spraying water which has been warmed by the cooling load into the flash evaporator. Evaporation of a part of this water occurs, which cools the remainder, which is then again returned to the cooling load to receive more heat.

The performance of the steam ejector may be studied theoretically by the use of the temperature-entropy diagram, Fig. 5. Unlike its usual application, however, the amount of working fluid is different for one portion of the cycle than for the other. Dry saturated steam under high

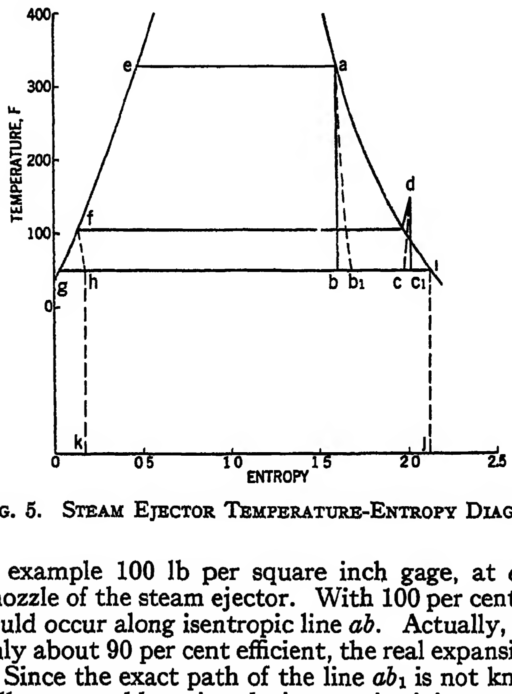


Fig. 5. Steam Ejector Temperature-Entropy Diagram

pressure, for example 100 lb per square inch gage, at a, is expanded through the nozzle of the steam ejector. With 100 per cent efficiency, the expansion would occur along isentropic line ab. Actually, however, most nozzles are only about 90 per cent efficient, the real expansion being along the line ab_1. Since the exact path of the line ab_1 is not known, the work area is normally assumed by using the isentropic giving a work area $abgea$. The velocity at the mouth of the nozzle may be determined in the usual manner using this area and the velocity coefficient of the nozzle

At the evaporator pressure or slightly below, the vapor from the nozzle mixes with virtually dry saturated vapor from the evaporator. An impact loss also occurs at this point due to the mixture of vapors at different velocities. This results in bringing the state point of the mixture to c. Compression then occurs along the line cd, the work of compression per pound being $cdfgc$. In computing the work area, however, the point c is not actually known. Therefore, the work area c_1dfgc_1 is used in expressing the efficiency of the ejector, the line c_1d being an isentropic. The

losses are expressed by nozzle efficiency, impact loss and diffuser efficiency. The work of compression, however, is performed on the mass of the mixture. Thus, the available work is reduced in proportion to:

$$\frac{M_{\text{primary}}}{M_{\text{mixture}}}$$

The impact loss is commonly determined from the formula:

$$MV_{\text{primary}} + MV_{\text{secondary}} = MV_{\text{mixture}} \tag{8}$$

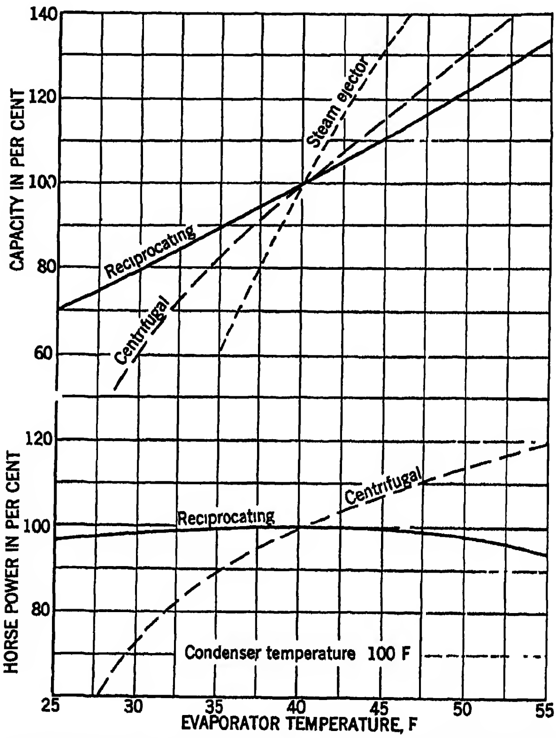


Fig 6 Performance Characteristics of Compression Refrigeration Machines at Constant Speed

Common efficiencies for commercial ejectors are: nozzle efficiency 90 per cent, diffuser efficiency 60 to 70 per cent. Customary steam rates in pounds per ton are approximately as follows:

Evaporator temp.	50 F	Steam press	100 lb	Steam press	12 lb
Condenser temp	105 F	Steam rate	30 lb per hour per ton	Steam rate	45 lb per hour per ton
Evaporator temp	40 F	Steam press.	100 lb	Steam press	12 lb
Condenser temp	105 F	Steam rate	40 lb per hour per ton	Steam rate	70 lb per hour per ton

The steam ejector uses large amounts of condenser water and steam, and therefore this system has relative advantages when large quantities of water and steam are available at reduced costs. Characteristic curves

show that the ejector performance is independent of condenser temperature up to a certain temperature, depending upon the particular design, and above this temperature the ejector breaks and loses all capacity. (See also Chapter 11.)

Characteristics of Compression Systems

The different types of compression systems have quite different characteristics of capacity and power with varying evaporator temperature and with varying condenser temperature, as will be seen from curves in Figs. 6 and 7.

The capacity of the reciprocating and rotary compressor varies slowly

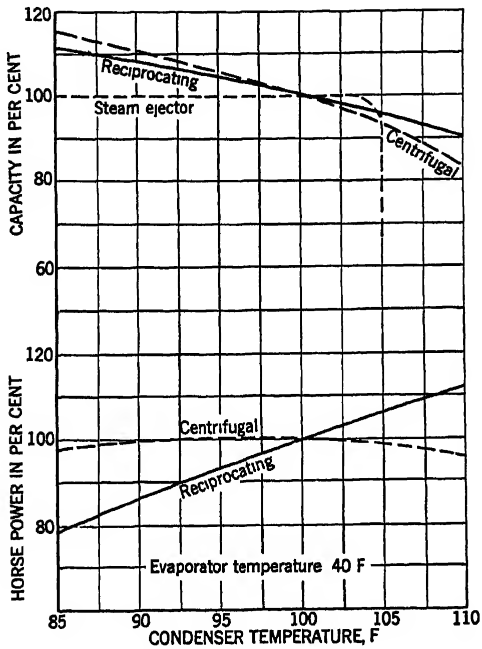


FIG. 7. PERFORMANCE CHARACTERISTICS OF COMPRESSION REFRIGERATION MACHINES AT CONSTANT SPEED

with a change of evaporator temperature, and the variance of power requirements, in the air conditioning range of operation, is small for a change of evaporator temperature. On the other hand, the capacity and power of the centrifugal machine vary rapidly, and the capacity of the steam ejector also varies considerably. Thus, both these latter types tend to be more nearly self-regulating than the reciprocating and rotary compression type. On the other hand, the operating range of the latter near standard capacity is superior. Although the capacity of the reciprocating and rotary compressor is little affected by the condenser temperature, the power of the compressor is greatly affected, while the reverse is true for the centrifugal compressor. As previously indicated, the condenser temperature has no effect on the capacity of the steam ejector type of

compressor until a certain point is reached, beyond which the capacity is zero. The steam consumption for the performance characteristic curves shown in Figs. 6 and 7 remains constant for all evaporator and condenser temperatures.

CLOSED ABSORPTION SYSTEMS

Years ago the closed absorption system was in favor for industrial refrigeration and ice making. It is now of little commercial importance, however, except for very special applications (notably the gas-fired refrigerator). The most usual refrigerant has been ammonia, and in some cases water has been used as an absorbent. However, a long list of refrigerant-absorbent combinations have been proposed and quite a number have been tested, either experimentally or commercially. There are some commercial installations using a dry adsorber instead of a liquid absorber.

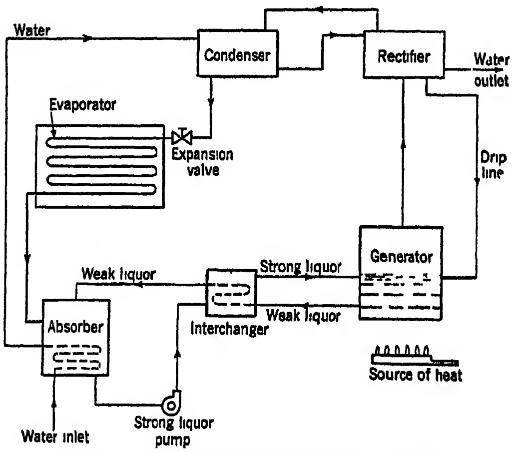


FIG. 8. CLOSED ABSORPTION SYSTEM

Fig. 8 shows a typical diagram of a closed absorption system. A mixture of refrigerant and absorbent is evaporated in the generator, passes to an analyzer and rectifier where it is purified, and then to a condenser where the refrigerant and remaining absorbent is condensed. It then passes through an expansion valve to an evaporator, where heat is absorbed from a cooling load. From the evaporator the vapor and residual absorbent passes to an absorber; where it meets absorbent which is initially low (weak) in refrigerant concentration. The absorbent absorbs the vapor, and the strong absorbent liquor is transferred to the generator through an interchanger with the weak liquor returning from the generator.

Cooling water is ordinarily used in the absorber to remove the heat of absorption and maintain the absorptive power of the absorber at a maximum.

Like the steam ejector system, the absorption system compares most favorably when a cheap source of cooling water and steam or other heat

source is available. Unlike the ejector system, the comparative performance is usually best with a wide range of temperature between the evaporator and absorber, since with a good refrigerant-absorbent combination, the amount of heat and water required for a given refrigerating effect increases slowly with an increase of evaporator-condenser temperature range.

There are innumerable variations on the arrangement shown in Fig. 8 which include multiple absorption systems, parallel absorption systems, systems using inert gas to raise the pressure in the evaporator and absorber to that of the condenser.

OPEN ADSORPTION SYSTEMS

For air conditioning installations, some experimental work and some commercial work has been done, using open adsorption refrigeration systems. The open liquid adsorption system is illustrated diagrammatically by Fig. 9.

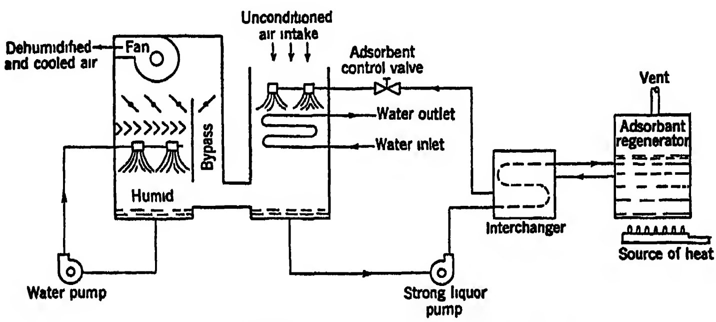


FIG. 9. OPEN LIQUID ADSORPTION SYSTEM

Air from the outside and from the conditioned space is drawn through an adsorbent spray which, with the air, is cooled by passing over water cooled surfaces. The adsorbent material adsorbs much of the water vapor from the air. The cooling coil removes the heat of adsorption and may cool the air below entering temperature, if the water is cold. The dehumidified and partially cooled air is then passed to a humidifier where water-vapor is added with further cooling by evaporation. A bypass may be provided so that some of the dry air bypasses the humidifier in order to prevent the dew point of the mixture from rising too high. The air then passes through a fan and is delivered to the conditioned space. The adsorbent, rich in water vapor, is discharged through an interchanger to a regenerator, from which it again flows to the adsorber.

The adsorbent materials most commonly considered are solids which are put into solution as brines (see also Chapter 11). The difficulty with most adsorbents is that their adsorptive power is not great enough to dehumidify the air sufficiently. Thus, they have to be used inefficiently or do an inadequate job of dehumidification.

Fig. 10 is a diagrammatic representation of an open solid material adsorption system. The spray and cooling coil of the open adsorber is

replaced by a bed of adsorptive material and a separate water cooling coil, and a second section of adsorber must be provided for alternate reactivation while the first is adsorbing. A set of dampers or a rotary device must be arranged to connect the beds alternately to the air to be conditioned and to the hot gases for regeneration. Otherwise the liquid adsorption and solid adsorption systems are identical.

The most efficient solid adsorption systems use some internal means of heating the adsorber bed before activation, and cooling it after activation, and other means may also be employed for improving the performance.

These systems are especially affected by the temperature of the cooling water, since if the air leaving the cooling surface is at a higher temperature than the air entering, additional work must be done in the generator or activator in order to compensate for the extra sensible heat of the leaving air. On the other hand, if the leaving temperature is lower than the entering temperature, the reverse is true. When the cooling water

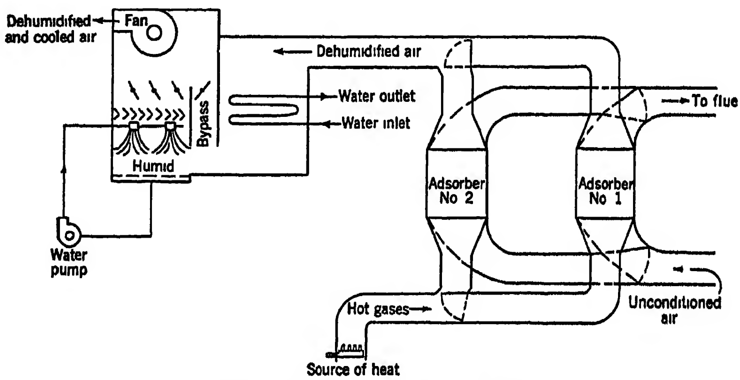


Fig. 10. Open Solid Material Adsorption System

temperature is low, similar economies may, however, be obtained with compression systems by the use of an air pre-cooling coil.

From this discussion it may be seen that these systems function to best advantage on a high air temperature, and on one in which the initial moisture content is high. Thus the use of high room temperatures and comparatively large amounts of outside air are encouraged in connection with these systems. In order to offset the effect of high air temperature, some effort is made to keep the humidity lower than usual. The ratio of the heat input to the refrigerant output, where the temperature of the air entering and leaving the adsorber cooler is not widely different, may vary from 4 to 1 for some of the solid adsorption systems, to as low as 1.5 to 1 for some of the liquid adsorption systems.

THE REVERSE CYCLE

The idea of heating by the reverse refrigeration cycle has captured the imagination of many people and has been much discussed. In principle, heat is absorbed in an evaporator from some available source of heat, pumped to a higher temperature and delivered to a condenser. The heat

from the condenser is used for heating purposes. The compressor acts as a heat pump whose fundamental function is to raise the potential of the heat. The theoretical work of compression in relation to the heat delivered is·

$$\frac{T_2}{T_1 - T_2}$$

where

T_1 = absolute temperature of evaporator.

T_2 = absolute temperature of condenser.

Thus, with a small spread of temperature between the evaporator and the condenser, 6 or 8 times as much heat may be obtained theoretically and 4 or 5 times practically, as the work put in. There are a number of limitations, however, the most serious of which is the lack of ready availability of a practical source of heat.

1 Well water is the most desirable since its temperature is high even in the winter and thus a large amount of heat may be removed in relation to the weight of water handled.

2 Air may be used but its specific heat is low and its temperature uncertain. When the most heat is needed, the temperature of the air is lowest, thus resulting in the least favorable temperature combination.

3 It has been proposed to obtain heat by freezing water, but this is still in the theoretical stage

Some of the other factors which act as limitations are the large temperature spread when using air as a source of heat and when attempting to cool with even moderately low outside temperatures, the frequent disparity between the size of the cooling load and heating load requiring extra equipment for a complete heating load, and the relatively high initial cost of equipment at present available for the reverse cycle in comparison with that available for heating by conventional means.

Because of these limitations, the present application of the system is largely limited to temperate climates, such as Florida and Southern California, or to heating only for intermediate seasons, or to other localities which have peculiar advantages as, for instance, the ready availability of well water. In these locations it is frequently possible to do all of the heating necessary with the refrigeration equipment so that the extra cost is only that of reversing the functions of the condenser and evaporator.

REFERENCES

Handbook of Mechanical Refrigeration, by H. J. Macintire.

Official Refrigeration Service Manual, by L. K. Wright.

Power Plant Testing, by J. A. Moyer.

Refrigerating Data Book 1937-38, *American Society of Refrigerating Engineers.*

Refrigeration, by J. A. Moyer and R. U. Fittz.

Thermodynamics for Engineers, by J. A. Ewing.

PROBLEMS IN PRACTICE

1 ● Dichlorodifluoromethane (F_{12}) at a saturated temperature of 30 F but superheated 25 F to 55 F is compressed to a saturation temperature of 90 F with a compressor having an overall efficiency of 70 per cent. It leaves the condenser sub-cooled 5 F. What is the: *a.* Work per pound; *b.* Refrigerating effect; *c.* Pounds per minute per ton; *d.* Horsepower per ton; *e.* Heat rejected to condenser neglecting radiation; *f.* Equivalent discharge temperature?

a. $\frac{1.151}{0.151} \times 43.16 \times 144 \times 0.939 \times \frac{515}{490}\left[\left(\frac{114.3}{43\ 16}\right)^{\frac{0.151}{1\ 151}} - 1\right] \times \frac{1}{0.70} = 9000$ ft-lb

(*By Pressure—Volume Method*)

Check· $(93.35 - 85\ 25) \times \frac{1}{0\ 7} = 11\ 6$ Btu per pound $= 9000$ ft-lb (*By Heat Content Method*)

b. $95\ 25 - 27\ 48 = 57.77$ Btu per pound.

c. $\frac{200}{57.77} = 3\ 46$ lb per minute per ton

d. $9000 \times \frac{3.46}{33{,}000} = 0\ 944$ hp per ton

e. $200 + 0\ 944 \times 42\ 5 = 240$ Btu per minute per ton.

Check: Heat content leaving compressor $= 85\ 25 + 11\ 6 = 96.85$
$(96.85 - 27.48)\ 3.46 = 240$ Btu per minute per ton.

f. Heat Content $= 96.85$.
Pressure $= 114.3$ lb per square inch
From Table 3, Temp. $= 146$ F.

2 ● If the velocity of vapor in the suction pipe is 50 fps and there are 10 velocity heads lost between evaporator and compressor, what is the saturation temperature at the evaporator?

$$h = \frac{V_2^*}{2g}$$

$$h = 10 \times \left(\frac{50}{2g}\right)^2 = 388 \text{ ft head.}$$

$$\text{Head per degree} \doteq (47\ 28 - 43\ 16) \times \left(\frac{0.863 + 0\ 792}{2}\right) \times \frac{144}{5} \doteq 98.4 \text{ ft.}$$

Check: Head per degree $= 9000 \times \frac{0\ 7}{(90 - 30)} = 105$ ft. (*Approx.*)

$$\text{Temperature} = 30 + \frac{388}{98.4} = 34 \text{ F.}$$

3 ● If the refrigerant is sub-cooled to 70 F, what is the effect on: *a.* Work per pound: *b.* Refrigerating effect; *c.* Horsepower per ton?

a. No effect on work per pound.

b. $(85\ 25 - 23{,}90) = 61.35$.
$61.35 - 57.77 = 3.58$ Btu increase

$\frac{3.58}{57.77} = 6.2$ per cent increase.

c $\frac{200}{61\ 35} = 3\ 27$ lb per minute per ton

$9000 \times \frac{3.27}{33{,}000} = 0\ 891$ hp per ton.

$0\ 944 - 0\ 891 = 0\ 053$ hp decrease

$\frac{0\ 053}{0\ 944} = 5\ 6$ per cent decrease

4 ● What is the approximate change in capacity of the following types of systems per degree at 40 F: *a.* Reciprocating; *b.* Centrifugal; *c.* Ejector?

a 2 5 per cent *b* 3 0 per cent. *c* 7 5 per cent.

5 ● *a.* Which type of system will maintain the most uniform evaporator temperature with change of load?

**   *b.* Which system will maintain the most uniform load with change of evaporator temperature?**

From Fig. 6

a. Steam ejector

b. Reciprocating and centrifugal.

6 ● *a.* What is the velocity of steam expanding from 100 lb per square inch gage, saturated to 0.0178 lb per square inch absolute, corresponding to 50 F if the nozzle has an efficiency of 90 per cent?

***b.* What is the velocity of the mixture of this steam with one-third the mass of entrained steam moving at 300 fps?**

a. $V = \sqrt{778 \times 2g \times 0\ 9 \times (1189\ 0 - 810.3)}$

$V = 4130$ fps

b. $V_{\text{mix.}} = \frac{3 \times 4130 + 1 \times 300}{4} = 3172$ fps.

7 ● If air entering an open adsorption system at 80 F and 50 per cent relative humidity is dehumidified and cooled to a temperature of 90 F and 12 per cent relative humidity, how much air is cooled per ton of refrigeration and what is the latent heat of the water which is adsorbed?

Entering Conditions	*Leaving Conditions*
66.6 F WB.	59 1 F WB.
30.85 Btu Total Heat	25 59 Btu Total Heat
76 0 Grains per Pound	24.5 Grains per Pound

$$\text{cfm} = \frac{200 \times 13\ 5}{(30.85 - 25.59)} = 513 \text{ cfm per ton}$$

$$L = \frac{513 \times (760 - 24\ 5) \times 1044}{13.5 \times 7000} = 292.5 \text{ Btu.}$$

$$\textit{Check: } L = 200 + \left(\frac{513 \times 0.2415 \times 10}{13.5}\right) = 291.6 \text{ Btu.}$$

8 ● If air entering an open adsorption system at 80 F and 50 per cent relative humidity is cooled to 75 F and 12 per cent relative humidity, how much air is required per ton and what is the latent heat of the water which is absorbed?

Entering Conditions	*Leaving Conditions*
66 6 F WB	50.3 F WB
30 85 Btu Total Heat	20.35 Btu Total Heat
76.0 Grains per Pound	14.4 Grains per Pound

$$\text{cfm} = \frac{200 \times 13\ 5}{(30\ 85 - 20.35)} = 257 \text{ cfm per ton}$$

$$L = \frac{257 \times (76\ 0 - 14\ 4) \times 1044}{13.5 \times 7000} = 175.2 \text{ Btu.}$$

$$\textit{Check:}\ L = 200 - \left(\frac{257 \times 0.2415 \times 5}{13\ 5}\right) = 177\ 0 \text{ Btu}$$

9 ● Why is it possible for a reversed cycle refrigeration unit to show a better performance in heating operation than when operating during the cooling season?

In normal use a refrigerating machine is arranged to remove heat and the heat removed is thrown away. The driving energy is converted into heat, most of which is added to the heat removed and is also thrown away.

In the reversed refrigeration cycle the heat removed is not thrown away, but, together with the heat converted from the driving energy, it is utilized to heat the building.

10 ● In a large ice plant, for approximately 147,000 Btu equivalent input to the motor driving the ammonia compressor a ton of ice is produced, in the production of which 288,000 Btu are removed from one ton of water at 32 F to transform it to ice at 32 F. What is the ratio of the heat removed in this process to the input of the motor?

In this case the heat removed, or pumped from the water to the air is approximately 1.96 times the heat equivalent of the input to the motor.

11 ● If the cycle is reversed in Question 10 and the heat is transferred or pumped from the air to the water, what is the ratio of the heat removed in the process to the motor input?

In this case 2.96 Btu is put into the water for each Btu equivalent input to the motor, for the total heat put into the water now represents the heat pumped plus the heat converted from the driving energy, or motor input.

Chapter 3

PHYSICAL AND PHYSIOLOGICAL PRINCIPLES OF AIR CONDITIONING

Vitiation of Air, Heat Regulation in Man, Effects of Heat, Effects of Cold and Temperature Changes, Acclimatization, Effective Temperature Index of Warmth, Optimum Air Conditions, Winter and Summer Comfort Zone, Optimum Humidity, Air Quality and Quantity, Air Movement and Distribution, Natural and Mechanical Ventilation, Heat and Moisture Losses, Ultra-Violet Radiation and Ionization, Recirculation and Ozone, Ventilation Standards

VENTILATION is defined in part as "the process of supplying or removing air by natural or mechanical means to or from any space." (See Chapter 44.) The word in itself implies quantity but not necessarily quality. From the standpoint of comfort and health, however, the problem is now considered to be one of securing air of the proper quality rather than of supplying a given quantity.

The term *air conditioning* in its broadest sense implies control of any or all of the physical or chemical qualities of the air. More particularly, it includes the simultaneous control of temperature, humidity, movement, and purity of the air. The term is broad enough to embrace whatever other additional factors may be found desirable for maintaining the atmosphere of occupied spaces at a condition best suited to the physiological requirements of the human body.

VITIATION OF AIR

Under the artificial conditions of indoor life, the air undergoes certain physical and chemical changes which are brought about by the occupants themselves. The oxygen content is somewhat reduced, and the carbon dioxide slightly increased by the respiratory processes. Organic matter, which is usually perceived as odors, comes from the nose, mouth, skin and clothing. The temperature of the air is increased by the metabolic processes, and the humidity raised by the moisture emitted from the skin and lungs. There is also a marked decrease in both positive and negative ions in the air of occupied rooms but the significance of this factor is still questionable[1].

Contrary to old theories, the usual changes in oxygen and carbon dioxide are of no physiological concern because they are much too small even under the worst conditions. The amount of carbon dioxide in air is

[1]Changes in Ionic Content in Occupied Rooms Ventilated, by Natural and Mechanical Methods, by C. P Yaglou, L C. Benjamin and S P Choate (A S H V E TRANSACTIONS, Vol 37, 1931)

often used in ventilation work as an index of odors of human origin, but the information it affords rarely justifies the labor involved in making the observation[2, 3]. Little is known of the identity and physiological effects of the organic matter given off in the process of respiration. The former belief that the discomfort experienced in confined spaces was due to some toxic volatile matter in the expired air is now limited, in the light of numerous researches, to the much less dogmatic view that the presence of such a substance has not been demonstrated. The only certain fact is that expired and transpired air is odorous and offensive, and it is capable of producing loss of appetite and a disinclination for physical activity. These reasons, whether esthetic or physiological, call for the introduction of a certain minimum amount of clean outdoor air to dilute the odoriferous matter to a concentration which is not objectionable.

A certain part of the dissemination of disease in confined spaces is caused by the emission of pathogenic bacteria from infected persons Droplets sprayed into the air in talking, coughing, sneezing, etc., do not all fall immediately to the ground within a few feet from the source, as it was formerly believed. The large droplets do, of course, but minute droplets less than 0.1 mm. in diameter evaporate to dryness before they fall the height of a man. Nuclear residues from such sources, which may contain infective organisms drift long distances with the air currents and the virus may remain alive long enough to be transmitted to other persons in the same room or building. Wells[4] recovered droplet nuclei from cultures of resistant micro-organisms a week after inoculation into a tight chamber of 300 cu ft capacity. Typical organisms of infections of the upper respiratory tract (pneumococcus type I, B. diphtheriae, Streptococcus hemolyticus, and Streptococcus viridans) were found to die out quite soon when exposed to light and air, and could be recovered from the air in small numbers only 48 hours after inoculation. Organisms typical of the intestinal tract (B. coli, B. typhosus, B. paratyphosus, A. and B. dysenteriae) were not recovered 12 hours after inoculation.

The significant factors in infection are believed to be the numbers of infective organisms encountered, the frequency of exposure, and the resistance of the individual including the degree of acquired immunity. The probability of encountering a sufficient number of organisms to break down the natural body defense is related to the air space per person and the quantity of clean air supplied. Except in badly ventilated rooms, the danger is believed to be "much contracted in space, limited in time and restricted to comparatively few diseases."[5]

Practical possibilities in sterilizing air supplies by the use of ultra-violet light are now being studied[6].

The primary factors in air conditioning work, in the absence of any specific contaminating source, are temperature, radiation, drafts and

[2]Indices of Air Change and Air Distribution, by F C. Houghten and J. L. Blackshaw (A.S.H.V.E. Transactions, Vol 39, 1933).

[3]Ventilation Requirements, by C P. Yaglou, E. C Riley and D. I. Coggins (A.S.H.V.E. Journal Section, *Heating, Piping and Air Conditioning*, January, 1936).

[4]Air-Borne Infection and Sanitary Air Control, by W. F. Wells, (*Journal Industrial Hygiene*, November, 1935)

[5]Preventive Medicine and Hygiene, by Milton J Rosenau (6th edition, pp. 909-917, D. Appleton-Century Co, N. Y, 1935).

[6]Viability of B Coli Exposed to Ultra-Violet Radiation in Air, by W. F. Wells, and G. M. Fair (*Science* 1935, *82* p 280).

body odors. As compared with these physical factors, the chemical factors are, as a general rule, of secondary importance.

HEAT REGULATION IN MAN

The importance of the thermal factors arises from the profound influence which they exert upon body temperature, comfort and health. Body temperature depends on the balance between heat production and heat loss. The heat resulting from the combustion of food within the body maintains the body temperature well above that of the surrounding air. At the same time, heat is constantly lost from the body by radiation, conduction and evaporation. Since, under ordinary conditions, the body temperature is maintained at its normal level of about 98.6 F, the heat production must be balanced by the heat loss. In healthy persons this takes place automatically by the action of the heat regulating mechanism.

According to the general view, special areas in the skin are sensitive to heat and cold. Nerve courses carry the sense impressions to the brain and the response comes back over another set of nerves, the motor nerves, to the musculature and to all the active tissues in the body, including the endocrine glands. In this way, a two-sided mechanism controls the body temperature by (1) regulation of internal heat production (chemical regulation), and (2) regulation of heat loss by means of automatic variation in the rate of cutaneous circulation and the operation of the sweat glands (physical regulation). The mechanisms of adjustment are complex and little understood at the present time. Coordination of these different mechanisms seems to vary greatly with different air conditions.

With rising air temperatures up to 75 F or 80 F, metabolism, or internal heat production, decreases slightly[7], probably by an inhibitory action on heat producing organs, especially the adrenal glands, which seem to exert the major influence on basic combustion processes in the body. The blood capillaries in the skin become dilated by reflex action of the vasomotor nerves, allowing more blood to flow into the skin, and thus increase its temperature and consequently its heat loss. The increase in peripheral circulation is at the expense of the internal organs. If this method of cooling is not in itself sufficient, the stimulus is extended to the sweat glands which allow water to pass through the surface of the skin, where it is evaporated. This method of cooling is the most effective of all, as long as the humidity of the air is sufficiently low to allow for evaporation. In high humidities, where the difference between the dew-point temperature of the air and body temperature is not sufficient to allow rapid evaporation, equally good results may be obtained by increasing the air movement, and hence the heat loss by conduction and evaporation.

In cold environments, in order to keep the body warm there is an actual increase in metabolism brought about partly by voluntary muscular contractions (shivering) and partly by an involuntary reflex upon the heat producing organs. The surface blood vessels become constricted, and the blood supply to the skin is curtailed by vasomotor shifts to the internal organs in order to conserve body heat. The sweat glands become inactive.

---

[7]Heat and Moisture Losses from the Human Body and Their Relation to Air Conditioning Problems, by F C. Houghten, W W. Teague, W E Miller and W. P Yant (A S H.V.E. TRANSACTIONS, Vol. 35 1929).

EFFECTS OF HEAT

Although the human organism is capable of adapting itself to variations in environmental conditions, its ability to maintain heat equilibrium is limited. The upper limit of effective temperature to which the human organism is capable of adapting itself without serious discomfort or injury to health is 90 deg ET for men at rest and between 80 and 90 deg ET for men at work depending upon the rate of work. Within these limits a new equilibrium is established at a higher body temperature level through a chain of physiological adjustments. The heat regulating center fails, when the external temperature is so abnormally high that bodily heat cannot be eliminated as fast as it is produced. Part of it is retained in the body, causing a rise in skin and deep tissue temperature, an increase in the heart rate, and accelerated respiration. (See Table 1) In extreme

TABLE 1 PHYSIOLOGICAL RESPONSES TO HEAT OF MEN AT REST AND AT WORK[a]

Effective Temp	Actual Cheek Temp (Deg Fahr)	Men at Rest			Men at Work 90,000 Ft-Lb of Work per Hour			
		Rise in Rectal Temp (Deg Fahr per Hour)	Increase in Pulse Rate (Beats per Min per Hour)	Approximate Loss in Body Weight by Perspiration (Lb per Hr)	Total Work Accomplished (Ft-Lb)	Rise in Body Temp (Deg Fahr per Hr)	Increase in Pulse Rate (Beats per Min per Hr)	Approximate Loss in Body Wt by Perspiration (Lb per Hr)
60	....	....	....	---- --	225,000	0 0	6	0.5
70	....	0 0	0	0.2	225,000	0 1	7	0.6
80	96.1	0 0	0	0 3	209,000	0 3	11	0.8
85	96.6	0 1	1	0.4	190,000	0.6	17	1 1
90	97.0	0 3	4	0 5	153,000	1.2	31	1 5
95	97 6	0 9	15	0 9	102,000	2.3	61	2.0
100	99 6	2 2	40	1 7	67,000	4.0	103[b]	2 7
105	104.7	4 0	83	2 7	49,000	6.0[b]	158[b]	3.5[b]
110	....	5 9[b]	137[b]	4.0[b]	37,000	8.5[b]	237[b]	4.4[b]

[a]Data by A.S H V E. Research Laboratory.
[b]Computed value from exposures lasting less than one hour.

heat, the metabolic rate is markedly increased owing to the excessive rise in body temperature[8], and a vicious cycle results which may eventually lead to serious physiologic damage.

Examples of this are met with in unusually hot summer weather and in hot industries where the radiant heat from hot objects renders heat loss from the body by radiation and convection impossible. Consequently, the workers depend entirely on evaporation for the elimination of body heat. They stream with perspiration and drink liquids abundantly to replace the loss.

One of the deleterious effects of high temperatures is that the blood is diverted from the internal organs to the surface capillaries, in order to serve in the process of cooling. This affects the stomach, heart, lungs and other vital organs, and it is suggested that the feeling of lassitude and discomfort experienced is due in part to the anæmic condition of the brain. The stomach loses some of its power to act upon the food, owing to a

[8]Basal Metabolism Before and After Exposure to High Temperatures and Various Humidities, by W J McConnell, C P Yaglou and W. B. Fulton (A.S H.V.E TRANSACTIONS, Vol. 31, 1925).

diminished secretion of gastric juice, and there is a corresponding loss in the antiseptic and antifermentive action which favors the growth of bacteria in the intestinal tract[9]. These are considered to be the potent factors in the increased susceptibility to gastro-intestinal disorders in hot summer weather. The victim may lose appetite and suffer from indigestion, headache and general enervation, which may eventually lead to a premature old age.

In warm atmospheres, particularly during physical work, a considerable amount of chloride is lost from the system through sweating. The loss of this substance may lead to attacks of cramps, unless the salts are replaced in the drinking water. In order to relieve both cramps and fatigue, Moss[10] recommends the addition of 6 grams of sodium chloride and 4 grams of potassium chloride to a gallon of water.

The deleterious physiologic effects of high temperatures exert a powerful influence upon physical activity, accidents, sickness and mortality. Both laboratory and field data show clearly that physical work in warm atmospheres is a great effort, and that production falls progressively as the temperature rises. The incidence of industrial accidents reaches a minimum at about 68 F, increasing above and below that temperature. Sickness and mortality rates increase progressively as the temperature rises.

EFFECTS OF COLD AND TEMPERATURE CHANGES

The action of cold on human beings is not well known. Cold affects the human organism in two ways: (1) through its action on the body as a whole, and (2) through its action on the mucous membranes of the upper respiratory tract. Little exact information is available on the latter.

On exposure to cold, the loss of heat is increased considerably and only within certain limits is compensation possible by increased heat production and decreased peripheral circulation. The rectal temperature often rises upon exposure to cold but the pulse rate and skin temperature fall. The blood pressure increases, owing to constriction in the peripheral vessels. Just how cold affects health is not well understood. It imposes an extra load upon the heat-producing organs to maintain body temperature. The strain falls largely upon digestion, metabolism, blood circulation, and the kidneys, and indirectly upon the nervous system[11].

Although the seasonal increase in morbidity and mortality sets in with the approach of cold weather and subsides in the warm summer months, little is known of the specific causative factors and their mechanism of action. Over-crowding of buildings, overheated rooms, lack of ventilation, and close personal contacts are frequently held responsible for our winter ills, but the evidence is not conclusive.

In extremely cold atmospheres compensation by increased metabolism becomes inadequate. The body temperature falls and the reflex irritability of the spinal cord is markedly affected. The organism may finally pass into an unconscious state which ends in death.

---

[9]Influence of Effective Temperature upon Bactericidal Action of Gastro-Intestinal Tract, by Arnold and Brody (*Proceedings Society Exp Biol Med* Vol 24, 1927, p 832)

[10]Some Effects of High Air Temperatures upon the Miner, by K. N. Moss (*Transactions Institute of Mining Engineers*, Vol 66, 1924, p 284)

[11]Loc. Cit. Note 5

Cannon showed that excessive loss of heat is associated with increased activity of the adrenal medulla[12]. The extra output of adrenin hastens heat production which protects the organism against cooling. Bast[13] found a degeneration of thyroid and adrenal glands upon exposure to cold.

A moderate amount of variability in temperature is known to be beneficial to health, comfort, and the performance of physical and mental work. On the other hand, extreme changes in temperature, such as those experienced in passing from a warm room to the cold air out-of-doors, appear to be harmful to the tissues of the nose and throat which are the portals for the entry of respiratory diseases.

Experiments show that chilling causes a constriction of the blood vessels of the palate, tonsils, throat, and nasal mucosa, which is accompanied by a fall in the temperature of the tissues. On rewarming, the palate and throat do not always regain their normal temperature and blood supply. This anæmic condition favors bacterial activity and it probably plays a part in the disposition of common cold and other respiratory diseases. It is believed that the lowered resistance is due to a diminution in the number and phagocytic activity of the leucocytes (white blood cells) brought about by exposure to cold and by changes in temperature.

Sickness records in industries seem to strengthen this belief. The Industrial Fatigue Research Board of England[14] found that in workers exposed to high temperatures and to changes in temperature, namely, steel melters, puddlers, and general laborers, there is an excess of all sickness, the excess among the puddlers being due chiefly to respiratory diseases and rheumatism. The causative factor was not the heat itself but the sudden changes in temperature to which the workers were exposed. The tin-plate millmen who were not exposed to chills, since they work almost continuously throughout the shift, had no excess of rheumatism and respiratory diseases. On the other hand, the blast-furnacemen, who work mostly in the open, showed more respiratory sickness than the steel workers. This experience in British factories is well in accord with the findings in American industries[15, 16]. According to these data the highest pneumonia death rate is associated with dust, extreme heat, exposure to cold, and to sudden changes in temperature.

ACCLIMATIZATION

Acclimatization and the factor of psychology are two important influences in air conditioning which cannot be ignored. The first is man's ability to adapt himself to changes in air conditions; the second is an intangible matter of habit and suggestion.

Some persons regard the unnecessary endurance of cold as a virtue.

[12]Studies on the Condition of Activity of Endocrine Glands, by W B. Cannon, A. Guerido, S. W. Britton and E M. Bright (*American Journal of Physiology*, Vol. 79, 1926, p. 460)

[13]Studies in Exhaustion Due to Lack of Sleep, by T H. Bast, J. S. Supernaw, B. Lieberman and J. Munro (*American Journal of Physiology*, Vol. 85, 1928, p. 135).

[14]Fatigue and Efficiency in the Iron and Steel Industry, by H. M. Vernon (*Industrial Fatigue Research Board*, Report No. 5, 1920, London)

[15]Iron Foundry Workers Show Highest Percentage of Deaths from Pneumonia (*Statistical Bulletin*, Metropolitan Life Insurance Company, 1928)

[16]The Pneumonia Problem in the Steel Industry, by D. K Brundage and J. J. Bloomfield, (*Journal of Industrial Hygiene*, 14, December, 1932)

They believe that the human organism can adapt itself to a wide range of air conditions with no apparent discomfort or injury to health. In the light of the present knowledge of air conditioning these views are not justified. Acclimatization to extreme conditions involves a strain upon the heat regulating system and it interferes with the normal physiologic functions of the human body. Thousands of years in the heat of Africa do not seem to have acclimatized the Negro to a temperature averaging 80 F. The same holds true of northern races with respect to cold, although the effects are mitigated by artificial control. All this seems to indicate that adaptation to an environment averaging between 60 and 80 F is a very primitive trait[17].

Within these limits, however, there does occur a definite adaptation to external temperature level. People and animals raised under conditions of tropical moist heat have a lower rate of heat production than do those who grow up in cooler environments. This causes them to stand chilling poorly as they are unable to quickly increase internal combustion to keep up the body temperature. For this reason they have trouble standing the cold, stormy weather of the temperate zones, and when exposed to it are very susceptible to respiratory infections. Likewise, people living in cool climates suffer greatly in the moist heat of the tropics until their adrenal activity has slowed down. Within a couple of years, however, they find themselves standing the heat much better and disliking cold They become acclimated by a definite change in the combustion level within the body[18].

In certain individuals the psychologic factor is more powerful than acclimatization. A fresh air fiend may suffer in a room with windows closed regardless of the quality of the air. As a matter of fact, instances are known in which paid subjects refused to stay in a windowless but properly conditioned experimental chamber because the atmosphere felt suffocating to them upon entering the room.

EFFECTIVE TEMPERATURE INDEX OF WARMTH

Sensations of warmth or cold depend, not only on the temperature of the surrounding air as registered by a dry-bulb thermometer, but also upon the temperature indicated by a wet-bulb thermometer. Dry air at a relatively high temperature may feel cooler than air of considerably lower temperature with a high moisture content. Air motion makes any moderate condition feel cooler.

On the other hand, in cold environments an increase in humidity produces a cooler sensation. The dividing line at which humidity has no effect upon warmth varies with the air velocity and is about 46 F (dry-bulb) for still air and about 51, 56 and 59 F for air velocities of 100, 300 and 500 fpm, respectively. Radiation from cold or warm surfaces is another important factor under certain conditions.

Combinations of temperature, humidity and air movement which induce the same feeling of warmth are called thermo-equivalent con-

[17]Civilization and Climate, by Ellsworth Huntington, Yale University Press, 1924

[18]Air Conditioning in its Relation to Human Welfare, by C. A Mills, M D (A S H V E. Transactions Vol 40, 1934)

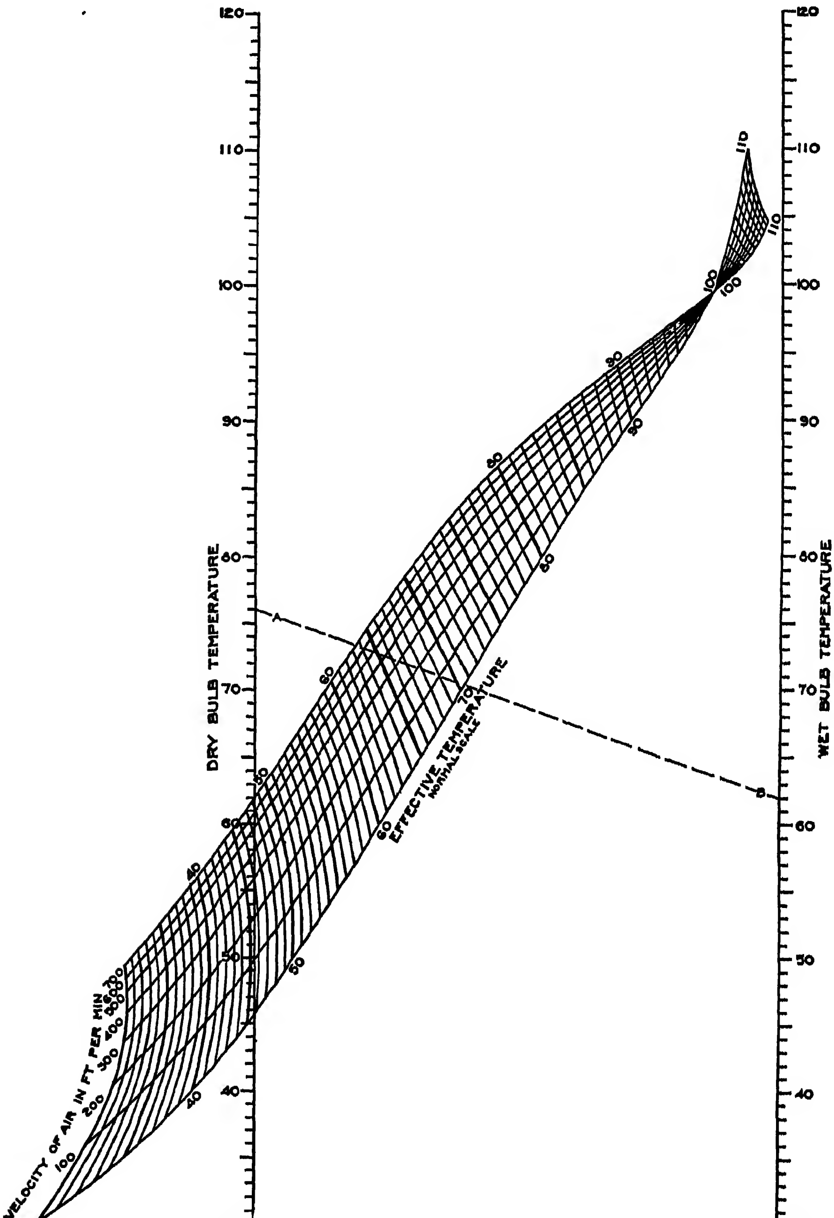


FIG 1 EFFECTIVE TEMPERATURE CHART SHOWING NORMAL SCALE OF EFFECTIVE TEMPERATURE. APPLICABLE TO INHABITANTS OF THE UNITED STATES UNDER FOLLOWING CONDITIONS

A Clothing: Customary indoor clothing *B Activity.* Sedentary or light muscular work. *C. Heating Methods* Convection type, *i e* warm air, direct steam or hot water radiators, plenum systems.

ditions. A series of tests[19, 20, 21, 22] at the A.S.H.V.E. Research Laboratory, Pittsburgh, established the equivalent conditions met with in general air conditioning work. This scale of thermo-equivalent conditions not only indicates the sensation of warmth, but also determines the physiological *effects* on the body induced by heat or cold. For this reason, it is called the *effective temperature* scale or index.

Effective temperature is an empirically determined index of the degree of warmth perceived on exposure to different combinations of temperature, humidity, and air movement. It was determined by trained subjects who compared the relative warmth of various air conditions in two adjoining conditioned rooms by passing back and forth from one room to the other.

Effective temperature is not in itself an index of comfort, except under ordinary humidity conditions (30 to 60 per cent) when the individual is least conscious of humidity. Moist air at a comparatively low temperature, and dry air at a higher temperature may each feel as warm as air of an intermediate temperature and humidity, but the *comfort* experienced in the three air conditions would be different, although the effective temperature is the same.

Air of proper warmth may, for instance, contain excessive water vapor, and in this way interfere with the normal physiologic loss of moisture from the skin, leading to damp skin and clothing and producing more or less discomfort; or the air may be excessively dry, producing appreciable discomfort to the mucous membrane of the nose and to the skin which dries up and becomes chapped from too rapid loss of moisture.

The numerical value of the effective temperature index for any given air condition is fixed by the temperature of calm (15 to 25 fpm air movement) and saturated air which induces a sensation of warmth or cold like that of the given condition. Thus, any air condition has an effective temperature of 60 deg, for instance, when it induces a sensation of warmth like that experienced in calm air at 60 deg saturated with moisture. The effective temperature index cannot be measured directly but it is computed from the dry- and wet-bulb temperature and the velocity of air using charts (See Figs. 1 or 2, 3 and 4) or tables. The accuracy in estimating effective temperature is ± 0.5 F, because the human organism cannot perceive smaller temperature differences. Therefore, there is no justification in trying to read chart values closer than 0.5 F, as this implies fictitious accuracy.

The charts shown in Figs. 1, 2, 3 and 4 apply to average normal and healthy persons adapted to American living and working conditions. Application is limited to sedentary or light muscular activity, and to rooms heated by the usual American convection methods (warm air, central fan and direct hot water and steam heating systems) in which the difference between the air and wall surface temperatures may not be too

[19]Determining Lines of Equal Comfort, by F C Houghten and C P Yaglou (A S H V E TRANSACTIONS, Vol 29, 1923, p 361)

[20]Cooling Effect on Human Beings by Various Air Velocities, by F C Houghten and C. P Yaglou (A S H V E. TRANSACTIONS, Vol 30, 1924, p 193)

[21]Effective Temperature with Clothing, by C P Yaglou and W. E. Miller (A S.H V E TRANSACTIONS, Vol 31, 1925, p 89)

[22]Effective Temperature for Persons Lightly Clothed and Working in Still Air, by F C Houghten, W W Teague and W E Miller (A S H V E TRANSACTIONS, Vol 32, 1926)

great. The charts do not apply to rooms heated by radiant method such as British panel system, open coal fires and similar usages. They will probably not apply to races other than the white or perhaps to inhabitants of other countries where the living conditions, climate, heating methods, and clothing are materially different than those of the subjects employed in experiments at the Research Laboratory.

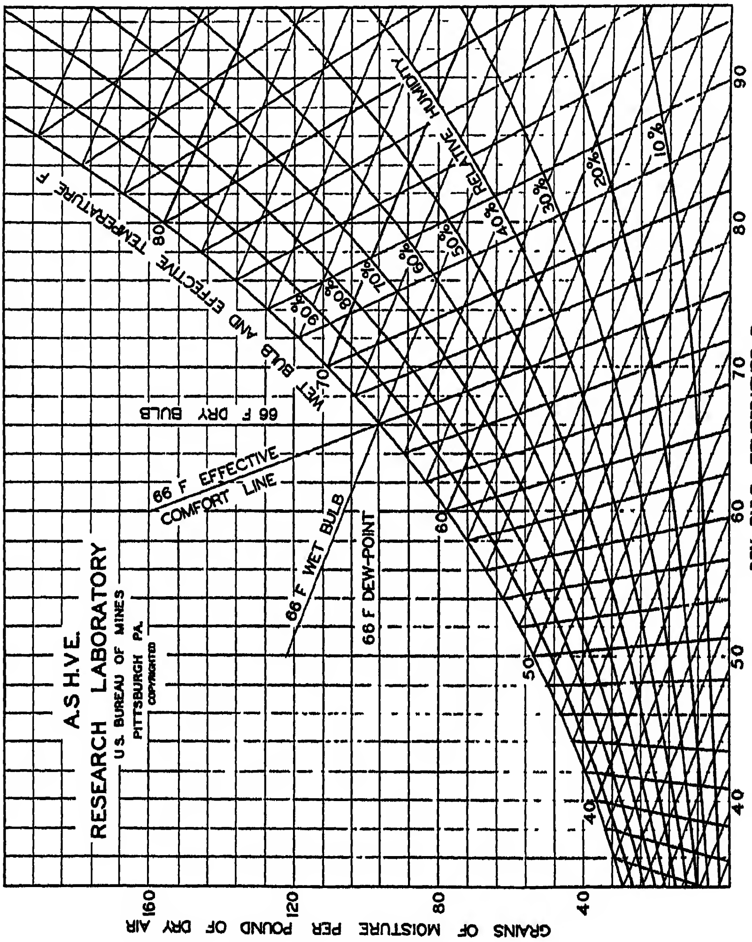


Fig. 2. Psychrometric Chart, Persons at Rest, Normally Clothed, in Still Air

In rooms in which the average wall surface temperature is considerably below or above air temperature, a correction must be applied to the readings of the dry-bulb thermometer to allow for such negative or positive radiation. In Fig. 5 is given the cooling effect of cold walls as determined at the A S.H.V.E. Research Laboratory[23] by trained subjects

[23]Cold Walls and Their Relation to the Feeling of Warmth, by F. C. Houghten and Paul McDermott (A S H V E Transactions, Vol 39, 1933)

passing back and forth from a small experimental room having three cold walls, to a control room with walls and air at the same temperature.

It can be seen in Fig. 5 that with air and walls at 70 F in the control (warm wall room), the cooling effect of three cold walls at 55 F of the experimental room was 4 F. Therefore, for the same feeling of warmth,

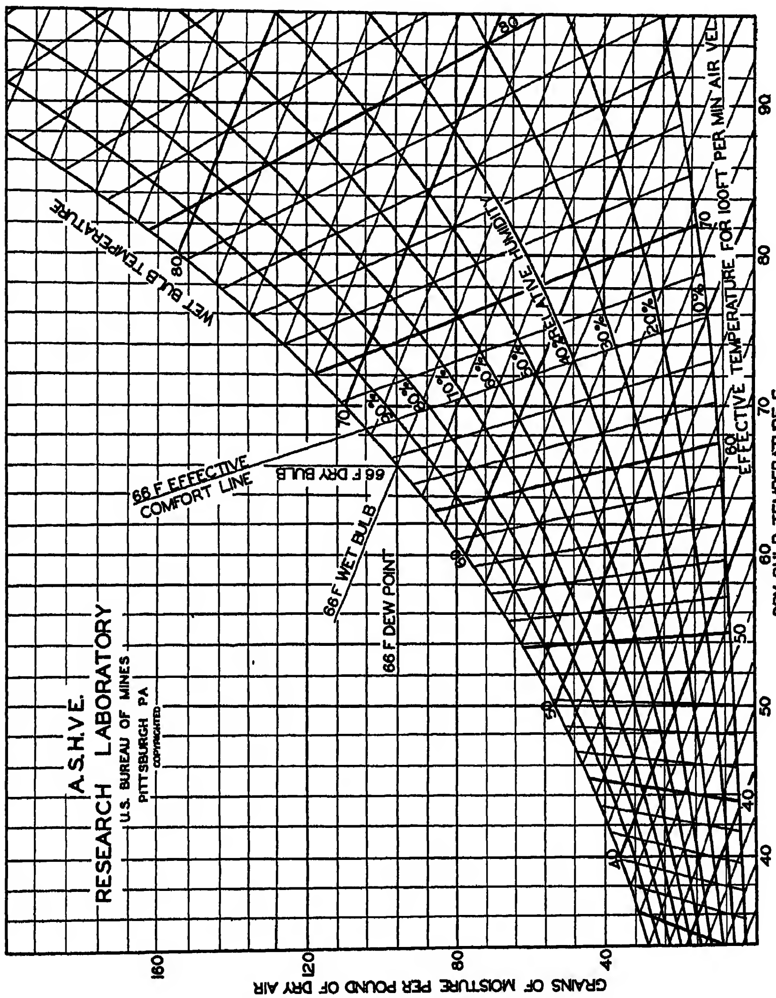


Fig. 3. Psychrometric Chart, Persons Normally Clothed at Rest, in 100 Fpm Air Velocity

the temperature in the experimental room should be increased to 74 F. The reverse would hold in rooms with high-wall surface temperature; a lower air temperature would be required to compensate for positive radiations to the occupants.

OPTIMUM AIR CONDITIONS

No single comfort standard can be laid down which would meet every need. There is an inherent individual variation in the sensation of warmth or comfort felt by persons when exposed to an identical atmospheric condition. The state of health, age, sex, clothing, activity, and

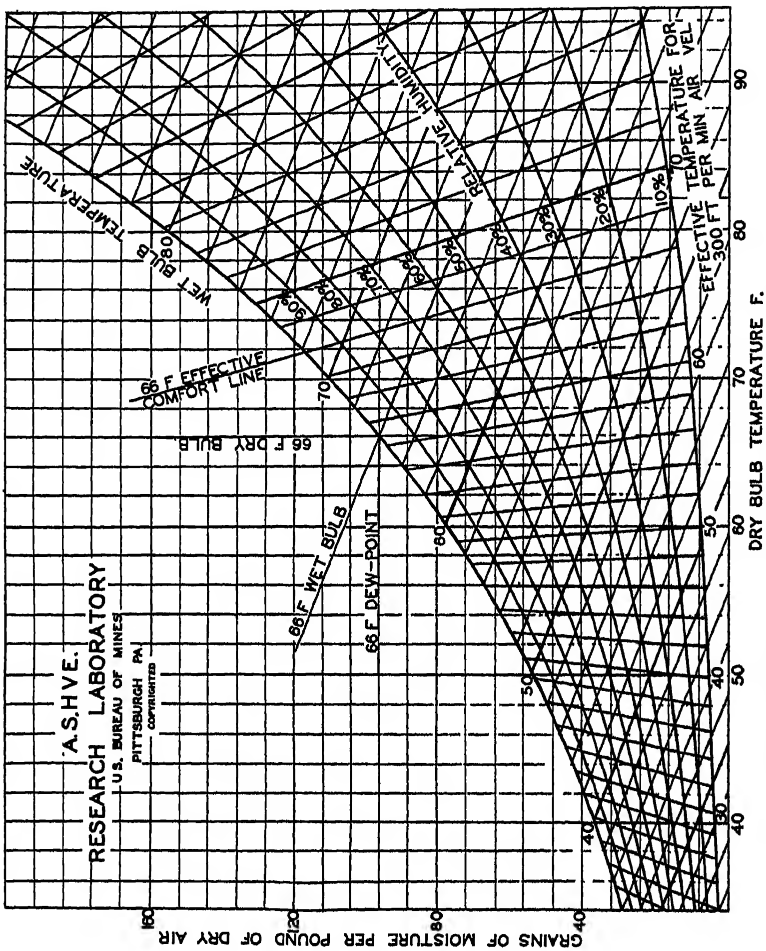


Fig. 4. Psychrometric Chart, Persons Normally Clothed at Rest, in 300 Fpm Air Velocity

the degree of acquired adaptation seem to be the important factors affecting the comfort standards.

Since the prolonged effects of temperature, humidity and air movement on health are not known to the same extent as their effects on comfort, the optimum conditions for health may not be identical with those for comfort. On general physiologic grounds, however, the two do not differ greatly since this is in accordance with the efficient operation of the

heat regulating mechanism of the body. This belief is strengthened by results of studies on premature infants over a four-year period[24]. By adjusting the temperature and humidity so as to stabilize the body temperature of these infants, the incidence of diarrhœa and mortality was decreased, gains in body weight increased and infections were reduced to a minimum.

Winter Comfort Zone and Comfort Line

In Fig. 6 is shown the A S.H.V.E. winter comfort zone which was determined experimentally with large groups of men and women subjects wearing customary indoor winter clothing. The extreme comfort zone includes conditions between 60 and 74 deg ET in which one or more of the experimental subjects were comfortable. The average comfort zone

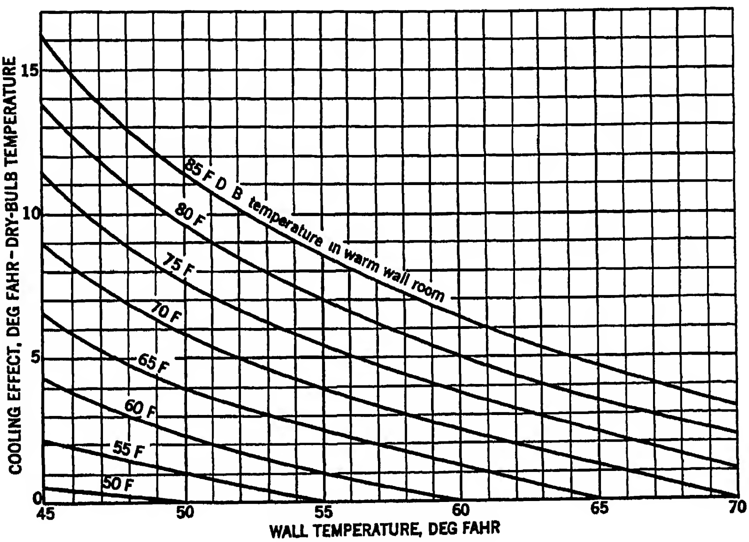


FIG. 5. COOLING EFFECT OF THREE COLD WALLS IN A SMALL EXPERIMENTAL ROOM, AS DETERMINED BY COMPARISON WITH SENSATIONS IN A ROOM OF UNIFORM WALL AND AIR TEMPERATURE

includes conditions between 63 and 71 deg ET conducive to comfort in 50 per cent or more of the experimental subjects. The most popular effective temperature was found to be 66 deg, and was adopted by the Society[25] as the *winter comfort* line for individuals at rest wearing customary winter clothing.

The comfort line separates the cool air conditions to its left from the warm air conditions to its right. Under the air conditions existing along or defined by the comfort line, the body is able to maintain thermal

[24]Application of Air Conditioning to Premature Nurseries in Hospitals, by C P Yaglou, Philip Drinker and K D. Blackfan (A S H V E TRANSACTIONS, Vol 36, 1930).

[25]How to Use the Effective Temperature Index and Comfort Charts (A S H V E TRANSACTIONS, Vol. 38, 1932)

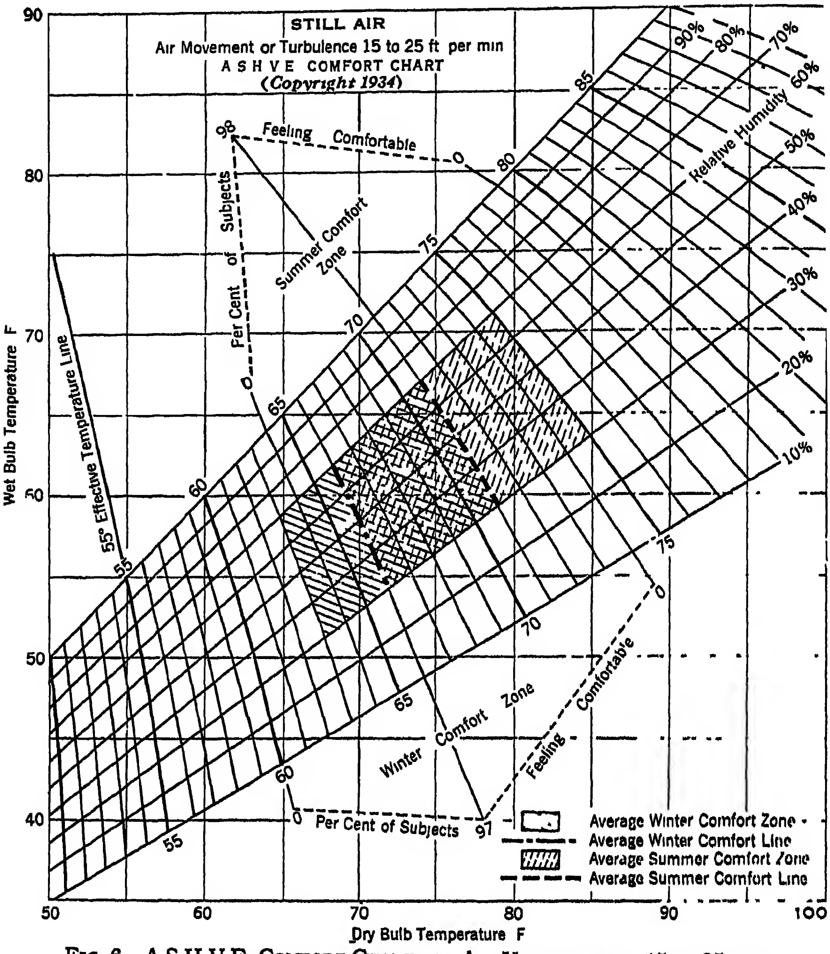


FIG. 6. A S.H.V.E. COMFORT CHART FOR AIR VELOCITIES OF 15 TO 25 FPM (STILL AIR)[26, 27]

Note.—Both summer and winter comfort zones apply to inhabitants of the United States only. Application of winter comfort line is further limited to rooms heated by central station systems of the convection type. The line does not apply to rooms heated by radiant methods. Application of summer comfort line is limited to homes, offices and the like, where the occupants become fully adapted to the artificial air conditions. The line does not apply to theaters, department stores, and the like where the exposure is less than 3 hours.

equilibrium with its environment with the least conscious sensation to the individual, or with the minimum physiologic demand on the heat regulating mechanism. This environment involves not only the condition of the air with respect to temperature and humidity, but also the condition of the surrounding objects and wall surfaces. The comfort zone tests were

[26]Determination of the Comfort Zone With Further Verification of Effective Temperatures Within This Zone, by F C Houghten and C. P. Yaglou (A.S H V.E. TRANSACTIONS, Vol. 29, 1923, p. 301).

[27]The Summer Comfort Zone; Climate and Clothing, by C. P. Yaglou and Philip Drinker (A.S.H.V.E. TRANSACTIONS, Vol. 35, 1929).

made in rooms with wall surface temperatures approximately the same as the room dry-bulb temperature. For walls of large area having unusually high or low surface temperatures, however, a somewhat lower or higher range of effective temperature is required to compensate for the increased gain or loss of heat to or from the body by radiation as shown in Fig. 5. (See also Chapter 38).

The average winter comfort line (66 deg ET) applies to average American men and women living inside the broad geographic belt across the United States in which central heating of the convection type is generally used during four to eight months of the year. It does not apply to rooms heated by radiant energy, rooms with excessive glass area or rooms with poorly insulated or cold walls. Even in the warm south and southwestern climates, and in the very cold north-central climate of the United States, the comfort chart would probably have to be modified according to climate, living and working conditions, and the degree of acquired adaptation.

In densely occupied spaces, such as classrooms, theaters and auditoriums, somewhat lower temperatures may be necessary than those indicated by the comfort line on account of counter-radiation between the bodies of occupants in close proximity[28].

The sensation of comfort, insofar as the physical environment is concerned, is not absolute but varies considerably among certain individuals. Therefore, in applying the air conditions indicated by the comfort line, it should not be expected that all the occupants of a room will feel perfectly comfortable. When the winter comfort line is applied in accordance with the foregoing recommendations, the majority of the occupants will be perfectly comfortable, but there will always be a few who would feel *a bit too cool* and a few *a bit too warm.* These individual differences among the minority should be counteracted by suitable clothing.

Air conditions lying outside the average comfort zone but within the extreme comfort zone may be comfortable to certain persons. In other words, it is possible for half of the occupants of a room to be comfortable in air conditions outside the *average* comfort zone, but in the majority of cases, if not in all, these conditions will be well within the extreme comfort zone as determined experimentally.

The comfort chart (Fig. 6) applies to adults between 20 and 70 years of age living in the northeastern parts of the United States. For prematurely born infants, the optimum temperature varies from 100 F to 75 F, depending upon the stage of development. The optimum relative humidity for these infants is placed at 65 per cent[29]. No data are yet available on the optimum air conditions for full term infants and young children up to school age. Satisfactory air conditions for these age groups are assumed to vary from 75 F to 68 F with natural indoor humidities. For school children, the studies of the *New York State Commission on Ventilation* place the optimum air conditions at 66 F to 68 F temperature with a moderate humidity (not specified) and a moderate but not excessive amount of air movement (not specified)[30].

---

[28]Loc. Cit. Note 27

[29]Loc. Cit. Note 24

[30]Ventilation (Report N Y State Commission on Ventilation E P Dutton and Co, N Y., 1923)

Satisfactory comfort conditions for men at work are found to vary from 40 deg to 70 deg ET, depending upon the rate of work and amount of clothing worn[31]. In hot industries, 80 deg ET is considered the upper limit compatible with efficiency, and, whenever possible, this should be reduced to 70 deg ET or less.

Summer Comfort Zones

The summer comfort zone is much more difficult to fix than the winter zone owing to the complicating factor of sweating in warm weather. A given air condition which is comfortable for persons with dry skin and clothing may prove too cold for those perspiring, as is the case, for instance, with employees and customers in a cooled store, restaurant, or theater, on a warm summer day. The conditions to be maintained in different types of public buildings depend to a large extent upon the occupants' length of stay and upon the prevailing outdoor condition.

In Fig. 6 is shown the summer comfort zone for exposures of 3 hours or more, after adaptation has taken place. The average zone extends from 66 to 75 deg ET, with a comfort line at 71 deg ET, as determined at the Harvard School of Public Health[32]. These effective temperatures average about 4 deg higher than those found in winter when customary winter clothing was worn. The variation from winter to summer is probably due partly to adaptation to seasonal weather and partly to differences in the clothing worn in the two seasons.

The best effective temperature (for exposures lasting 3 hours or more) was found to follow the average monthly outdoor temperature more closely than the prevailing outdoor temperature. It remained at approximately the same value in July, August and September, and although the average monthly temperature did not vary much, the prevailing outdoor temperature ranged from 70 F to 99.5 F. A decrease in the optimum temperature became apparent only when the prevailing outdoor temperature fell to 66 F, which is below the customary room temperature in the United States for summer and winter.

Crowding the experimental chamber lowered the comfortable effective temperature from 70.8 deg when the gross floor area per occupant was 44 sq ft and the air space 380 cu ft, to 69.4 deg when the floor area was reduced to 14 sq ft and the air space to 120 cu ft per occupant.

The basic summer comfort zone, shown in Fig. 6 has more academic than practical significance. It prescribes conditions of choice for continuous exposures, as in homes, offices, etc., without regard to costs, prevailing outdoor air conditions, and temperature contrasts upon entering or leaving the cooled space. A great number of persons seem to be content with a higher plane of indoor temperature, particularly when the matter of first cost and cost of operation of the cooling plant is given due consideration.

According to previous investigations[33], an indoor temperature of about 80 F with relative humidities below 55 per cent, or 74.5 deg ET and lower, result in satisfactory comfort conditions in the living quarters of a residence,

---

[31]Loc Cit Note 22
[32]Loc. Cit Note 27
[33]Study of Summer Cooling in the Research Residence for the Summer of 1934, by A. P. Kratz, S. Konzo, M. K Fahnestock and E L. Broderick (A S.H V E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, January, 1935)

and while this condition is not representative of optimum comfort it provides for sufficient relief in hot weather to be acceptable to the majority of users. Experience in a number of air conditioned office buildings, including the New Metropolitan Life Building in New York[34], indicates that a temperature of about 80 F with a relative humidity between 45 and 55 per cent (73 to 74.5 deg ET) is generally satisfactory in meeting the requirements of the employees.

In artificially cooled theaters, restaurants, and other public buildings where the period of occupancy is short, the contrast between outdoor and indoor air conditions becomes the deciding factor in regard to the temperature and humidity to be maintained. The object of cooling such places in the summer is to provide sufficient relief from the heat without causing sensations of chill or intense heat on entering and leaving the building.

TABLE 2 DESIRABLE INDOOR AIR CONDITIONS IN SUMMER CORRESPONDING TO OUTDOOR TEMPERATURES

Applicable to Exposures Less Than 3 Hours

OUTDOOR TEMPERATURE (DEG FAHR)	INDOOR AIR CONDITIONS		
		CONSTANT DEW-POINT 57 F	
DRY-BULB	EFFECTIVE TEMP	DRY-BULB[a]	WET-BULB[a]
95	73	80.0	65 0
90	72	78 0	64 5
85	71	76 5	64 0
80	70	75.0	63 5
75	69	73 5	63 0
70	68	72 0	62 5

[a]Dry- and wet-bulb readings give the corresponding effective temperatures indicated in the adjacent column Recent research (Comfort Standards for Summer Air Conditioning, by F C Houghten and Carl Gutberlet, A S H V E JOURNAL SECTION, *Heating, Piping and Air Conditioning*, November, 1935, and Cooling Requirements for Summer Air Conditioning, by F C Houghten, F E Giesecke, Cyril Tasker and Carl Gutberlet, A S H V E JOURNAL SECTION, *Heating, Piping and Air Conditioning*, December, 1936), indicates that the effective temperature only is important from a standpoint of comfort and that the dry- and wet-bulb temperatures may vary over a considerable range if the proper effective temperature is maintained

Effective temperatures as high as 75 F at times have been found satisfactory in very warm weather. There are two schools of thought concerning the relation between temperature and humidity to be maintained. For a given effective temperature some engineers including the operators of cooling plants favor a comparatively low temperature with a high humidity as this results in a reduction of refrigeration requirements. Preliminary experiments at the A.S.H.V.E. Laboratories[35] would seem to indicate no appreciable impairment of comfort with relative humidities as high as 80 per cent, provided the effective temperature is between 70 and 75 deg.

The second school favors a higher dry-bulb temperature, according to the prevailing outdoor dry-bulb, with a comparatively low humidity (well below 50 per cent); the main purpose being to reduce temperature

[34]The Air Conditioned System of the New Metropolitan Building—First Summer's Experience, by W. J McConnell and I B Kagey (A S H V E TRANSACTIONS, Vol 40, 1934)

[35]Comfort Standards for Summer Air Conditioning, by F C Houghten and Carl Gutberlet (A S H V E JOURNAL SECTION, *Heating, Piping and Air Conditioning*, November, 1935)

contrasts upon entering and leaving the *cooled* space and to keep the clothing and skin dry. This second scheme requires more refrigeration with the present conventional type of apparatus.

Current practice in theaters, restaurants, etc., follows a schedule similar to that shown in Table 2 This schedule has been the subject of much discussion and criticism but so far no satisfactory substitute has been offered. It is, in fact questionable whether entirely satisfactory air conditions could be adduced for practical use to meet the changing requirements of patrons from the time they enter to the time they leave a cooled space. Too many uncontrollable variables enter into the problem. Work now going on at the A.S.H.V.E. Laboratories and other interested institutions may throw considerable light on this complex problem.

For cooled banks and stores where the customers come and go spending but a few minutes in the cooled space, observations[36] indicate a schedule about 1 deg dry-bulb or effective temperature higher than that shown in Table 2. Laboratory experiments with exposures of 2 to 10 min indicate temperatures 2 to 10 F higher than those in Table 2 but with much lower relative humidities.

It should be kept in mind that southern people, with their more sluggish heat production and lack of adaptability, will demand a comfort zone several degrees higher than that for the more active people of northern climates. Instead of the summer comfort line standing at 71 deg as here given, it was found to be much higher for foreigners in Shanghai where climatic conditions are similar to those of our gulf states. This difference in adaptability of people forms a very real problem for air conditioning engineers. Cooling of theaters, restaurants, and other public buildings in southern climates cannot be based on northern standards without considerable modification.

Optimum Humidity

Just what the optimum range of humidity is, is a matter of conjecture. There seems to exist a general opinion, supported by some experimental and statistical data, that warm, dry air is less pleasant than air of a moderate humidity, and that it dries up the mucous membranes in such a way as to increase susceptibility to colds and other respiratory disorders [37, 38, 39]. Owing to the cooling effect of evaporation, higher temperatures are necessary, and this condition may lead to discomfort and lassitude. Moist air, on the other hand, interferes with the normal evaporation of moisture from the skin, and again may cause a feeling of oppression and lassitude, especially when the temperature is also high. For the premature infant, a high relative humidity of about 65 per cent is demonstrably beneficial to health and growth[40] until the infants reach a

[36]How Cool? Inside Temperature should Depend upon Type of Occupancy, by J. H. Walker (*Heating and Ventilating*, October, 1932)

[37]Reactions of the Nasal Cavity and Post-Nasal Space to Chilling of the Body Surface, by Mudd, Stuart, et al (*Journal Experimental Medicine*, 1921, Vol. 34, p. 11).

[38]Reactions of the Nasal Cavity and Post-Nasal Space to Chilling of the Body Surfaces, by A. Goldman, et al and Concurrent Study of Bacteriology of Nose and Throat (*Journal Infectious Diseases*, 1921, Vol. 29, p 151)

[39]The Etiology of Acute Inflammations of the Nose, Pharynx and Tonsils, by Mudd, Stuart, et al (Am Otol, Rhinol, ,and Laryngol., 1921).

[40]Loc Cit. Note 24

weight of about 5 lb. No such clear-cut evidence exists in the case of adult persons. In the comfort zone experiments of the A.S.H.V.E. Research Laboratory, the relative humidity was varied between the limits of 30 and 70 per cent approximately, but the most comfortable range has not been determined. In similar experiments at the Harvard School of Public Health, the majority of the subjects were unable to detect sensations of humidity (*i.e.*, too high, too low, or medium) when the relative humidity was between 30 per cent and 60 per cent with ordinary room temperatures. This is in accord with studies by Howell[41], Miura[42] and others.

The limitation of the comfort zones in Fig. 6 with respect to humidity must not be taken too seriously. Relative humidities below 30 per cent may prove satisfactory from the standpoint of comfort, so long as extremely low humidities are avoided. From the standpoint of health, however, the consensus seems to favor a relative humidity between 40 and 60 per cent In mild weather such comparatively high relative humidities are entirely feasible, but in cold or sub-freezing weather they are objectionable on account of condensation and frosting on the windows. They may even cause serious damage to certain building materials of the exposed walls by condensation and freezing of the moisture accumulating inside these materials. Unless special precautions are taken to properly insulate the affected surfaces, it will be necessary to reduce the degree of artificial humidification in sub-freezing weather to less than 40 per cent, according to the outdoor temperature. Information on the prevention of condensation on building surfaces is given in Chapter 7. The principles underlying humidity requirements and limitations are discussed more fully elsewhere[43].

The purpose of artificial humidification may be easily defeated by failure to change the spray water of the humidifier at least daily. Where this condition occurs, the air is characterized by a *lack of freshness*, and under extreme conditions by a musty sour odor in the conditioned space.

AIR QUALITY AND QUANTITY

Air Quality

In occupied spaces in which the vitiation is entirely of human origin, the chemical composition of the air, the dust, and often the bacteria content may be dismissed from consideration so that the problem consists in maintaining a suitable temperature with a moderate humidity, and in keeping the atmosphere free from objectionable odors. Such unpleasant odors, human or otherwise, can be easily detected by persons entering the room from clean, odorless air.

In industrial rooms where the primary consideration is the control of air pollution (dusts, fumes, gases, etc.), or contamination not removable at the source of production, the clean air supply must be sufficient to dilute the polluting elements to a concentration below the physiological threshold (See Chapters 15 and 16).

[41]Humidity and Comfort, by W H. Howell (*The Science Press*, April, 1931).

[42]Effect of Variation in Relative Humidity upon Skin Temperature and Sense of Comfort, by U Miura (*American Journal of Hygiene*, Vol 13, 1931, p 432).

[43]Humidification for Residences, by A. P. Kratz (*University of Illinois Engineering Experiment Station Bulletin* No 230, July 28, 1931)

Air Quantity

The air supply to occupied spaces must always be adequate to satisfy the physiological requirements of the occupants. It must be sufficient to maintain the desired temperature, humidity, and purity with reasonable uniformity and without drafts. In many practical instances there are two air quantities to be considered, (a) outdoor air supply, and (b) total air supply. The difference between the two gives the amount of air to be recirculated.

TABLE 3 MINIMUM OUTDOOR AIR REQUIREMENTS TO REMOVE OBJECTIONABLE BODY ODORS

(Provisional values subject to revision upon completion of work)

TYPE OF OCCUPANTS	AIR SPACE PER PERSON CU FT	OUTDOOR[a] AIR SUPPLY CFM PER PERSON
Heating season with or without recirculation. Air not conditioned		
Sedentary adults of average socio-economic status	100	25
Sedentary adults of average socio-economic status	200	16
Sedentary adults of average socio-economic status	300	12
Sedentary adults of average socio-economic status	500	7
Laborers	200	23
Grade school children of average class	100	29
Grade school children of average class	200	21
Grade school children of average class	300	17
Grade school children of average class	500	11
Grade school children of poor class	200	38
Grade school children of better class	200	18
Grade school children of best class	100	22
Heating season. Air humidified by means of centrifugal humidifier. Water atomization rate 8 to 10 gph. Total air circulation 30 cfm per person.		
Sedentary Adults	200	12
Summer season. Air cooled and dehumidified by means of a spray dehumidifier. Spray water changed daily. Total air circulation 30 cfm per person.		
Sedentary Adults	200	<4

[a]Impressions upon entering room from relatively clean air at threshold odor intensity.

When the only source of contamination is the occupant, the minimum quantity of outdoor air needed appears to be that necessary to remove objectionable body odors, or tobacco smoke. The concentration of body odor in a room, in turn, depends upon a number of factors, including socio-economic status of occupants, outdoor air supply, air space allowed per person, odor adsorbing capacity of air conditioning processes, temperature, and other factors of secondary importance. With any given group of occupants and type of air conditioner the intensity of body odor perceived upon entering a room from relatively clean air was found to

vary inversely with the logarithm of outdoor air supply and the logarithm of the air space allowed per person.

The minimum outdoor air supply necessary to remove objectionable body odors under various conditions, as determined experimentally at the Harvard School of Public Health[44], is given in Table 3.

Outdoor air requirements for the removal of objectionable tobacco smoke odors have yet to be determined. Practical values in the field vary from 5 to 15 cfm per person; this air quantity may and should be a part

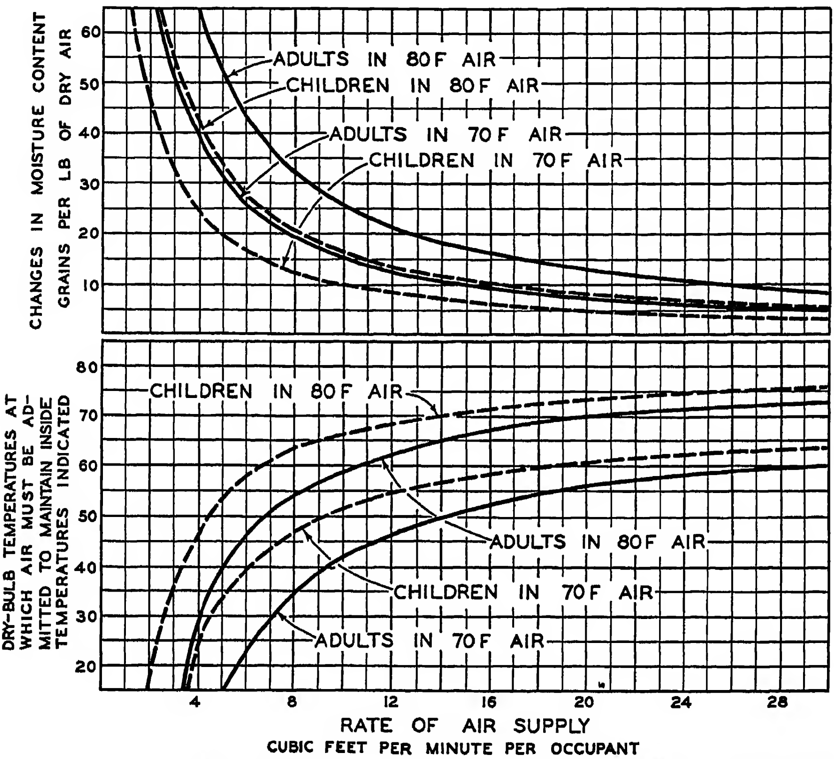


FIG 7 RELATION AMONG RATE OF AIR CHANGE PER OCCUPANT, MOISTURE CONTENT OF ENCLOSURE, AND DRY-BULB TEMPERATURE OF INCOMING AIR

of that necessary for other requirements, *i.e.*, removal of body odors, heat, moisture, etc.

The total quantity of air to be circulated through an enclosure is governed largely by the needs for controlling temperature and air distribution when either heating or cooling is required. The factors which determine total air quantity include the type and nature of the building, locality, climate, height or rooms, floor area, window area, extent of occupancy, and last but not least, the method of distribution.

Serious difficulties are often encountered in attempting to cool a room

[44]Loc. Cit Note 3.

with a poor distribution system or with an air supply which is too small to result in uniform distribution without drafts. Some systems of distribution produce drafts with but a few degrees temperature rise, while other systems operate successfully with a temperature rise as high as 35 F. The total air quantity introduced in any particular case is inversely proportional to the temperature rise, and depends largely upon the judgment and ingenuity of the engineer in designing the most suitable system for the particular conditions.

The changes in moisture content resulting from occupation in the atmosphere of a room supplied with various volumes of outside air is shown in Fig. 7. Data are given for an adult, 5 ft 8 in in height, weighing 150 lb, and having a body surface of 19.5 sq ft and for a child, 12 years of age, 4 ft 7 in. in height weighing 76.6 lb and having a body surface area of 12.6 sq ft. Also given in Fig. 7 is the temperature of incoming air necessary to maintain a room temperature of either 70 of 80 F as indicated assuming that there is no heat gain or loss to the room by transmission through the walls, solar radiation or other sources

AIR MOVEMENT AND DISTRIBUTION

Stagnant warm air, not matter how pure, is not stimulating and it detracts to some extent from the quality of air. Experience, and recent field studies by the A.S H.V.E. Research Laboratory[45] place the desirable air movement between 15 and 25 fpm under ordinary room temperatures during the heating season. Objectionable drafts are likely to occur when the velocity of the air current is 40 fpm and the temperature of the air current 2 F or more below the customary winter room temperature. Higher velocities are not objectionable in the summer time when the air temperature exceeds 80 F. Variations in air movement and temperature in different parts of occupied rooms are often indicative of relative air distribution. The work of the A.S.H.V.E. Research Laboratory indicates that an air movement between 15 and 25 fpm with a temperature variation of 3 F or less in different parts of a room, 36 in. above floor, insure satisfactory distribution. Considerable evidence was obtained in these tests to show that measurements of carbon dioxide are not essential for the study of air distribution, or for indirect measurements of outdoor air supply, which can be obtained more conveniently from the increase in moisture content of the ventilating current.

HEAT AND MOISTURE GIVEN UP BY HUMAN BODY

In conditioning air for comfort and health it is necessary to know the rate of sensible and latent heat liberation, from the human body, which in conjunction with other heat loads (see Chapters 5 and 7) determine the capacity of the conditioner. The data in common use are those of the A.S.H.V.E. Research Laboratory[46] shown in Figs. 8, 9, 10 and 11. Other useful data are given in Tables 4, 5 and 6, which are self-explanatory.

[45]Classroom Drafts in Relation to Entering Air Stream Temperature, by F. C. Houghten, H H. Trimble, Carl Gutberlet and M. F. Lichtenfels (A.S.H V E JOURNAL SECTION, *Heating, Piping and Air Conditioning*, July, 1935)

[46]Thermal Exchanges between the Bodies of Men Working and the Atmospheric Environment, by F. C. Houghten, W W Teague, W. E. Miller and W. P. Yant (*American Journal of Hygiene*, Vol. XIII, No. 2, March, 1931, pp. 415-431).

TABLE 4. RELATION BETWEEN METABOLIC RATE AND ACTIVITY[a]

ACTIVITY	HOURLY METABOLIC RATE FOR AVG PERSON OR TOTAL HEAT DISSIPATED, BTU PER HOUR	HOURLY SENSIBLE HEAT DISSIPATED, BTU PER HOUR	HOURLY LATENT HEAT DISSIPATED, BTU PER HOUR	MOISTURE DISSIPATED, PER HOUR	
				GRAINS	LB
Average Person Seated at Rest[1]	384	225	159	1070	0 153
Average Person Standing at Rest[1]	431	225	206	1390	0 199
Tailor[2]	482	225	257	1740	0 248
Office Worker Moderately Active	490	225	265	1790	0 256
Clerk, Moderately Active, Standing at Counter	600	225	375	2530	0 362
Book Binder[2]	626	225	401	2710	0 387
Shoe Maker[2]; Clerk, Very Active Standing at Counter	661	225	436	2940	0 420
Pool Player	680	230	450	3040	0 434
Walking 2 mph[3, 4], Light Dancing	761	250	511	3450	0 493
Metal Worker[2]	862	277	585	3950	0 564
Painter of Furniture[2]	876	280	596	4020	0.575
Restaurant Serving, Very Busy.	1000	325	675	4560	0 651
Walking 3 mph[3]	1050	346	704	4750	0 679
Walking 4 mph[3, 4], Active Dancing, Roller Skating	1390	452	938	6330	0 904
Stone Mason[2]	1490	490	1000	6750	0 964
Bowling	1500	490	1010	6820	0 974
Man Sawing Wood[2]	1800	590	1210	8170	1 167
Slow Run[4]	2290	......	......	......	......
Walking 5 mph[3]	2330	......	......	......	......
Very Severe Exercise[5]	2560	......	......	......	......
Maximum Exertion Different People[4]	3000 to 4800	......	......	......	......

[a]Metabolism rates noted based on tests actually determined from the following authoritative sources [1]A S H V E Research Laboratory, [2]Becker and Hamalainen, [3]Douglas, Haldane, Henderson and Schneider; [4]Henderson and Haggard; and [5]Benedict and Carpenter Metabolic rates for other activities estimated. Total heat dissipation integrated into latent and sensible rates by actual tests for metabolic rates up to 1250 Btu per hour, and extrapolated above this rate Values for total heat dissipation apply for all atmospheric conditions in a temperature range from approximately 60 to 90 F dry-bulb Division of total heat dissipation rates into sensible and latent heat holds only for conditions having a dry-bulb temperature of 79 F. For lower temperatures, sensible heat dissipation increases and latent heat decreases, while for higher temperatures the reverse is true

TABLE 5 DEGREES OF PERSPIRATION FOR PERSONS SEATED AT REST UNDER VARIOUS ATMOSPHERIC CONDITIONS

DEGREE OF PERSPIRATION[a]	ATMOSPHERIC CONDITION					
	95 Per Cent Relative Humidity			20 Per Cent Relative Humidity		
	E T	D. B	W B	E T	D B	W. B
Forehead clammy	**73.0**	73 6	72 4	**75.0**	87.0	60.7
Body clammy	**73.0**	73 6	72.4	**75.0**	87.0	60.7
Body damp	**79.0**	79.7	78.4	**81.0**	97.5	67.5
Beads on forehead	**80.0**	80 8	79 4	**87.0**	109 4	75.2
Body wet	**84.5**	85 4	84.0	**86.5**	108.5	74.6
Perspiration on forehead runs and drips	**88.0**	89 0	87 6	**94.0**	125.2	85 4
Perspiration runs down body	**88.5**	89.5	88.1	**90.0**	116.0	79.5

[a]Forty per cent of subjects registered degree of perspiration equal to or greater than indicated.

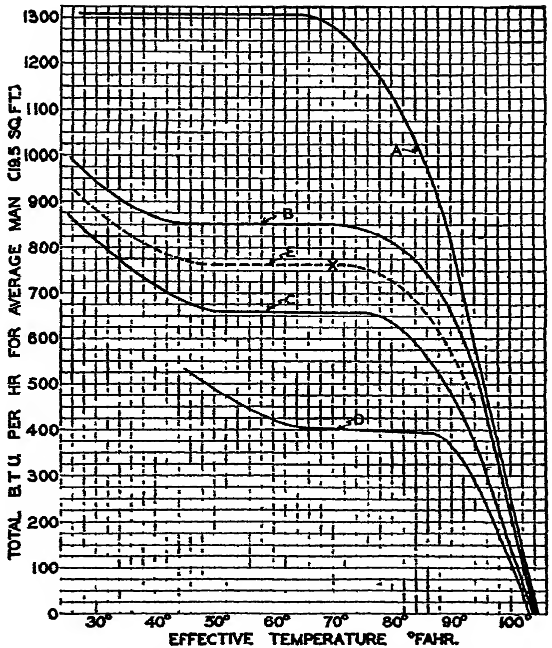


FIG. 8. RELATION BETWEEN TOTAL HEAT LOSS FROM THE HUMAN BODY AND EFFECTIVE TEMPERATURE FOR STILL AIR[a]

[a]Curve *A*—Men working 66,160 ft-lb per hour. Curve *B*—Men working 33,075 ft-lb per hour. Curve *C*—Men working 16,538 ft-lb per hour. Curve *D*—Men seated at rest. Curves *A* and *C* drawn from data at an effective temperature of 70 deg only and extrapolating the relation between curves *B* and *D*, which were drawn from data at many temperatures.

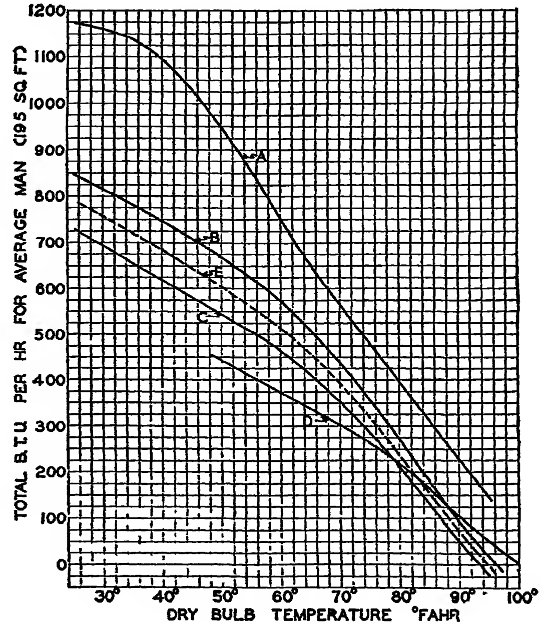


FIG. 9. RELATION BETWEEN SENSIBLE HEAT LOSS FROM THE HUMAN BODY AND DRY-BULB TEMPERATURE FOR STILL AIR[a]

[a]Curve *A*—Men working 66,150 ft-lb per hour Curve *B*—Men working 33,075 ft-lb per hour Curve *C*—Men working 16 538 ft-lb per hour Curve *D*—Men seated at rest Curves *A* and *C* drawn from data at a dry-bulb temperature of 81 3 F only and extrapolating the relation between curves *B* and *D* which were drawn from data at many temperatures

Table 6. Degrees of Perspiration for Persons at Work Under Various Atmospheric Conditions

Work Rate 33,000 Ft Lb per Hour

Degree of Perspiration[a]	Atmospheric Condition					
	95 Per Cent Relative Humidity			20 Per Cent Relative Humidity		
	E T	D B	W B	E T	D B	W.B
Forehead clammy	59.0	59 4	58 3	69.5	80 5	56 5
Body clammy	50.0	50 2	49 3	57.0	61 6	44 2
Body damp	60.0	60 3	59 3	62.5	69 6	49 5
Beads on forehead	68.0	68 5	67 5	76.0	91 0	63 4
Body wet	69.0	69 6	68 5	71.0	82 8	53 0
Perspiration on forehead runs and drips	78.5	79.3	78 0	82.0	100 5	70 2
Perspiration runs down body	79.0	79.8	78 5	81.0	99.8	69 0

[a]Forty per cent of subjects registered degree of perspiration equal to or greater than indicated

ULTRA-VIOLET RADIATION AND IONIZATION

In spite of the rapid advances in the field of air conditioning during the past few years, the secret of reproducing indoor atmospheres of as stimulating qualities as those existing outdoors under ideal weather conditions, has not as yet been found. Extensive studies have failed to elucidate the cause of the stimulating quality of country air, qualities which are lost when such air is brought indoors and particularly when it is handled by mechanical means. Ultra-violet light and ionization have been suggested but the evidence so far is inconclusive or negative[47].

NATURAL AND MECHANICAL VENTILATION

Under favorable conditions natural ventilation methods properly combined with means for heating may be sufficient to provide for the foregoing objectives in homes, uncrowded offices, small stores, etc.

In large offices, large school rooms, and in public and industrial buildings, natural ventilation is uncertain and makes heating difficult. The chief disadvantage of natural methods is the lack of control; they depend largely on weather and upon the velocity and direction of the wind. Rooms on the windward side of a building may be difficult to heat and ventilate on account of drafts, while rooms on the leeward side may not

[47]Changes in Ionic Content in Occupied Rooms, Ventilated by Natural and Mechanical Methods, by C. P Yaglou, L. C Benjamin and S. P Choate (A S H V E Transactions, Vol 37, 1931) Physiologic Changes During Exposure to Ionized Air, by C P Yaglou, A D Brandt and L C. Benjamin (A S H V E Transactions, Vol 39, 1933) Diurnal and Seasonal Variations in the Small Ion Content of Outdoor and Indoor Air, by C P Yaglou and L C. Benjamin (A S H V E Transactions, Vol 40, 1934) The Nature of Ions in Air and Their Possible Physiological Effects, by L. B. Loeb (A S H V E Transactions, Vol 40, 1934). The Influence of Ionized Air upon Normal Subjects, by L. P Herrington (*Journal Clinical Investigation, 14*, January, 1935) The Effect of High Concentrations of Light Negative Atmospheric Ions on the Growth and Activity of the Albino Rat, by L P. Herrington and Karl L Smith (*Journal Ind Hygiene, 17*, November, 1935) Subjective Reactions of Human Beings to Certain Outdoor Atmospheric Conditions, by C.-E. A Winslow and L P Herrington (A S H V E Journal Section, *Heating, Piping and Air Conditioning*, November, 1935)

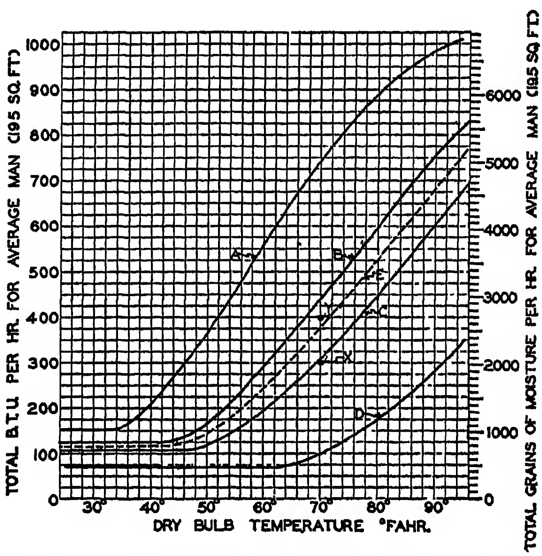


FIG. 10. LATENT HEAT AND MOISTURE LOSS FROM THE HUMAN BODY BY EVAPORATION, IN RELATION TO DRY-BULB TEMPERATURE FOR STILL AIR CONDITIONS[a]

[a]Curve *A*—Men working 66,150 ft-lb per hour. Curve *B*—Men working 33,075 ft-lb per hour. Curve *C*—Men working 16,538 ft-lb per hour. Curve *D*—Men seated at rest. Curves *A* and *C* drawn from data at a dry-bulb temperature of 81.3 F only and extrapolating the relation between Curves *B* and *D* which were drawn from data at many temperatures

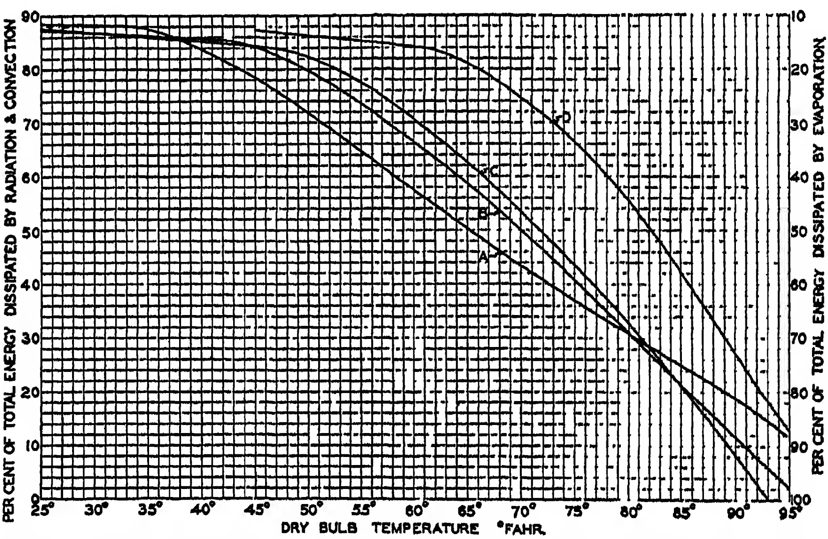


FIG. 11. HEAT LOSS FROM THE HUMAN BODY BY EVAPORATION, RADIATION AND CONVECTION IN RELATION TO DRY-BULB TEMPERATURE FOR STILL AIR CONDITIONS[a]

[a]Curve *A*—Men working 66,150 ft-lb per hour. Curve *B*—Men working 33,075 ft-lb per hour. Curve *C*—Men working 16,538 ft-lb per hour. Curve *D*—Men seated at rest. Curves *A* and *C* drawn from data at a dry-bulb temperature of 81.3 F only and extrapolating the relation between Curves *B* and *D* which were drawn from data at many temperatures

receive an adequate amount of air from out-of-doors. The partial vacuum produced on the leeward side under the action of the wind may even reverse the flow of air so that the leeward half of the building has to take the *drift* of the air from the rooms of the windward half. Under such conditions no outdoor air would enter through a leeward window opening, but room air would pass out.

In warm weather natural methods of ventilation afford little or no control of indoor temperature and humidity. Outdoor smoke, dust and noise constitute other limitations of natural methods.

A.S.H.V.E. VENTILATION STANDARDS†

As Adopted in August, 1932

It is the intent of the Committee in presenting this report to confine itself to a statement of those requirements which, based on present day knowledge (1932), will provide adequate ventilation for spaces intended for human occupancy The following standards shall apply to all spaces occupied by human beings in all buildings for which ventilation regulations are to be established

SECTION I—AIR TEMPERATURE AND HUMIDITY

The *temperature* and *humidity* of the air in such occupied spaces, and in which the only source of contamination is the occupant, shall be maintained at all times during occupancy at an *Effective Temperature*, as hereinafter stated.

The relative humidity shall be not less than 30 per cent, nor more than 60 per cent in any case The Effective Temperature shall range between 64 deg and 69 deg when heating or humidification is required, and between 69 deg and 73 deg when cooling or dehumidification is required

These Effective Temperatures shall be maintained at a level of 36 in. above the floor. (See Appendix, Tables A and B).

SECTION II—AIR QUALITY

The air in such occupied spaces shall at all times be free from toxic, unhealthful or disagreeable gases and fumes and shall be relatively free from odors and dust.

In every space coming within the provisions of these requirements and in which the quality of the air is below the standards prescribed by good medical and engineering practices, due to toxic substances, bacteria, dust, excessive temperature, excessive humidity, objectionable odors, or other similar causes, means for ventilating shall be provided so that the quality of the air shall be raised to these standards

SECTION III—AIR MOTION

The air in such occupied spaces shall at all times be in constant motion sufficient to maintain a reasonable uniformity of temperature and humidity, but not such as to cause objectionable drafts in any occupied portion of such spaces

The air motion in such occupied spaces, and in which the only source of contamination is the occupant, shall have a velocity of not more than 50 ft per minute, measured at a height of 36 in. above the floor.

SECTION IV—AIR DISTRIBUTION

The air in all rooms and enclosed spaces shall, under the provisions of these requirements, be distributed with reasonable uniformity, and the variation in the carbon dioxide content of the air shall be taken as a measure of such distribution.

The air in a space ventilated in accordance with these requirements, and in which the only source of contamination is the occupant, shall be distributed and circulated so that the variation in the concentration of carbon dioxide, when measured at a height of 36 in. above the floor, shall not exceed one part in 10,000

†Report of A S H V.E Committee on Ventilation Standards consisting of W H Driscoll, *Chairman*, J. J. Aeberly, F Paul Anderson, L A. Harding, D D Kimball, J R McColl, C L Riley, W A Rowe, Perry West and A. C Willard, presented at the Semi-Annual Meeting of the Society, Milwaukee, Wis, June, 1932, and adopted by the Society in August, 1932 (See A S H V E TRANSACTIONS, Vol 38, 1932)

SECTION V—AIR QUANTITY

The quantity of air used to ventilate the given space during occupancy shall always be sufficient to maintain the standards of air temperature, air quality, air motion and air distribution as herein required. Not less than 10 cu ft per minute per occupant of the total air circulated to meet these requirements shall be taken from an outdoor source

APPENDIX

TABLE A EFFECTIVE TEMPERATURES RANGING FROM 64 DEG TO 69 DEG FOR VARIOUS DRY-BULB TEMPERATURES AND RELATIVE HUMIDITIES FOR STILL AIR FOR PERSONS NORMALLY CLOTHED AND SLIGHTLY ACTIVE[a]

(For use when heating or humidification is required)

DRY-BULB TEMPERATURES (DEG FAHR)	RELATIVE HUMIDITIES (PER CENT)						
	30	35	40	45	50	55	60
	EFFECTIVE TEMPERATURES (DEGREES)						
67						64.0	64.3
68			64.0	64.2	64.5	64.8	65.1
69	64.1	64.4	64.8	65.1	65.4	65.7	66.0
70	64.8	65.1	65.4	65.8	66.2	66.5	66.8
71	65.5	65.8	66.2	66.6	67.0	67.3	67.7
72	66.2	66.5	66.9	67.3	67.7	68.1	68.5
73	67.0	67.3	67.7	68.1	68.5	68.9	
74	67.7	68.0	68.4	68.8			
75	68.4	68.7					
76	69.0						

[a]See Fig 6.

TABLE B EFFECTIVE TEMPERATURES RANGING FROM 69 DEG TO 73 DEG FOR VARIOUS DRY-BULB TEMPERATURES AND RELATIVE HUMIDITIES FOR STILL AIR FOR PERSONS NORMALLY CLOTHED AND SLIGHTLY ACTIVE[a-b]

(For use when cooling or dehumidification is required)

DRY-BULB TEMPERATURES (DEG FAHR)	RELATIVE HUMIDITIES (PER CENT)						
	30	35	40	45	50	55	60
	EFFECTIVE TEMPERATURES (DEGREES)						
73							69.3
74					69.3	69.7	70.1
75			69.1	69.5	70.0	71.5	71.0
76	69.0	69.4	69.9	70.5	70.8	71.3	71.8
77	69.7	70.2	70.7	71.2	71.6	72.1	72.6
78	70.4	70.9	71.4	71.9	72.4	73.0	
79	71.1	71.6	72.2	72.6			
80	71.8	72.4	72.9				
81	72.5						

[a]See Fig. 6.

[b]This table applies primarily to cases in which the human body has reached equilibrium with the surrounding air A higher plane of summer effective temperatures is required in places of public assembly where the period of occupancy is short, than is required for offices and industrial plants where the period of occupancy is of longer duration When the period of occupancy is two hours or less, the dry-bulb temperature shall be 72 F plus one-third of the difference between the outside dry-bulb temperature and 70 F, and the relative humidity shall not exceed 60 per cent (See also Table 2.)

Definitions

For the purposes of these standards the terms used shall be defined as follows:

Ventilation: The process of supplying or removing air by natural or mechanical means, to or from any space Such air may or may not have been conditioned. (See *Air Conditioning*).

Air Conditioning: The simultaneous control of all or at least the first three of those factors affecting both the physical and chemical conditions of the atmosphere within any structure. These factors include temperature, humidity, motion, distribution, dust, bacteria, odors, toxic gases, and ionization, most of which affect in greater or lesser degree human health or comfort.

Dry-Bulb Temperature The temperature of the air which is indicated by any type of thermometer which is not affected by the water vapor content or relative humidity of the air

Dust Solid material in a finely divided state, the particles of which are large and heavy enough to fall with increasing velocity, due to gravity in still air For instance, particles of fine sand or grit, such as are blown on a windy day, the average diameter of which is approximately 0 01 centimeter, may be called dust

Effective Temperature· An arbitrary index of the degree of warmth or cold felt by the human body in response to temperature, humidity, and movement of the air Effective temperature is a composite index which combines the readings of temperature, humidity, and air motion into a single value. The numerical value of the effective temperature scale has been fixed by the temperature of saturated air which induces an identical sensation of warmth

Humidity The water vapor (either saturated or superheated steam) occupying any space, which may or may not contain other vapors and gases at the same time

Relative Humidity A ratio, although usually expressed in per cent, used to indicate the degree of saturation existing in any given space resulting from the water vapor present in that space The presence of air or other gases in the same space at the same time has nothing to do with the relative humidity of the space, which depends merely on the temperature and partial pressure of the vapor

Spaces in Which the Only Source of Contamination is the Occupant. Spaces in which the atmospheric contamination results entirely from the respiratory processes of the occupant, including heat, moisture, and odors given off by the body. No manufacturing or industrial processes or other sources of atmospheric contamination, including heat and moisture, than people are considered under this title

RECIRCULATION AND OZONE

The amount of recirculated air may be varied to suit changes in weather and seasonal requirements, so as to conserve heat in winter and refrigeration in summer, but the saving in operating cost should not be obtained at the expense of air quality.

Ozone has been used for deodorizing recirculated air by oxidation or *masking*. Under favorable conditions some success is possible but from the practical standpoint it is difficult to regulate the ozone output so as to just neutralize undesirable odors at all times during the occupancy of a room. The difficulties appear to be mainly due to a wide variability in the rate of ozone disappearance in different rooms, or in the same room at different times, according to the characteristics of a room, the absolute humidity, impurities in the air, number and type of occupants, and probably other factors which require considerable study before ozone can be safely and economically applied.

The allowable concentrations in the breathing zone are very small, between 0.01 to 0.05 parts of O_3 per million parts of air. These are much too small to influence bacteria. Higher concentrations are associated with a pungent unpleasant odor and considerable discomfort to the occupants. One part per million causes respiratory discomfort in man, headaches and depression, lowers the metabolism, and may even lead to coma[48].

Toilets, kitchens, and similar rooms, in buildings using recirculation, should be ventilated separately by mechanical exhaust in order to prevent objectionable odors from diffusing into other parts of the building.

PROBLEMS IN PRACTICE

1 ● What is the purpose of conditioning the air of occupied rooms?

Chiefly comfort, to be secured by control of temperature, humidity, air distribution, and body odors in the air Other factors have yet to be studied.

2 ● What are the most comfortable air conditions?

Comfort standards are not absolute, but they are greatly affected by the physical condition of the individual, and the climate, season, age, sex, clothing, and physical activity. For the northeastern climate of the United States, the conditions which meet the require-

[48]*The British Medical Journal*, Editorial, June 25, 1932, p. 1182. See also Loc. Cit. Note 5

ments of the majority of people consist of temperatures between 68 and 72 F in winter and between 70 and 85 F in summer, the latter depending largely upon the prevailing outdoor temperature. The most desirable relative humidity range seems to be between 30 and 60 per cent.

3 ● Are the optimum conditions for comfort identical with those for health?

There are no absolute criteria of the prolonged effects of various air conditions on health For the present it can be only inferred that bodily discomfort may be an indication of adverse conditions leading to poor health

4 ● Given dry-bulb and wet-bulb temperatures of 76 F and 62 F, respectively, and an air velocity of 100 fpm, determine: (1) effective temperature of the condition; (2) effective temperature with calm air; (3) cooling produced by the movement of the air.

(1) In Fig. 1 draw line *AB* through given dry- and wet-bulb temperatures. Its intersection with the 100 ft velocity curve gives 69 deg for the effective temperature of the condition. (2) Follow line *AB* to the right to its intersection with the 20 fpm velocity line, and read 70 4 deg for the effective temperature for this velocity or so-called still air. (3) The cooling produced by the movement of the air is 70 4 − 69 = 1 4 deg ET.

5 ● Assume that the design of an air conditioning system for a theater is to be based on an outdoor dry-bulb temperature of 95 F and a wet-bulb temperature of 78 F with an indoor relative humidity of 50 per cent. According to Table 2, the dry-bulb temperature in the auditorium should be 80 F. Estimate the sensible and latent heat given up per person.

The sensible heat given up per person per hour may be obtained from Fig. 9. With an abscissa value of 80 F, Curve *D* for men seated at rest gives a value (on the ordinate scale) of 220 Btu per person per hour as the sensible heat loss The latent heat given up by a person seated at rest may be obtained from Fig. 10. With an abscissa value of 80 F, Curve *D* indicates a latent heat loss of 175 Btu per hour (left hand scale) or a moisture loss of 1190 grains per hour (right hand scale).

6 ● Neglecting the gain or loss of heat by transmission or infiltration through walls, windows and doors, how many cubic feet of outside air, with dry- and wet-bulb temperatures of 65 F and 59 F, respectively, (63.1 deg ET) must be supplied per hour to an auditorium containing 1000 people in order that the inside temperature shall not exceed 75 F dry-bulb and 65 F wet-bulb?

Figs. 9 and 10 give 265 Btu sensible heat and 905 grains of moisture per person with a dry-bulb temperature of 75 F in the auditorium. Therefore, 265,000 Btu of sensible heat and 905,000 grains of moisture will be added to the air in the auditorium per hour.

Taking 0.24 as the specific heat of air, 2.4 Btu per pound of air will be absorbed in raising the dry-bulb temperature from 65 to 75 F, and $\frac{265,000}{2.4} = 110,400$ lb of air or $110,400 \times 13.4 = 1,479,000$ cfh of air will be required. This is equivalent to $\frac{1,479,000}{1000 \times 60}$ = 24 7 cfm per person.

The moisture content of the inside air is 76 grains per pound of dry air and that of the outside condition is 65 grains. From a psychrometric chart the increase in moisture content will therefore be 11 grains per pound of dry air. Hence $\frac{905,000}{11.0} = 82,300$ lb of air at the specified condition will be required. This is equivalent to $82,300 \times 13.4 = 1,103,000$ cfh of air or $\frac{1,103,000}{1000 \times 60} = 18.4$ cfm of air per person.

The higher volume of 24 7 cfm per person will be required to keep the dry-bulb temperature from rising above the 75 F specified The wet-bulb temperature will therefore not rise to the maximum of 65 F.

Chapter 4

NATURAL VENTILATION

Wind Forces, Stack Effect, Openings, Windows, Doors, Skylights, Roof Ventilators, Stacks, Principles of Control, General Rules, Measurements, Dairy Barn Ventilation, Garage Ventilation

VENTILATION by natural forces, supplemented in certain cases with mechanical forces, finds extensive application in industrial plants, public buildings, schools, dwellings, garages, and in farm buildings.

The natural forces available for the displacement of air in buildings are the wind and the difference in temperature of the air inside and outside the building. The arrangement and control of ventilating openings should be such that the two forces act cooperatively and not in opposition.

Wind Forces

In considering the use of natural wind forces for the operation of a ventilating system, account must be taken of (1) average and minimum wind velocities, (2) wind direction, (3) seasonal, daily and hourly variations in wind velocity and direction, and (4) local wind interference by buildings and trees.

Table 1, Chapter 8, gives values for the average summer wind velocities and the prevailing wind directions in various localities throughout the United States, while Table 2, Chapter 7, lists similar values for the winter. In almost all localities the summer wind velocities are lower than those in the winter, and in about two-thirds of the localities the prevailing direction is different during the summer and winter. While average wind velocities are seldom below 5 mph, there are many hours in each month during which the wind velocity is from 3 to 5 mph, even in localities where the seasonal average is considerably above 5 mph. There are relatively few places where the hourly wind velocity falls much below 3 mph for more than 10 daylight hours per month. Usually a natural ventilating system should be designed to operate satisfactorily with a wind velocity of 3 to 6 mph, depending on locality.

The following formula may be used for calculating the quantity of air forced through ventilation openings by the wind, or for determining the proper size of such openings:

$$Q = EAV \tag{1}$$

where

Q = air flow in cubic feet per minute.
A = free area of inlet (or outlet) openings in square feet.
V = wind velocity in feet per minute,
= miles per hour $\times$ 88
E = effectiveness of openings.

(E should be taken at from 50 to 60 per cent if the inlet openings face the wind and from 25 to 35 per cent if the inlet openings receive the wind at an angle.)

If outlet openings, where air leaves a building, are smaller than inlet openings, where air enters a building, the air will be less effective than indicated by the constant E.

The accuracy of the results obtained by the use of Formula 1 depends upon the placing of the openings, as the formula assumes that ventilating openings have a flow coefficient slightly greater than that of a square-edge orifice. If the openings are not advantageously placed with respect to the wind, the flow per unit area of the openings will be less, and if unusually well placed, the flow will be slightly more than that given by the formula. Inlets should be placed to face directly into the prevailing wind, while outlets should be placed in one of the following four places

1. On the side of the building directly opposite the direction of the prevailing wind
2. On the roof in the low pressure area caused by the jump of the wind (see Fig 1).
3. In a monitor on the side opposite from the wind.
4. In roof ventilators or stacks exposed to the full force of the wind[1].

Forces Due to Stack Effect [2]

The stack effect produced within a building is due to the difference in weight of the warm column of air within the building and the cooler air outside. The flow due to stack effect is proportional to the square root of the draft head, or approximately:

$$Q = 9.4\, A \sqrt{H\,(t - t_o)} \tag{2}$$

where

Q = air flow in cubic feet per minute.
A = free area of inlets or outlets (assumed equal) in square feet.
H = height from inlets to outlets, in feet.
t = *average* temperature of indoor air in height H, in degrees Fahrenheit
t_o = temperature of outdoor air, in degrees Fahrenheit.
9.4 = constant of proportionality, including a value of 65 per cent for effectiveness of openings. This should be reduced to 50 per cent (constant = 7.2) if conditions are not favorable.

The height between inlets and outlets should be the maximum which the building construction will allow.

In some cases the necessary air flow will be known from the requirements of the building occupancy, and the area necessary for certain assumed temperature differences may be calculated. Or the areas may be fixed by the building construction, and the maximum air flow for various differences between indoor and outdoor temperatures may be calculated. In any case, the conditions which give the minimum air flow are those which control the design, as the system must have ample capacity even under the most unfavorable conditions which are those of mild or warm weather.

TYPES OF OPENINGS

The engineering problems of a natural ventilation system consist of the *design, location, and control of ventilating openings* to best utilize the

[1]Airation of Industrial Buildings, by W. C. Randall (A S H V.E. Transactions, Vol. 34, 1928)
[2]Neutral Zone in Ventilation, by J E Emswiler (A S H.V E. Transactions, Vol 32, 1926).
Predetermining Airation of Industrial Buildings, by W C. Randall and E. W. Conover (A.S.H.V.E. Transactions, Vol 37, 1931).

natural ventilation forces, in accordance with the requirements of building occupancy. The types of openings may be classified as:

1. Windows, doors, monitor openings, and skylights
2. Roof ventilators.
3. Stacks connecting to registers
4. Specially designed inlet or outlet openings

Windows, Doors and Skylights

Windows have the advantage of transmitting light, as well as providing ventilating area when open. Their movable parts are arranged to open in

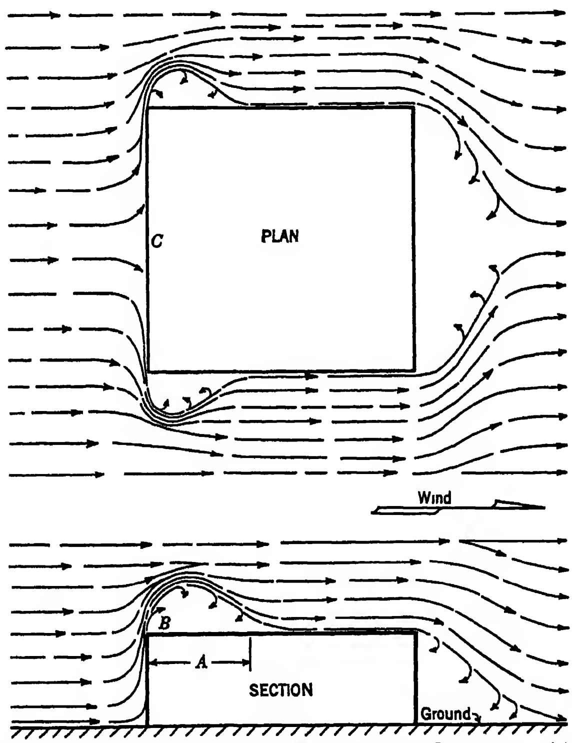


Fig. 1. The Jump of Wind from Windward Face of Building. (*A*—Length of Suction Area; *B*—Point of Maximum Intensity of Suction; *C*—Point of Maximum Pressure)

various ways; they may open by sliding as in the ordinary double-hung windows, by tilting on horizontal pivots at or near the center, or by swinging on pivots at the top or bottom. Whatever the form and type of window used, the amount of clear area that can be made available is the factor of greatest importance in ventilation.

All types of sash (double-hung, top, center or bottom horizontal pivoted, or vertical pivoted) have about the same air flow capacity for the same clear area. Air leakage through *closed* windows is important during high winds (Chapter 6).

The proper distribution of air in occupied spaces is an element almost as important as that of sufficient air quantity. Advantageous pivoting of sash is very useful for securing good air distribution. Deflectors are sometimes used for the same purpose, and these devices should be considered a part of the ventilation system.

Door openings are seldom included in the ventilation calculations, though they may be of great value for extreme summer conditions, and should be considered in this connection as well as in garage design.

Skylight and monitor openings are of importance as these and the roof ventilators are outlets, while the lower windows are usually inlets on the windward side and outlets on the leeward side. In general the areas of inlets and outlets should be about equal. It is important to make a check on this ratio in any installation, as any great excess of area of one set of openings over another means waste opening area. The operating devices used for sash, monitors, skylights and roof ventilators should be well selected as poor operating devices may defeat the entire design.

Roof Ventilators

The function of a roof ventilator is to provide a storm and weather proof air outlet, which is sensitive to wind action for producing additional flow capacity, and at the same time is subject to manual or automatic control by suitable dampers. The capacity of a ventilator at a constant wind velocity and temperature difference, depends upon four things: (1) its location on the roof, (2) the resistance it offers to air flow, (3) the area and location of openings provided for air inflow at a lower level, and (4) the ability of the ventilator head to utilize the kinetic energy of the wind for inducing flow by centrifugal or ejector action. Frequently one or more of these capacity factors is overlooked in a ventilator installation.

For maximum flow induction, a ventilator should be located on that part of the roof which receives the full wind without interference. (See Fig. 1.) This does not mean that no ventilators are to be installed within the suction region created by the wind jumping over the building, or in a light court, or on a low building between two high buildings. Ventilators are highly effective in such low-pressure areas, but their ejector action, caused by wind velocity, is of little importance in these locations, and hence their size should be increased proportionally.

Ventilator resistance depends on (1) type of inlet, (2) area of openings and passages, and (3) number of turns or changes of direction of the air flow. The inlet grille, if any, should have ample free area, and the ventilator should always be provided with a taper-cone inlet in order to produce the effect of a bell-mouth nozzle (flow coefficient 0.97) rather than that of a square-entrance orifice (flow coefficient 0.60). In other words, the grilles should be oversize as compared with the ventilator, and they should be connected by tapering collars. If the ventilator head construction produces changes in the direction of air flow, the area of the flow passages should be increased accordingly.

Air inlet openings at lower levels in the building are of course necessary for the economical use of ventilator capacity. The inlet openings should be at least equal to, and preferably twice as great as the combined throat areas of all roof ventilators. The air discharged by a roof ventilator

depends on wind velocity and temperature difference, but due to the four capacity factors already mentioned, no simple formula can be devised for expressing ventilator capacity.

Several types of roof ventilators are shown in Figs. 2 to 11. These may be classified as *stationary*, Figs. 2 to 6, *pivoted* or *oscillating*, Figs. 7 to 9, or *rotating*, Figs. 10 and 11. When selecting roof ventilators, some attention should be paid to ruggedness of construction, storm-proofing features, dampers and damper operating mechanisms, possibilities of noise from dampers or other moving parts, and possible maintenance costs.

It should be kept in mind that a suitable combination of roof ventilators with mechanical ventilation frequently offers the best solution of a ventilating problem. The natural ventilation units may be used to supplement power driven supply fans, and under favorable weather conditions it may be possible to shut down the power driven units. Where low operating costs are very important, such a combination has great advantages. Roof ventilators with built-in electric fans are attracting increased attention because they combine the advantages of low installation and operating cost with those of continuous service.

Controls

In connection with any combination between natural and fan ventilation, the controls are of importance. Both the fans and the ventilator dampers may be controlled by some combination of three methods: (1) hand operation, (2) thermostat operation, and (3) control by wind velocity. The thermostat station may be located anywhere in the building, or it may be located within the ventilator itself The purpose of wind velocity control is to obtain a definite volume of exhaust regardless of the natural forces, the fan motor being energized when the natural exhaust capacity falls below a certain minimum, and again shut off when the wind velocity rises to the point where this minimum volume can be supplied by natural forces.

Stacks

Stacks are really chimneys and utilize both the inductive effect of the wind and the force of temperature difference (the so-called *gravity* action). While their openings projecting above the roof are not provided with any special construction for developing suction by the action of the wind, the plain vertical opening is also effective in this respect. Like the roof ventilator, the stack outlet should be located so that the wind may act upon it from any direction.

Stacks are applicable particularly in the case of schools, apartments, residences and small office buildings. Partitions interfere with general air circulation, and some type of outlet from each room is necessary. If the building is not too tall, and the requirements of occupancy are moderate, a system of stacks with registers in each room may be more economical than a system of mechanical ventilation employing fans. In making the comparison, however, the building space occupied by the stacks should be considered.

With little or no wind, chimney effect or temperature difference will produce outflow through the stacks and an equal inflow through windows

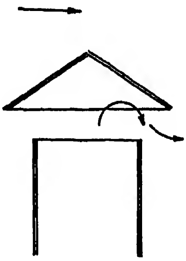


Fig. 2

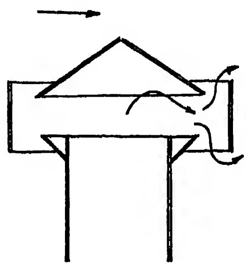


Fig. 3

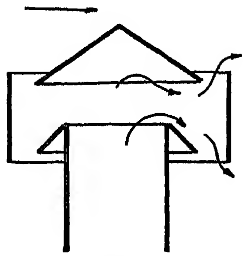


Fig. 4

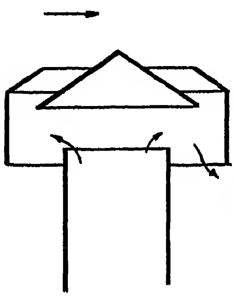


Fig. 5

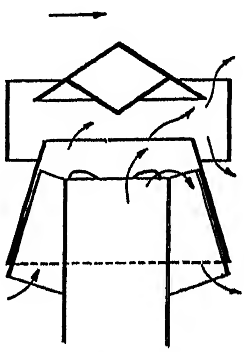


Fig. 6

Six Common Types of Stationary Ventilators

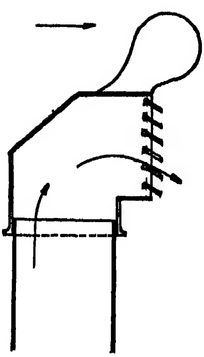


Fig. 7

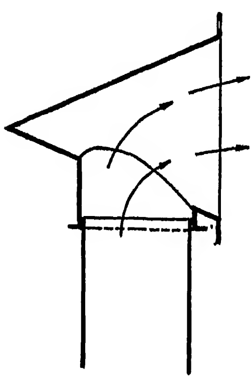


Fig. 8

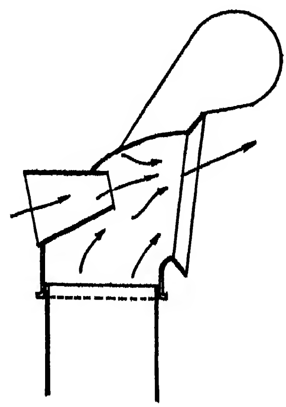


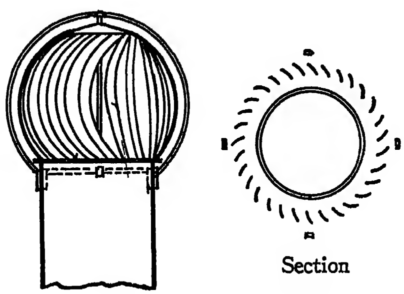
Fig. 9

Three Typical Oscillating Ventilators

in all sides of the building. With wind, the inductive force at the top of ventilating shafts is more powerful than that on the leeward side of the building, so that air is drawn in through leeward openings by a combination of the forces of wind and temperature difference. On the windward side, the direct forcing pressure of the wind is of course added to the temperature difference effect. Thus forces are available for causing inflow at practically every window of such a building. Adequacy of stack size must, of course, be provided.

PRINCIPLES OF AIR FLOW CONTROL

The air flow through a ventilation opening depends on the two factors already discussed, namely, (1) the natural forces available, (2) the openings available, and the resistance to flow offered by these openings. The design problem includes, of course, a determination of the *desired air*



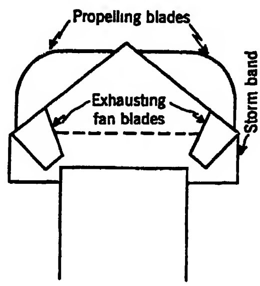


FIG. 10. ROTATING VENTILATORS FIG. 11

quantity and distribution in order that the openings may be properly placed.

The purpose of ventilation is to carry off either excess heat or air impurities, and the desired air quantities depend upon the amount of heat or of impurities present. The amount of heat can be determined, in the case of forge shops for example, from the amount of fuel burned, which in turn is based upon the production capacity for which the building is being designed. In the case of foundries, the heat given off by the metal in cooling from the molten state can be used. In some instances, not all of the heat may be dissipated to the air, but a fair estimate of the amount to be removed by the air can usually be made.

The next step is to select the temperature difference to be maintained. Knowing the amount of heat to be removed and having selected a desirable temperature difference, the amount of air to be passed through the building per minute to maintain this temperature difference can be determined by means of the following equation:

$$H = \frac{c\,60\,Q\,(t - t_0)}{V} \tag{3}$$

where

$c = 0.24$ = specific heat of air.

V = specific volume of the air, cubic feet per pound, about 13.5. (See Chapter 1.)

H = heat to be carried off, in Btu per hour
Q = air flow in cubic feet per minute.
t = inside temperature, degrees Fahrenheit
t_0 = outside temperature, degrees Fahrenheit.

For disposing of air impurities, the required air flow must be such that the outside air will dilute the impurities to a degree that they are no longer objectionable. For human occupancy, such as in auditoriums and classrooms, 10 cfm per person is usually taken as the minimum of outside air necessary for ventilation (see Chapter 3). For garage ventilation, sufficient air must be admitted to dilute the carbon monoxide content of the indoor air to 1 in 10,000 (see Garage Ventilation in this Chapter).

Air *quantity* and *quality* are not the only requirements. For human occupancy, air *distribution* is important. In ventilation the air distribution is almost entirely a matter of the number, the design, and the location of inlets and outlets. In locating openings, special precautions should be taken against the formation of dead air spaces or *pockets* within the zone of occupancy (see Chapter 19).

Suggested methods for estimating the air flow due to temperature difference alone and to wind alone have already been given. It must be remembered that when both forces are acting together, even without interference, the resulting air flow is not equal to the sum of the two estimated quantities. The same openings have been assumed in both cases, and since the resistance to flow through the openings varies approximately with the square of the velocity[3], this resistance becomes a limiting factor as the flow through the openings is increased.

Recent investigations[4] show that the total flow is only 10 per cent above the flow caused by the greater force when the two forces are nearly equal, and this percentage decreases rapidly as one force increases above the other. Tests on roof ventilators indicate that this is too conservative in the direction of low total flow quantities, but there is in any case a large judgment factor involved. The wind velocity and direction, the outdoor temperature, or the indoor activities cannot be predicted with certainty, and great refinement in calculations is therefore not justified. When designing for winter conditions, an added variable is the heat lost by direct flow through walls and windows and by infiltration.

Example 1 Assume a drop forge shop, 200 ft long, 100 ft wide, and 30 ft high. The cubical content is 600,000 cu ft, and the height of the air outlet over that of the inlet is 30 ft. Oil fuel of 18,000 Btu per lb is used in this shop at the rate of 15 gal per hour (7.75 lb per gal) Temperature differences are 10 F in summer and 30 F in winter, and the wind velocity is 5 mph in summer and 8 mph in winter. What is the necessary area for the inlets and outlets, and what is the rate of air flow through the building?

Solution. The system must be designed for the summer conditions as these are the more severe. The heat to be removed per hour is:

$$H = 15 \times 7.75 \times 18{,}000 = 2{,}092{,}500 \text{ Btu.}$$

By Equation 3, the air flow required to remove this heat with a temperature difference of 10 deg is:

$$Q = \frac{VH}{c\,60\,(t - t_0)} = \frac{13.5 \times 2{,}092{,}500}{0.24 \times 60 \times 10} = 196{,}172 \text{ cfm.}$$

[3]Loc Cit Notes 1 and 2

[4]This is true for *turbulent* flow only It would be more correct to state that the resistance varies approximately with V^2 for high to moderate velocities, with $V^{1.8}$ for moderate to low velocities, and with the first power of the velocity for very low velocities through small openings.

This is equal to 19 6 air changes per hour. The assumption is made that the average temperature difference between indoors and outdoors is the same as the temperature rise of the air from the inlet opening to the outlet opening Actually, the latter difference is larger and so the value of 19 6 air changes per hour is conservative as it allows for more cooling than is necessary for an *average* temperature difference of 10 deg.

If 196,172 cfm are to be circulated by the force of the temperature difference alone, the area of opening would be, by Equation 2

$$A = \frac{Q}{9\,4\,\sqrt{H\,(t - t_o)}} = \frac{196{,}172}{9.4\,\sqrt{30 \times 10}} = 1{,}205 \text{ sq ft}$$

If this area of openings were provided, a wind velocity of 5 mph, acting alone, would produce a flow according to Equation 1, of:

$$Q = EAV = 0\,50 \times 1{,}205 \times 5 \times 88 = 265{,}100 \text{ cfm}$$

If the inlet openings do not face the wind, but are at an angle with it, about half this amount may be considered to flow.

A factor of judgment must now be exercised in making the selection of the area of openings to be specified Apparently 1205 sq ft are a very generous allowance because either a direct wind of 5 mph or an average temperature difference of 10 deg acting alone will more than suffice to carry away the heat, and when the two forces are acting together, the system may have an excess capacity of 25 per cent to 50 per cent, especially if the outlets are made up partially of roof ventilators which employ the force of the wind for producing a suction effect. On the other hand, the wind may at times come from an unfavorable direction, or its velocity may fall below 5 mph or the building construction may not permit a full 2400 sq ft of inlet window area and an equal amount of monitor or roof ventilator outlet area. In case the two sets of openings are not equal, their effectiveness is reduced.

From this example, it must be apparent that while formulas may furnish a reliable guide, the final solution of a problem of natural ventilation requires a common sense analysis of local conditions to supplement and to modify the dictates of the formulas.

GENERAL RULES

A few of the important requirements in addition to those already outlined are:

1 Inlet openings should be well distributed, and should be located on the windward side near the bottom, while outlet openings are located on the leeward side near the top Outside air will then be supplied to the zone of occupancy.

2. Direct short circuits between openings on two sides at a high level may clear the air at that level without producing any appreciable ventilation at the level of occupancy.

3. Roof ventilators should be located 20 to 40 ft apart each way and preferably on the ridge of the roof. The closer spacings are used when ventilating rooms with low ceilings.

4. Greatest flow per square foot of total opening is obtained by using inlet and outlet openings of nearly equal areas.

5. In an industrial building where furnaces, that give off heat and fumes, are to be installed, it is better to locate them in the end of the building exposed to the prevailing wind. The strong suction effect of the wind at the roof near the windward end will then cooperate with temperature difference, to provide for the most active and satisfactory removal of the heat and gas laden air.

6. In case it is impossible to locate furnaces in the windward end, that part of the building in which they are to be located should be built higher than the rest, so that the wind, in splashing therefrom will create a suction. The additional height also increases the effect of temperature difference to cooperate with the wind.

7. In the use of monitors, windows on the windward side should usually be kept closed, since, if they are open, the inflow tendency of the wind counteracts the outflow tendency of temperature difference. Openings on the leeward side of the monitor result in cooperation of wind and temperature difference.

8 In order that the force of temperature difference may operate to maximum advantage, the vertical distance between inlet and outlet openings should be as great as possible. Openings in the vicinity of the neutral zone are less effective for ventilation.

9 In order that temperature difference may produce a motive force, there must be vertical distance between openings That is, if there are a number of openings available in a building, but all are at the same level, there will be no motive head produced by temperature difference, no matter how great that difference might be.

10. In the design of window ventilated buildings, where the direction of the wind is quite constant and dependable, the orientation of the building together with amount and grouping of ventilation openings can be readily arranged to take full advantage of the force of the wind. On the other hand, where the direction of the wind is quite variable, it may be stated as a general principle that windows should be arranged in sidewalls and monitors so that there will be approximately equal area on all sides. Thus, no matter what the wind's direction, there will always be some openings directly exposed to the pressure force of the wind, and others opposed to a suction force, and effective movement through the building will be assured.

11 The intensity of suction or the vacuum produced by the jump of the wind is greatest just back of the building face. The area of suction does not vary with the wind velocity, but the flow due to suction is directly proportional to wind velocity.

12. Openings much larger than the calculated areas are sometimes desirable, especially when changes in occupancy are possible, or to provide for extremely hot days. In the former case, free openings should be located at the level of occupancy for psychological reasons.

13. Special consideration should be given to the possibility of sidewall or monitor windows being closed on account of weather conditions. Such possibilities favor roof ventilators and specially designed stormproof inlets.

MEASUREMENT OF NATURAL AIR FLOW

The determination of the performance of any ventilating system involves measurements which are not easy to make. The difficulties are increased in the case of natural ventilation, since the motive forces and the air velocities are very small. The measurements necessary for giving the *capacity* of a system are (1) velocity of the wind, (2) velocity of the air through inlet and outlet openings, (3) outdoor air temperature, and (4) average indoor air temperature.

Measuring Wind Velocity. The cup-type of anemometer as used for Weather Bureau observations is sufficiently accurate for this measurement. Some more accurate instruments as well as direct-reading types have been developed for airport service, but for ventilation work it is the average wind velocity over a long period which determines the capacity of the system. Hence the use of the Weather Bureau instrument, with an observation period of one hour or more, is satisfactory. If observations of wind direction are required, these should be taken by observing a sensitive weather vane at frequent intervals (about every 5 minutes) during the same period.

Velocity of Air Through Openings. The vane type anemometer is the most practical instrument for this measurement.

Use a small (4 in.) low-speed anemometer, and correct all readings according to a recent calibration. Mount the anemometer in a strap iron clamp with a long handle for convenience. Divide each opening into 5 in. squares (by string or wire) and hold the anemometer in the center of each square for a definite period of from 15 to 30 seconds. Record the result of the traverse as soon as completed and start another one immediately. A series of traverses over a period of one hour, or the full period covered by the wind velocity observations with a fairly steady wind, may be considered a satisfactory test for that wind velocity. It is preferable to have an anemometer observer at each opening. If the opening is covered by a grille or register, use the proper correction factors (see Chapter 43).

Outdoor Temperature. It is easy to make an error of 1 to 5 deg in observing the outdoor air temperature. An accurate thermometer, calibrated in 1 deg divisions should be used. The thermometer should be mounted in the shade at about mid-height of the building and not too near the building wall or adjacent to an air outlet. The heat from a wall or roof which has been exposed to the sun is easily transmitted to a thermometer, with resulting high readings.

Average Indoor Temperature. It is important to note that the capacity of an opening (such as roof ventilator) does *not* depend on the difference in the temperatures measured adjacent to the opening. It depends rather on the difference between the *average* temperature of the column of air inside the building and that outside. Indoor temperatures should therefore be observed at various heights to secure a good average.

DAIRY BARN VENTILATION [5]

A successful barn ventilating system is one which continuously supplies the proper amount of air required by the stock, with proper distribution and without drafts, and one which removes the excessive heat, moisture, and odors, and maintains the air at a proper temperature, relative humidity, and degree of cleanliness.

Barn temperatures below freezing and above 80 F affect milk production. Milk producing stock should be kept in a barn temperature between 45 and 50 F. Dry stock, at reduced feeding, may be kept in a barn 5 to 10 deg higher. Calf barns are generally kept at 60 F, while hospital and maternity barns usually have a temperature of 60 F or somewhat higher.

The heat produced by a cow of an average weight of 1000 lb may be taken as 3000 Btu per hour. The average rate of moisture production by a cow giving 20 lb of milk per day is 15 lb of water per day, or 4375 grains per hour. To set a standard of permissible relative humidity for cow barns is difficult. For 45 F an average relative humidity of 80 per cent is satisfactory, with 85 per cent as a limit.

Where the barn volume is within the limit that can be heated by the stabled animals, the air supply need not be heated. The air should be

[5]Dairy Barn Ventilation, by F L Fairbanks (A S H V E TRANSACTIONS, Vol 34, 1928).

Cow Barn Ventilation, by Alfred J Offner (A S H V E TRANSACTIONS, Vol 39, 1933)

For additional information on this subject refer to *Technical Bulletin*, U S Department of Agriculture (1930), by M A R Kelley

supplied through or near the ceiling. It is better to have the exhaust openings near the floor as larger volumes of warm air are then held in the barn and there is better temperature control with less likelihood of sudden change in barn temperature.

If a cow weighs 1000 lb and produces 3000 Btu of heat per hour, and if a barn for the cow has 600 cu ft of air space with 130 sq ft of building exposure, one cow will require 2600 to 3550 cfh of ventilation, depending on the temperature zone in which the barn is located. The permissible heat losses through the structure, based on one cow and depending on the temperature zone, vary between 0.043 and 0.066 Btu per hour per cu ft of barn space, and 0.197 to 0.305 Btu per hour per sq ft of barn exposure.

GARAGE VENTILATION [6]

On account of the hazards resulting from carbon monoxide and other physiologically harmful or combustible gases or vapors in garages, the importance of proper ventilation of these buildings cannot be overemphasized. During the warm months of the year, garages are usually ventilated adequately because the doors and windows are kept open. As cold weather sets in, more and more of the ventilation openings are closed and consequently on extremely cold days the carbon monoxide concentration runs high.

Many garages can be satisfactorily ventilated by natural means particularly during the mild weather when doors and windows can be kept open. However, the A.S.H.V.E. Code for Heating and Ventilating Garages, adopted in 1929 and revised in 1935, states that natural ventilation may be employed for the ventilation of storage sections where it is practical to maintain open windows or other openings at all times. The code specifies that such openings shall be distributed as uniformly as possible in at least two outside walls, and that the total area of such openings shall be equivalent to at least 5 per cent of the floor area. The code further states that where it is impractical to operate such a system of natural ventilation, a mechanical system shall be used which shall provide for either the supply of 1 cu ft of air per minute from out-of-doors for each square foot of floor area, or for removing the same amount and discharging it to the outside as a means of flushing the garage.

Research

Research on garage ventilation undertaken by the A.S.H.V.E. Committee on Research at Washington University, St. Louis, Mo., and at the

[6]Code for Heating and Ventilating Garages (A.S.H.V.E. TRANSACTIONS, Vol. 35, 1929), (A.S.H.V.E. Reprint, January, 1935).

Airation Study of Garages, by W. C. Randall and L. W. Leonhard (A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930)

Carbon Monoxide Concentration in Garages, by A. S. Langsdorf and R. R. Tucker (A.S.H.V.E. TRANSACTIONS, Vol 36, 1930).

Carbon Monoxide Distribution in Relation to the Ventilation of an Underground Ramp Garage, by F. C. Houghten and Paul McDermott (A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932)

Carbon Monoxide Distribution in Relation to the Ventilation of a One-Floor Garage, by F. C. Houghten and Paul McDermott (A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932).

Carbon Monoxide Distribution in Relation to the Heating and Ventilation of a One-Floor Garage, by F. C. Houghten and Paul McDermott (A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933).

Carbon Monoxide Surveys of Two Garages, by A. H. Sluss, E. K. Campbell and Louis M. Farber (A.S.H.V.E. TRANSACTIONS, Vol. 40, 1934).

University of Kansas, Lawrence, Kans., in cooperation with the A.S.H. V.E. Research Laboratory, and at the A.S.H.V.E. Research Laboratory has resulted in authoritative papers on the subject.

Some of the conclusions from work at the Laboratory are listed below:

1 Upward ventilation results in a lower concentration of carbon monoxide at the breathing line and a lower temperature above the breathing line than does downward ventilation, for the same rate of carbon monoxide production, air change and the same temperature at the 30-in. level

2. A lower rate of air change and a smaller heating load are required with upward than with downward ventilation.

3. In the average case upward ventilation results in a lower concentration of carbon monoxide in the occupied portion of a garage than is had with complete mixing of the exhaust gases and the air supplied However, the variations in concentration from point to point, together with the possible failure of the advantages of upward ventilation to accrue, suggest the basing of garage ventilation on complete mixing and an air change sufficient to dilute the exhaust gases to the allowable concentration of carbon monoxide

4. The rate of carbon monoxide production by an idling car is shown to vary from 25 to 50 cfh, with an average rate of 35 cfh.

5. An air change of 350,000 cfh per idling car is required to keep the carbon monoxide concentration down to one part in 10,000 parts of air.

PROBLEMS IN PRACTICE

1 ● What factors may make the adoption of a system of ventilation depending upon wind movement inadvisable in new construction?

a. Variation in direction of wind.
b. Variation in wind velocity.
c Inability to clean incoming air.
d. Inability to control location, size and shape of buildings on adjacent property.
e. Unsatisfactory warming of incoming air during cold weather.

2 ● a. What factors are important in the location and control of ventilating openings?

b. What types of ventilating openings are best suited to a proper distribution of the air supplied?

a. The proper distribution of air as required by the occupants, and the best utilization of natural ventilating forces. The general rules on page 93 apply particularly to these factors.

b. Windows with swinging sash and openings with deflectors may be used to direct air to the points desired.

3 ● a. What is the best location for ventilating openings?

b. How are the sizes of ventilating openings determined for proper air supply?

a. Inlet openings should be low and facing the prevailing winds where possible. Outlet openings should be high and on the side opposite the prevailing winds
b. For simple openings use Formula 1:

$$Q = EAV$$

and for stacks use Formula 2:

$$Q = 9.4\,A\,\sqrt{H\,(t - t_o)}$$

The use of these formulae is illustrated in Example 1 of the text of this chapter Inlet and outlet areas should be approximately the same for best results

4 ● a. What are the advantages of roof ventilators?

b. How are proper sizes determined for roof ventilators?

a Roof ventilators offer the best utilization of the inductive force of the wind, and they may be very economically fitted with built-in fans to supply the necessary circulation when the force of the wind is not sufficient.

b Because of the many factors affecting the flow through roof ventilators no accurate formula can be given. It is usual practice to make the combined throat area of all roof ventilators between one-half area and full area of the air inlets as determined by Formula 1.

5 ● What methods of control are used in ventilating systems?

Hand control, control by a thermostat located in the ventilated space or in the ventilator, or wind velocity control designed to keep the air discharge constant regardless of wind velocity

6 ● How is the quantity of air required for a building determined?

Sufficient air must be supplied to carry away the heat and impurities generated within a building. The temperature rise and concentration of impurities in the exhaust air must be held within specified limits (See Example 2.)

7 ● What measurements are necessary to determine the capacity of a ventilating system?

Wind velocity and air velocities through openings, determined by suitable anemometers, outdoor air temperatures, measured by a shaded thermometer not near objects heated by the sun or near exhaust air openings, indoor air temperatures, measured at various heights to secure a good average

8 ● How much air must be supplied for dissipating the heat generated in a dairy barn housing 100 cows if the outside temperature is 20 F and the inside temperature is to be maintained at 45 F?

The total heat generated is 100 × 3000 = 300,000 Btu per hour Then from Formula 3,

$$Q = \frac{VH}{c\,60\,(t - t_o)}$$

$$= \frac{13.5 \times 300{,}000}{0.24 \times 60\,(45 - 20)}$$

$$= 11{,}250 \text{ cfm.}$$

This amount of air should also keep down humidity and odors.

9 ● a. What precaution is necessary in the ventilation of garages using natural ventilation?

b. How much window area is required for a garage with 50 x 100 sq ft floor area if natural ventilation is used?

a. The carbon monoxide content of the air should be kept below 1 part in 10,000 and windows should be kept open at all times

b. The window area should aggregate 5 per cent of the floor area.

0.05 × 50 × 100 = 250 sq ft of window area.

This area should be evenly distributed along two sides of the building.

Chapter 5

HEAT TRANSMISSION COEFFICIENTS AND TABLES

Methods of Heat Transfer, Coefficients, Conductivity of Homogeneous Materials, Surface Conductance Coefficients, Air Space Conductance, Practical Coefficients, Table of Conductivities and Conductances, Tables of Over-all Coefficients of Heat Transfer for Typical Building Construction, Combined Coefficients of Transmission

IN order to maintain comfortable living temperatures within a building it is necessary to supply heat at the same rate that it is lost from the building. The loss of heat occurs in two ways, by direct transmission through the various parts of the structure and by air leakage or filtration between the inside and outside of the building. The purpose of this chapter is to show methods of calculation and to give practical transmission coefficients which may be applied to various structures to determine the heat loss by direct transmission. The amount lost by air filtration is determined by different methods, as outlined in Chapter 6, and must be added to that lost by direct transmission to obtain the total heating plant requirements.

METHODS OF HEAT TRANSFER

Heat transmission between the air on the two sides of a structure takes place by three methods, namely, radiation, convection and conduction. In a simple wall built up of two layers of homogeneous materials separated to give an air space between them, heat will be received from the high temperature surface by radiation, convection and conduction. It will then be conducted through the homogeneous interior section by conduction and carried across to the opposite surface of the air space by radiation, conduction and convection. From here it will be carried by conduction through to the outer surface and leave the outer surface by radiation, convection and conduction. The process of heat transfer through a built-up wall section is complicated in theory, but in practice it is simplified by dividing a wall into its component parts and considering the transmission through each part separately. Thus the average wall may be divided into external surfaces, homogeneous materials and interior air spaces. Practical heat transmission coefficients may be derived which will give the total heat transferred by radiation, conduction and convection through any of these component parts and if the selection and method of applying these individual coefficients is thoroughly understood it is usually a comparatively simple matter to calculate the over-all heat transmission coefficient for any combination of materials.

HEAT TRANSFER COEFFICIENTS

The symbols representing the various coefficients of heat transmission and their definitions are as follows:

U = thermal transmittance or over-all coefficient of heat transmission, the amount of heat expressed in Btu transmitted in one hour per square foot of the wall, floor, roof or ceiling for a difference in temperature of 1 deg F between the air on the inside and that on the outside of the wall, floor, roof or ceiling

k = thermal conductivity, the amount of heat expressed in Btu transmitted in one hour through 1 sq ft of a homogeneous material 1 in. thick for a difference in temperature of 1 deg F between the two surfaces of the material. The conductivity of any material depends on the structure of the material and its density. Heavy or dense materials, the weight of which per cubic foot is high, usually transmit more heat than light or less dense materials, the weight of which per cubic foot is low.

C = thermal conductance, the amount of heat expressed in Btu transmitted in one hour through 1 sq ft of a non-homogeneous material for the thickness or type under consideration for a difference in temperature of 1 deg F between the two surfaces of the material Conductance is usually used to designate the heat transmitted through such heterogeneous materials as plaster board and hollow clay tile

f = film or surface conductance, the amount of heat expressed in Btu transmitted by radiation, conduction and convection from a surface to the air surrounding it, or vice versa, in one hour per square foot of the surface for a difference in temperature of 1 deg F between the surface and the surrounding air To differentiate between inside and outside wall (or floor, roof or ceiling) surfaces, f_i is used to designate the inside film or surface conductance and f_o the outside film or surface conductance.

a = thermal conductance of an air space, the amount of heat expressed in Btu transmitted by radiation, conduction and convection in one hour through an area of 1 sq ft of an air space for a temperature difference of 1 deg F. The conductance of an air space depends on the mean absolute temperature, the width, the position and the character of the materials enclosing it.

R = resistance or resistivity which is the reciprocal of transmission, conductance, or conductivity, *i.e.*:

$$\frac{1}{U} = \text{over-all or air-to-air resistance.}$$

$$\frac{1}{k} = \text{internal resistivity.}$$

$$\frac{1}{C} = \text{internal resistance.}$$

$$\frac{1}{f} = \text{film or surface resistance.}$$

$$\frac{1}{a} = \text{air-space resistance.}$$

As an example in the application of these coefficients assume a wall with over-all coefficient U. Then,

$$H = A\,U\,(t - t_o) \qquad (1)$$

where

H = Btu per hour transmitted through the material of the wall, glass, roof or floor.

A = area in square feet of wall, glass, roof, floor, or material, taken from building plans or actually measured (Use the net inside or heated surface dimensions in all cases.)

$t - t_o$ = temperature difference between inside and outside air, in which t must always be taken at the proper level. Note that t may not be the *breathing-line* temperature in all cases.

If the heat transfer between the air and the inside surface of the wall is being considered, then,

$$H = A f_1 (t - t_1) \tag{2}$$

where

f_1 = inside surface conductance

t and t_1 = the temperatures of the inside air and the inside surface of the wall respectively.

In practice it is usually the over-all heat transmission coefficient that is required. This may be determined by a test of the complete wall, or it may be obtained from the individual coefficients by calculation. The simplest method of combining the coefficients for the individual parts of the wall is to use the reciprocals of the coefficients and treat them as resistance units. The total over-all resistance of a wall is equal numerically to the sum of the resistances of the various parts, and the reciprocal of the over-all resistance is likewise the over-all heat transmission coefficient of the wall. For a wall built up of a single homogeneous material of conductivity k and x inches thick the over-all resistance,

$$\frac{1}{U} = \frac{1}{f_1} + \frac{x}{k} + \frac{1}{f_o} \tag{3}$$

If the coefficients f_1, f_o and k, together with the thickness of the material x are known, the over-all coefficient U may be readily calculated as the reciprocal of the total heat resistance.

For a compound wall built up of three homogeneous materials having conductivities k_1, k_2 and k_3 and thicknesses x_1, x_2 and x_3 respectively, and laid together without air spaces, the total resistance,

$$\frac{1}{U} = \frac{1}{f_1} + \frac{x_1}{k_1} + \frac{x_2}{k_2} + \frac{x_3}{k_3} + \frac{1}{f_o} \tag{4}$$

For a wall with air space construction consisting of two homogeneous materials of thicknesses x_1 and x_2 and conductivities k_1 and k_2, respectively, separated to form an air space of conductance a, the over-all resistance,

$$\frac{1}{U} = \frac{1}{f_1} + \frac{x_1}{k_1} + \frac{1}{a} + \frac{x_2}{k_2} + \frac{1}{f_o} \tag{5}$$

Likewise any combination of homogeneous materials and air spaces can be put into the wall and the over-all resistance of the combination may be calculated by adding the resistances of the individual sections of the wall. In certain special forms of construction such as tile with irregular air spaces it is necessary to consider the conductance C of the unit as built instead of the unit conductivity k, and the resistance of the section is equal to $\frac{1}{C}$. The method of calculating the over-all heat transmission coefficient for a given wall is comparatively simple, but the selection of the proper coefficients is often complicated. In some cases the construction of the wall is such that the substituting of coefficients in the accepted

formula will give erroneous results. This is the case with irregular cored out air spaces in concrete and tile blocks, and walls in which there are parallel paths for heat flow through materials having different heat resistances. In such cases it is necessary to resort to test methods to check the calculations, and in practically all cases it has been necessary to determine fundamental coefficients by test methods.

Conductivity of Homogeneous Materials

The thermal conductivity of homogeneous materials is effected by several factors. Among these are the density of the material, the amount of moisture present, the mean temperature at which the coefficient is determined, and for fiberous materials the arrangement of fiber in the material. There are many fiberous materials used in building construction and considered as homogeneous for the purpose of calculation, whereas they are not really homogeneous but are merely considered so as a matter of convenience. In general, the thermal conductivity of a material increases directly with the density of the material, increases with the amount of moisture present, and increases with the mean temperature at which the coefficient is determined. The rate of increase for these various factors is not the same for all materials, and in assigning proper coefficients one should make certain that they apply for the conditions under which the material is to be used in a wall. Failure to do this may result in serious errors in the final coefficients.

Surface Conductance Coefficients

Heat is transmitted to or from the surface of a wall by a combination of radiation, convection and conduction. The coefficient will be effected by any factor which has an influence on any one of these three methods of transfer. The amount of heat by radiation is controlled by the character of the surface and the temperature difference between it and the surrounding objects. The amount of heat by conduction and convection is controlled largely by the roughness of the surface, by the air movement over the surface and by the temperature difference between the air and the surface. Because of these variables the surface coefficients may be subject to wide fluctuations for different materials and different conditions. The inside and outside coefficients f_i and f_o are in general effected to the same extent by these various factors and test coefficients determined for inside surfaces will apply equally well to outside surfaces under like conditions. Values for f_i in still and moving air at different mean temperatures have been determined for various building materials at the University of Minnesota under a cooperative agreement with the Society.[1]

The relation obtained between surface conductances for different materials at mean temperatures of 20 F is shown in Fig. 1. These values were obtained with air flow parallel to the surface and from other tests in which the angle of incident between the direction of air flow and the surface was varied from zero to 90 deg it would appear that these values might be lowered approximately 15 per cent for average conditions. While for average building materials there is a difference due to mean

[1]Surface Conductances as Affected by Air Velocity, Temperature and Character of Surface, by F. B. Rowley, A. B. Algren and J. L. Blackshaw (A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930).

temperature, the greatest variation in these coefficients is caused by the character of the surface and the wind velocity. If other surfaces, such as aluminum foil with low emissivity coefficients were substituted, a large part of the radiant heat would be eliminated. This would reduce the total coefficient for all wind velocities by about 0.7 Btu and would make but very little difference for the higher wind velocities. In many cases in building construction the heat resistance of the internal parts of the wall is high as compared with the surface resistance and the surface factors become of small importance. In other cases such as single glass windows the surface resistances constitute practically the entire resistance of the

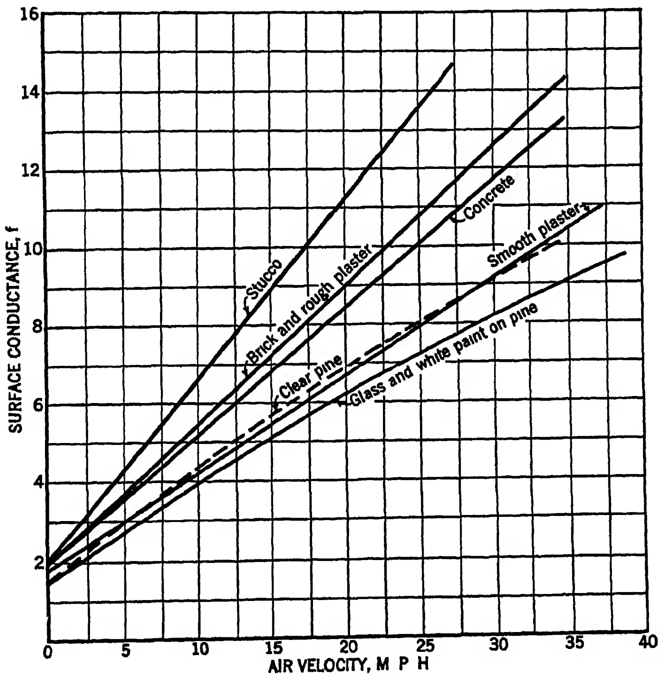


FIG. 1. CURVES SHOWING RELATION BETWEEN SURFACE CONDUCTANCES FOR DIFFERENT SURFACES AT 20 F MEAN TEMPERATURE

structure, and therefore become important factors. Due to the wide variation in surface coefficients for different conditions their selection for a practical building becomes a matter of judgment. In calculating the over-all coefficients for the walls of Tables 3 to 12, 1.65 has been selected as an average inside coefficient and 6.0 as an average outside coefficient for a 15-mile wind velocity. In special cases where surface coefficients become important factors in the over-all rate of heat transfer more selective coefficients may be required.

Air Space Conductance

Heat is conducted across an air space by a combination of radiation, conduction and convection. The amount of heat by radiation is governed

largely by the nature of the surface and the temperature difference between the boundary surfaces of the air space. Conduction and convection are controlled largely by the width and shape of the air space and the roughness of the boundary surfaces. The thermal resistances of air spaces bounded by extended parallel surfaces perpendicular to the direction of heat flow and at different mean temperatures have been determined for average building materials at the University of Minnesota in a cooperative research program with the Society.

The values given in Table 1 show the results of this study and apply to air spaces bounded by such materials as paper, wood, plaster, etc., having emissivity coefficients of from 0.9 to 0.95. The conductivity coefficients decrease with air space width until a width of about ¾ in. has been reached, after which the width has but very little effect. In these

TABLE 1 CONDUCTANCES OF AIR SPACES[a] AT VARIOUS MEAN TEMPERATURES

MEAN TEMP DEG FAHR	CONDUCTANCES OF AIR SPACES FOR VARIOUS WIDTHS IN INCHES						
	0.128	0 250	0 364	0 493	0.713	1.00	1.500
20	2 300	1 370	1.180	1 100	1 040	1.030	1.022
30	2.385	1 425	1.234	1.148	1.080	1.070	1.065
40	2 470	1 480	1 288	1.193	1.125	1.112	1.105
50	2 560	1 535	1.340	1.242	1 168	1.152	1.149
60	2.650	1 590	1.390	1.295	1.210	1.195	1.188
70	2.730	1 648	1.440	1 340	1 250	1.240	1.228
80	2.819	1 702	1.492	1.390	1.295	1.280	1.270
90	2 908	1 757	1 547	1.433	1.340	1.320	1.310
100	2.990	1 813	1 600	1.486	1.380	1.362	1.350
110	3.078	1.870	1.650	1.534	1.425	1.402	1.392
120	3.167	1.928	1 700	1.580	1.467	1.445	1.435
130	3.250	1 980	1.750	1.630	1.510	1.485	1.475
140	3.340	2.035	1.800	1.680	1.550	1.530	1.519
150	3.425	2.090	1.852	1.728	1.592	1.569	1.559

[a]Thermal Resistance of Air Spaces by F B Rowley and A. B. Algren (A S.H.V.E. TRANSACTIONS, Vol 35, 1929).

coefficients radiation is a large factor, and if surfaces with low emissivity coefficients are substituted for ordinary building materials the total amount of radiant heat will be reduced. The reduction in radiant heat caused by the low emissivity surface is independent of width of air space. Values of air spaces lined with bright foil on one or both sides for widths of ⅜ in. and ¾ in. are shown in Table 2 of conductivities. In assigning these values to a practical condition one must be certain that the surfaces of the material will be maintained in a polished condition and not allowed to tarnish and become coated with dust or moisture. The low emissivity coefficient is entirely a surface characteristic and if the surface is covered with any foreign material it will not reflect the heat any more than a mirror so covered would reflect light.

In comparing the conductance coefficients for air spaces with and without bright metallic surface lining it should be noted that the reduction in heat transfer is substantially as great when one surface is lined as it is when both surfaces are lined. The reason for this is that practically 95 per cent of the total radiant heat is intercepted by one surface lining

and there is but a small amount left to be stopped by the second surface lining. The effect of any low emissivity surface in stopping the transmission of radiant heat is the same regardless of whether it is on the high or low temperature side of the air space. For materials such as aluminum or bronze paint which stop only a small percentage of radiant heat there is a greater percentage of gain by addition of a second surface lining.

PRACTICAL COEFFICIENTS

For practical purposes it is necessary to have average coefficients that may be applied to various materials and types of construction without the necessity of making tests on the individual material or combination of materials. In Table 2 coefficients are given for a group of materials which have been selected from various sources Wherever possible the properties of material and conditions of tests are given. However, in selecting and applying these values to any construction a reasonable amount of caution is necessary; variations will be found in the coefficients for the same materials, which may be partly due to different test methods used, but which are largely due to variations in materials. The recommended coefficients which have been used for the calculation of over-all coefficients as given in Tables 3 to 12 are marked by an asterisk.

It should be recognized in these tables of calculated coefficients that space limitations will not permit the inclusion of all the combinations of materials that are used in building construction and the varied applications of insulating materials to these constructions Typical examples are given of combinations frequently used, but any special construction not given in Tables 3 to 12 can generally be computed by using the conductivity values given in Table 2 and the fundamental heat transfer formulae. For example, the tabulation of all of the values for multiple layers of insulating materials would present extensive and detailed problems of calculations for the varied application combinations, but the engineer having the fundamental conductivity values can quickly obtain the proper coefficients.

Attention is called to the fact that the conductivity values per inch of thickness do not afford a true basis for comparison between insulating materials as applied, although they are frequently used for that purpose. The value of an insulating material is measured in terms of its heat resistance, which not only depends upon the thermal conductivity coefficient per inch but also upon the thickness as installed and the manner of installation. For instance the material having a coefficient of 0.50 and 1 in. thick is equal in value to a material having a coefficient of 0.25 and a thickness of ½ in. Certain types of blanket installations are designed to be installed between the studs of a frame building in such manner as to give two air spaces. In order to get the full value of such materials they should be so installed that each air space is approximately 1 in. or more in thickness and the air spaces should be sealed at the top and bottom to prevent the circulation of air from one space to the other. Another common error in installing such a material is to nail the blanket on the outside of the studs underneath the sheathing, in which case one air space is lost and also the thickness of the insulating material is materially reduced at the studs. There are certain other types of insulation which are very porous, allowing air circulation within the material if not

properly installed. Besides the thermal conductivity there are such properties as resistance to rot, vermine and fire which should be considered in making comparisons.

Computed Transmission Coefficients

Computed heat transmission coefficients of many common types of building construction are given in Tables 3 to 13, inclusive, each construction being identified by a serial number. For example, the coefficient of transmission (U) of an 8-in. brick wall and ½ in. of plaster is 0.46, and the number assigned to a wall of this construction is *1-B*, Table 3.

Example 1. Calculate the coefficient of transmission (U) of an 8-in. brick wall with ½ in of plaster applied directly to the interior surface, based on an outside wind exposure of 15 mph. It is assumed that the outside course is of hard (high density) brick having a conductivity of 9.20, and that the inside course is of common (low density) brick having a conductivity of 5 0, the thicknesses each being 4 in The conductivity of the plaster is assumed to be 3 3, and the inside and outside surface coefficients are assumed to average 1.65 and 6 00, respectively, for still air and a 15 mph wind velocity

Solution k (hard high density brick) = 9 20, x = 4 0 in ; k (common low density brick) = 5 0; x = 4 0 in., k (plaster) = 3.3, x = ½ in ; f_i = 1 65; f_o = 6 0 Therefore.

$$U = \frac{1}{\frac{1}{6.0} + \frac{4.0}{9.20} + \frac{4.0}{5.0} + \frac{0.5}{3.3} + \frac{1}{1.65}}$$

$$= \frac{1}{0.167 + 0.435 + 0.80 + 0.152 + 0.606}$$

= 0.46 Btu per hour per square foot per degree Fahrenheit difference in temperature between the air on the two sides.

The coefficients in the tables were determined by calculations similar to those shown in Example 1, using Fundamental Formulae 3, 4 and 5 and the values of k (or C), f_i, f_o and a indicated in Table 2 by asterisks. In computing heat transmission coefficients of floors laid directly on the ground (Table 10), only one surface coefficient (f_i) is used. For example, the value of U for a 1-in. yellow pine floor (actual thickness, 25/32 in.) placed directly on 6-in. concrete on the ground, is determined as follows:

$$U = \frac{1}{\frac{1}{1.65} + \frac{0.781}{0.80} + \frac{6.0}{12.0}} = 0.48$$ Btu per hour per square foot per degree difference in temperature between the ground and the air immediately above the floor.

Rigid insulation refers to the so-called board form which may be used structurally, such as for sheathing. Flexible insulation refers to the blankets, quilts or semi-rigid types of insulation.

Actual thicknesses of lumber are used in the computations rather than nominal thicknesses. The computations for wood shingle roofs applied over wood stripping are based on 1 by 4 in. wood strips, spaced 2 in. apart. Since no reliable figures are available concerning the conductivity of Spanish and French clay roofing tile, of which there are many varieties, the figures for such types of roofs were taken the same as for slate roofs, as it is probable that the values of U for these two types of roofs will compare favorably.

TABLE 2. CONDUCTIVITIES (k) AND CONDUCTANCES (C) OF BUILDING MATERIALS AND INSULATORS[a]

The coefficients are expressed in Btu per hour per square foot per degree Fahrenheit per 1 in thickness, unless otherwise indicated

Material	Description: Cement	Fine Aggregate 0-No 4	Coarse Aggregate No 4-1½	Slump	Per Cent Voids	Density (Lb per Cu Ft)	Mean Temp (Deg Fahr)	Conductivity (k) or Conductance (C)	Resistivity $\left(\frac{1}{k}\right)$ or Resistance $\left(\frac{1}{C}\right)$	Authority
SAND AND GRAVEL, CONCRETE	1	2 00	2 75	0	11 5	144 7	75 06	13 10	0 08	(4)
	1	2 75	4 50	0	10 9	145 7	74 77	12 90	0 08	(4)
	1	3 50	5 50	0	11 2	144 5	75 00	13 20	0 08	(4)
	1	2 00	2 75	5	13 9	142 5	75 50	12 10	0 08	(4)
	1	2 00	2 75	5	13 9	142 5	74 74	12 40	0 08	(4)
	1	2 75	4 50	5	14 6	141 1	73 30	12 40	0 08	(4)
	1	2 75	4 50	5	14 6	141 1	74 89	12 10	0 08	(4)
	1	3 50	5 50	5	14 7	139 2	74 50	12 85	0 08	(4)
	1	3 50	5 50	5	14 7	139 2	75 15	12 50	0 08	(4)
	Avg Value for Sand and Gravel Concrete .					142 3		12 62	--- .	—
LIMESTONE CONCRETE	1	2 00	2 75	0	16 6	135 3	74 87	11 20	0 09	(4)
	1	2 75	4 50	0	15 4	137 8	75 18	12 00	0 08	(4)
	1	3 50	5 50	0	16 3	136 4	74 75	11 50	0 09	(4)
	1	2 00	2 75	3	20 9	130 1	74 85	10 50	0.10	(4)
	1	2 75	4 50	3	23 4	126 0	74 45	10 00	0 10	(4)
	1	3 50	5 50	3	23 4	127 3	75 26	9 79	0 10	(4)
	Avg Value for Limestone Concrete					132 15	. —	10 83	. —	—
CINDER CONCRETE	1	2 00	2 75	0	18 2	103 6	75 26	4 63	0 22	(4)
	1	2 75	4 50	0	19 9	98 7	75 71	4 30	0 23	(4)
	1	3 50	5 50	0	21 4	92 0	75 72	3 73	0 27	(4)
	1	2 00	2 75	3	22 8	101 4	74 95	4 89	0 20	(4)
	1	2 75	4 50	3	26 0	94 0	75 20	4 38	0 23	(4)
	1	3 50	5 50	3	24 4	94 4	75 55	4 24	0 24	(4)
	Avg Value for Cinder Concrete . . .					97 35	.	4 86	.	..
HAYDITE	1	2 00	2 75	0	18 0	80 7	74 82	4 15	0 25	(4)
	1	2 75	4 50	0	19 8	75 0	75 75	3 78	0 26	(4)
	1	3 50	5 50	0	21 8	71 7	74 82	3 67	0 27	(4)
	1	2 00	2 75	4	21 2	78 8	74 76	4 38	0 23	(4)
	1	2 75	4 50	4	22 2	72 4	75 39	3 89	0 26	(4)
	1	2 75	4 50	4	22 2	72 4	75 49	3 86	0 26	(4)
	1	3 50	5 50	4	23 9	71 0	75 46	4 00	0 25	(4)
	Avg Value for Haydite . .					74 57		3 96	- .	—

AUTHORITIES

[1]U S Bureau of Standards, tests based on samples submitted by manufacturers

[2]A C Willard, L C Lichty, and L A Harding, tests conducted at the University of Illinois

[3]J C Peebles, tests conducted at Armour Institute of Technology, based on samples submitted by manufacturers

[4]F. B Rowley, tests conducted at the University of Minnesota

[5]A S H V E Research Laboratory.

[6]E A Allcut, tests conducted at the University of Toronto

[7]Lees and Charlton.

*Recommended conductivities and conductances for computing heat transmission coefficients

†For thickness stated or used on construction, not per 1-in thickness

[a]For additional conductivity data see Chapters 3 and 15, 1937 *A S R E Data Book*

[b]If outside surface of block is painted with an impervious coat of paint, add 0 07 to resistance for sand and gravel blocks Add 0 18 to resistance for cinder blocks Add 0 17 to resistance for haydite blocks

[c]Recommended value See Heating, Ventilating and Air Conditioning, by Harding and Willard, revised edition, 1932

[d]See A S H V E Research Paper, Conductivity of Concrete, by F C Houghten and Carl Gutberlet (A S H V E TRANSACTIONS, Vol 38, 1932)

[e]The 6-in , 8-in , and 10-in hollow tile figures are based on two cells in the direction of heat flow The 12-in hollow tile is based on three cells in the direction of heat flow The 16-in hollow tile consists of one 10-in and one 6-in tile, each having two cells in the direction of heat flow

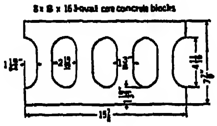
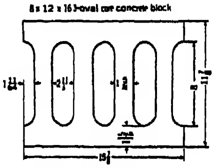
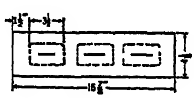
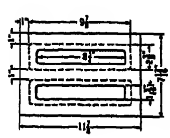
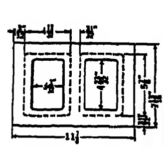
[f]Not compressed

[g]Roofing, 0 15-in thick (1 34 lb per sq ft), covered with gravel (0 83 lb per sq ft), combined thickness assumed 0 25

TABLE 2. CONDUCTIVITIES (k) AND CONDUCTANCES (C) OF BUILDING MATERIALS AND INSULATORS—Continued

The coefficients are expressed in Btu per hour per square foot per degree Fahrenheit per 1 in. thickness unless otherwise indicated

Material	Description: Cement	Fine Aggregate 0-No 4	Coarse Aggregate No 4-1½	Slump	Per Cent Voids	Density (Lb per Cu Ft)	Mean Temp (Deg Fahr)	Conductivity (k) or Conductance (C)	Resistivity $\left(\frac{1}{k}\right)$ or Resistance $\left(\frac{1}{C}\right)$	Authority
EXPANDED BURNED CLAY	1	8.00			18.4	57.9	75.57	2.28	0.44	(4)
STEAM TREATED LIMESTONE SLAG	1	7.00			27.1	74.6	74.49	2.27	0.44	(4)
PUMICE MINED IN CALIF.	1	8.00			26.5	65.0	74.68	2.42	0.41	(4)
BY-PRODUCT OF MANUFACTURE OF PHOSPHATES	1	8.00	Fineness Modulus 3.75		25.5	86.6	74.62	3.19	0.31	(4)
	1	8.00			21.1	91.1	74.43	3.42	0.29	(4)
HAYDITE	1	8.50			21.8	67.1	75.89	2.89	0.35	(4)
	1	8.50			21.8	67.1	74.60	2.815	0.34	(4)

Material	Description	Density (Lb per Cu Ft)	Mean Temp (Deg Fahr)	Conductivity (k) or Conductance (C)	Resistivity or Resistance	Authority
8 x 8 x 16 3-oval core concrete blocks 	Sand and gravel aggregate	126.4	40	0.900†	1.11	(4)
	Sand and gravel aggregate used for calculations			1.000†*	1.00	(4)
	Cores filled with 5.14 lb density cork		40	0.560†*	1.79	(4)
	Crushed limestone aggregate	134.3	40	0.856†*	1.17	(4)
	Cinder aggregate	86.2	40	0.577†	1.73	(4)
	Cinder aggregate used for calculations			0.600†*	1.66	(4)
	Cores filled with 69.7 lb density cinders		40	0.390†	2.56	(4)
	Cores filled with 5.12 lb density cork		40	0.250†*	4.00	(4)
	Cores filled with 14.2 lb density rock wool		40	0.266†*	3.76	(4)
	Haydite aggregate	67.7	40	0.495†*	2.02	(4)
	Cores filled with 5.06 lb density cork		40	0.206†	4.85	(4)
8 x 12 x 16 3-oval core concrete block 	Sand and gravel aggregate	124.9	40	0.777†	1.29	(4)
	Sand and gravel aggregate used for calculations			0.800†*		(4)
	Cinder aggregate	86.2	40	0.531†*	1.88	(4)
	Cores filled with 5.24 lb density cork		40	0.237†*	4.22	(4)
	Haydite aggregate	76.7	40	0.468†	2.13	(4)
	Cores filled with 5.6 lb density cork		40	0.168†*	5.94	(4)
	Cinder aggregate	100.0	40	1.00†	1.00	(4)
	Double wall with 1 in. air space between	100.0	40	0.358†	2.70	(4)
	1 in. space filled with 9.97 lb density rock wool	100.0	40	0.204†	4.90	(4)
	5 x 8 x 12 block sand and gravel aggregate	133.7	40	0.380†	2.63	(4)
	5 x 8 x 12 block sand and gravel aggregate[b]	134.0	40	0.947†	1.06	(4)

For notes see Page 107

TABLE 2 CONDUCTIVITIES (k) AND CONDUCTANCES (C) OF BUILDING MATERIALS AND INSULATORS—Continued

The coefficients are expressed in Btu per hour per square foot per degree Fahrenheit per 1 in thickness. unless otherwise indicated

Material	Description	Density (Lb per Cu Ft)	Mean Temp (Deg Fahr)	Conductivity (k) or Conductance (C)	Resistivity $\left(\frac{1}{k}\right)$ or Resistance $\left(\frac{1}{C}\right)$	Authority
MASONRY MATERIALS						
Brick	Low density			5 00*	0 20	.
	High density			9 20*	0 11	
Brickwork	Damp or wet			5 00[c]	0 20	(2)
Cement Mortar	Typical			12 00*	0 08	
Concrete	Various ages and mixes[d]			11 35to 16 36		(5)
	Cellular	40 0	75	1 06	0 94	(3)
	Cellular	50 0	75	1 44	0 69	(3)
	Cellular	60 0	75	1 80	0 56	(3)
	Cellular	70 0	75	2 18	0 46	(3)
	Typical fiber gypsum, 87 5% gypsum and 12 5% wood chips	51 2	74	1 66*	0 60	(4)
	Special concrete made with an aggregate of hardened clay—1-2-3 mix	101 0	70	3 98	0 25	(3)
Stone	Typical			12 50*	0 08	
Stucco	Typical			12.00*	0 08	
Tile	Typical hollow clay (4 in)			1 00†*	1 00	
	Typical hollow clay (6 in)[e]			0 64†*	1 57	
	Typical hollow clay (8 in)[e]			0 60†*	1 67	
	Typical hollow clay (10 in)[e]			0 58†*	1 72	-
	Typical hollow clay (12 in)[e]			0 40†*	2 50	--
	Typical hollow clay (16 in)[e]			0 31†*	3 23	..
	Hollow clay (2 in) ½-in plaster both sides	120 0	110	1 00†	1 00	(2)
	Hollow clay (4 in) ½-in plaster both sides	127 0	100	0 60†	1 67	(2)
	Hollow clay (6 in) ½-in plaster both sides	124 3	105	0 47†	2 13	(2)
	Hollow gypsum (4 in)			0 46†*	2 18	
	Solid gypsum	51 8	70	1 66	0 60	(4)
	Solid gypsum	75 6	76	2 96	0 34	(4)
Tile or Terrazzo	Typical flooring			12 00*	0 08	
INSULATION—BLANKET OR FLEXIBLE TYPES						
Fiber	Typical	-		0 27*	3 70	—
	Chemically treated wood fibers held between layers of strong paper[f]	3 62	70	0 25	4 00	(3)
	Eel grass between strong paper[f]	4 60	90	0 26	3 85	(1)
	" " " " "	3 40	90	0 25	4 00	(1)
	Flax fibers between strong paper[f]	4 90	90	0 28	3 57	(1)
	Chemically treated hog hair between kraft paper[f]	5 76	71	0 26	3 85	(3)
	Chemically treated hog hair between kraft paper and asbestos paper[f]	7 70	71	0 28	3 57	(3)
	Hair felt between layers of paper[f]	11 00	75	0 25	4 00	(3)
	Kapok between burlap or paper[f]	1 00	90	0 24	4 17	(1)
	Jute fiber[f]	6 70	75	0 25	4 00	(3)
	Ground paper between two layers, each ⅜-in thick made up of two layers of kraft paper (sample ¾-in thick)	12 1	75	0 40†	2 50	(4)
INSULATION—SEMI-RIGID TYPE						
Fiber	Felted cattle hair[f]	13 00	90	0 26	3 84	(1)
	" " "	11 00	90	0.26	3 84	(1)
	Flax[f]	12 10	70	0 30	3 33	(3)
	Flax and rye[f]	13 60	90	0 32	3.12	(1)
	Felted hair and asbestos[f]	7 80	90	0 28	3 57	(1)
	75% hair and 25% jute[f]	6 30	90	0 27	3 70	(1)
	50% hair and 50% jute[f]	6 10	90	0 26	3 85	(1)
	Jute[f]	6 70	75	0 25	4 00	(3)
	Felted jute and asbestos[f]	10 00	90	0 37	2.70	(1)
	Compressed peat moss	11 00	70	0 26	3 84	(3)
INSULATION—LOOSE FILL OR BAT TYPE						
Fiber	Made from ceiba fibers[f]	1 90	75	0 23	4.35	(3)
	" " " "	1 60	75	0 24	4.17	(3)

For notes see page 107.

TABLE 2. CONDUCTIVITIES (k) AND CONDUCTANCES (C) OF BUILDING MATERIALS AND INSULATORS—Continued

The coefficients are expressed in Btu per hour per square foot per degree Fahrenheit per 1 in thickness, unless otherwise indicated

Material	Description	Density (Lb per Cu Ft)	Mean Temp (Deg Fahr)	Conductivity (k) or Conductance (C)	Resistivity $\left(\frac{1}{k}\right)$ or Resistance $\left(\frac{1}{C}\right)$	Authority
INSULATION—LOOSE FILL OR BAT TYPE—Continued						
Fiber	Fibrous material made from dolomite and silica	1.50	75	0.27	3.70	(3)
	Fibrous material made from slag	9.40	103	0.27	3.70	(1)
Glass Wool	Fibrous material 25 to 30 microns in diameter, made from virgin bottle glass	1.50	75	0.27	3.70	(3)
Granular	Made from combined silicate of lime and alumina	4.20	72	0.24	4.17	(3)
	Made from expanded aluminum-magnesium silicate	6.20	42	0.32	3.12	(3)
Gypsum	Cellular, dry	30.00	90	1.00	1.00	(1)
	“ “	24.00	90	0.77	1.30	(1)
	“ “	18.00	90	0.59	1.69	(1)
	“ “	12.00	90	0.44	2.27	(1)
	Flaked, dry and fluffy	34.00	90	0.60	1.67	(1)
	“ “ “ “	26.00	90	0.52	1.92	(1)
	“ “ “ “	24.00	75	0.48*	2.08	(3)
	“ “ “ “	19.80	90	0.35	2.86	(1)
	“ “ “ “	18.00	75	0.34	2.94	(3)
Mineral Wool	All forms, typical			0.27*	3.70	.
Regranulated Cork	About 3/16-in particles	8.10	90	0.31	3.22	(1)
Rock Wool	Fibrous material made from rock	21.00	90	0.30	3.33	(1)
	“ “ “ “ “	18.00	90	0.29	3.45	(1)
	“ “ “ “ “	14.00	90	0.28	3.57	(1)
	“ “ “ “ “	10.00	90	0.27*	3.70	(1)
	Rock wool with a binding agent	14.50	77	0.33	3.03	(1)
	Rock wool with flax, straw pulp, and binder	14.50	75	0.38	2.63	(3)
	Rock wool with vegetable fibers	11.50	72	0.31	3.22	(3)
Sawdust	Various	12.00	90	0.41	2.44	(1)
Shavings	Various from planer	8.80	90	0.41	2.44	(1)
	From maple, beech and birch (coarse)	13.20	90	0.36	2.78	(1)
	Redwood bark	3.00	90	0.31	3.22	(1)
INSULATION—RIGID						
Corkboard	Typical			0.30*	3.33	.
	No added binder	14.00	90	0.34	2.94	(1)
	“ “ “	10.60	90	0.30	3.33	(1)
	“ “ “	7.00	90	0.27	3.70	(1)
	“ “ “	5.40	90	0.25	4.00	(1)
	Asphaltic binder	14.50	90	0.32	3.12	(1)
Fiber	Typical			0.33*	3.03	
	Chemically treated hog hair covered with film of asphalt	10.00	75	0.28	3.57	(3)
	Made from corn stalks	15.00	71	0.33	3.03	(3)
	“ “ exploded wood fibers	17.90	78	0.36	3.12	(4)
	“ “ hard wood fibers	15.20	70	0.32	3.12	(3)
	Insulating plaster 9/10-in thick applied to 3/8-in plaster board base	54.00	75	1.07†	0.93	(3)
	Made from licorice roots	16.10	81	0.34	2.94	(3)
	Made from 85% magnesia and 15% asbestos	19.30	86	0.51	1.96	(1)
	Made from shredded wood and cement	24.20	72	0.46	2.17	(3)
	“ “ sugar cane fiber	13.50	70	0.33	3.03	(3)
	Sugar cane fiber insulation blocks encased in asphalt membrane	13.80	70	0.30	3.33	(3)
	Made from wheat straw	17.00	68	0.33	3.03	(3)
	“ “ wood fiber	15.90	72	0.33	3.03	(3)
	“ “ “ “	15.00	70	0.33	3.03	(3)
	“ “ “ “	.	52	0.33	3.03	(6)
	“ “ “ “	8.50	72	0.29	3.45	(3)
	“ “ “ “	15.20	..	0.33	3.03	(3)
	“ “ “ “	16.90	90	0.34	2.94	(1)
BUILDING BOARDS						
Asbestos	Compressed cement and asbestos sheets	123.00	86	2.70	0.37	(1)
	Corrugated asbestos board	20.40	110	0.48	2.08	(2)

For notes see Page 107.

TABLE 2 CONDUCTIVITIES (k) AND CONDUCTANCES (C) OF BUILDING MATERIALS AND INSULATORS—Continued

The coefficients are expressed in Btu per hour per square foot per degree Fahrenheit per 1 in thickness, unless otherwise indicated

Material	Description	Density (Lb per Cu Ft)	Mean Temp (Deg Fahr)	Conductivity (k) or Conductance (C)	Resistivity $\left(\frac{1}{k}\right)$ or Resistance $\left(\frac{1}{C}\right)$	Authority
BUILDING BOARDS —Continued						
Asbestos	Pressed asbestos mill board	60.50	86	0.84	1.19	(1)
	Sheet asbestos	48.30	110	0.29	3.45	(2)
Gypsum	Gypsum between layers of heavy paper	62.80	70	1.41	0.71	(3)
	Rigid, gypsum between layers of heavy paper (½-in thick)	53.50	90	2.60†	0.38	(1)
	Gypsum mixed with sawdust between layers of heavy paper (0.39-in thick)	60.70	90	3.60†	0.28	(1)
Plaster Board	(⅜ in)		—	3.73†*	0.27	—
	(½ in)			2.82†*	0.35	—
ROOFING CONSTRUCTION						
Roofing	Asphalt, composition or prepared	70.00	75	6.50†*	0.15	(3)
	Built up—⅜-in thick	—	—	3.53†*	0.28	—
	Built up, bitumen and felt, gravel or slag surfaced[a]		—	1.33	0.75	(2)
	Plaster board, gypsum fiber concrete and 3-ply roof covering 2½ in thick	52.40	76	0.58†	1.72	(4)
Shingles	Asbestos	65.00	75	6.00†*	0.17	(3)
	Asphalt	70.00	75	6.50†*	0.15	(3)
	Slate	201.00		10.37*	0.10	(7)
	Wood			1.28†*	0.78	—
PLASTERING MATERIALS						
Plaster	Cement		—	8.00	0.13	(2)
	Gypsum, typical			3.30*	0.30	
	Thickness ⅜ in		73	8.80†	0.11	(4)
Metal Lath and Plaster	Total thickness ¾ in			4.40†*	0.23	
Wood Lath and Plaster	⅜-in plaster, total thickness ¾ in		70	2.50†*	0.40	(4)
BUILDING CONSTRUCTIONS						
Frame	1-in fir sheathing and building paper	—	30	0.71†*	1.41	(4)
	1-in fir sheathing, building paper, and yellow pine lap siding		20	0.50†*	2.00	(4)
	1-in fir sheathing, building paper and stucco		20	0.55	1.82	(4)
	Pine lap siding and building paper—siding 4 in wide		16	0.85†*	1.18	(4)
	Yellow pine lap siding			1.28†*	0.78	
Flooring	Maple—across grain	40.00	75	1.20	0.83	(3)
	Battleship linoleum (¼-in)			1.36†*	0.74	—
AIR SPACE AND SURFACE COEFFICIENTS						
Air Spaces	Over ¾-in faced with ordinary building materials		40	1.10†*	0.91	(4)
Surfaces, Ordinary	Still air (f_i)	—		1.65†*	0.61	(4)
	15 mph—(f_o)			6.00†*	0.17	(4)
Surface, Rough Stucco	15 mph—(f_o)			9.00†*	0.11	(4)
Surface, Bright Aluminum	Still air (f_i)		60	1.18†	0.85	—
AIR SPACES FACED WITH BRIGHT ALUMINUM FOIL						
	Air space, faced one side with bright aluminum foil, over ¾-in wide		50	0.46†*	2.17	(4)
	Air space, faced one side with bright aluminum foil, ⅜-in wide		50	0.62†	1.61	(4)
	Air space, faced both sides with bright aluminum foil, over ¾-in wide		50	0.41†*	2.44	(4)
	Air space, faced both sides with bright aluminum foil, ⅜-in wide		50	0.57†	1.75	(4)

For notes see Page 107.

Table 2 Conductivities (k) and Conductances (C) of Building Materials and Insulators—Continued

The coefficients are expressed in Btu per hour per square foot per degree Fahrenheit per 1 in. thickness, unless otherwise indicated

Material	Description	Density (Lb per Cu Ft)	Mean Temp (Deg Fahr)	Conductivity (k) or Conductance (C)	Resistivity $\left(\frac{1}{k}\right)$ or Resistance $\left(\frac{1}{C}\right)$	Authority
AIR SPACES FACED WITH BRIGHT ALUMINUM FOIL—Continued	Air space divided in two with single curtain of bright aluminum foil (both sides bright)					
	Each space over ¾-in. wide		50	0.23†*	4.35	(4)
	Each space ¾-in. wide		50	0.31†	3.23	(4)
	Air space with multiple curtains of bright aluminum foil bright on both sides, curtains more than ¾-in. apart, air circulation between spaces prevented					
	2 curtains, forming 3 spaces		50	0.15†*	6.8	(4)
	3 curtains, forming 4 spaces		50	0.11†*	9.22	(4)
	4 curtains, forming 5 spaces		50	0.09†*	11.6	(4)
SPACES FACED WITH NON-METALLIC REFLECTIVE SURFACE	Fabric with non-metallic reflective surface (⅛ in. thick) placed in center of a 1½ in. air space		70	0.33†	3.03	(3)
	Core of fiber board coated two sides with non-metallic reflective surface (⅜ in. thick) placed in space having approximately ¾ in. air space on each side.	23.4	70	0.27†	3.70	(3)
	Fiber board coated one side with non-metallic reflective surface (⅜ in. thick) placed in space having approximately ¾ in. air space on each side		75	0.49†	2.04	(3)
	Air space divided in two with fabric faced both sides with non-metallic reflective surface, each space over ¾-in. wide		40	0.33†	3.03	(4)
	Air space over ¾-in. wide faced one side with non-metallic reflective surface		40	0.67†	1.49	(4)
WOODS (Across Grain)						
Balsa		20.0	90	0.58	1.72	(1)
		8.8	90	0.38	2.63	(1)
		7.3	90	0.33	3.03	(1)
California Redwood	0% moisture	22.0	75	0.66	1.53	(4)
	0% "	28.0	75	0.70	1.43	(4)
	8% "	22.0	75	0.70	1.43	(4)
	8% "	28.0	75	0.75	1.33	(4)
	16% "	22.0	75	0.74	1.35	(4)
	16% "	28.0	75	0.80	1.25	(4)
Cypress		28.7	86	0.67	1.49	(1)
Douglas Fir	0% moisture	26.0	75	0.61	1.64	(4)
	0% "	34.0	75	0.67	1.49	(4)
	8% "	26.0	75	0.66	1.52	(4)
	8% "	34.0	75	0.75	1.33	(4)
	16% "	26.0	75	0.76	1.32	(4)
	16% "	34.0	75	0.82	1.22	(4)
Eastern Hemlock	0% moisture	22.0	75	0.60	1.67	(4)
	0% "	30.0	75	0.76	1.32	(4)
	8% "	22.0	75	0.63	1.59	(4)
	8% "	30.0	75	0.81	1.23	(4)
	16% "	22.0	75	0.67	1.49	(4)
	16% "	30.0	75	0.85	1.18	(4)
Hard Maple	0% moisture	40.0	75	1.01	0.99	(4)
	0% "	46.0	75	1.05	0.95	(4)
	8% "	40.0	75	1.08	0.93	(4)
	8% "	46.0	75	1.13	0.89	(4)
	16% "	40.0	75	1.15	0.87	(4)
	16% "	46.0	75	1.21	0.83	(4)

For notes see Page 107

TABLE 2. CONDUCTIVITIES (k) AND CONDUCTANCES (C) OF BUILDING MATERIALS AND INSULATORS—Continued

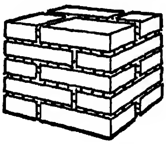
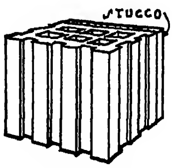
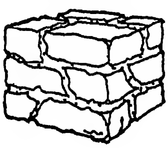
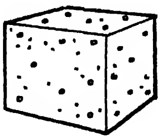
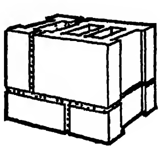
The coefficients are expressed in Btu per hour per square foot per degree Fahrenheit per 1 in thickness, unless otherwise indicated

Material	Description	Density (Lb per Cu Ft)	Mean Temp (Deg Fahr)	Conductivity (k) or Conductance (C)	Resistivity $\left(\frac{1}{k}\right)$ or Resistance $\left(\frac{1}{C}\right)$	Authority
WOODS—Continued						
Longleaf Yellow Pine	0% moisture	30 0	75	0 76	1 32	(4)
	0% "	40 0	75	0 86	1 16	(4)
	8% "	30 0	75	0 83	1 21	(4)
	8% "	40 0	75	0 95	1 05	(4)
	16% "	30 0	75	0 89	1 12	(4)
	16% "	40 0	75	1 03	0 97	(4)
Mahogany		34 3	86	0 90	1 11	(1)
Maple		44 3	86	1 10	0 91	(1)
Maple or Oak				1 15*	0 87	
Norway Pine	0% moisture	22 0	75	0 62	1 61	(4)
	0% "	32 0	75	0 74	1 35	(4)
	8% "	22 0	75	0 68	1 47	(4)
	8% "	32 0	75	0 83	1 21	(4)
	16% "	22 0	75	0 74	1 35	(4)
	16% "	32 0	75	0 91	1 10	(4)
Red Cypress	0% moisture	22 0	75	0 67	1 49	(4)
	0% "	32.0	75	0 79	1 27	(4)
	8% "	22 0	75	0 71	1 41	(4)
	8% "	32 0	75	0 84	1 19	(4)
	16% "	22 0	75	0 74	1 35	(4)
	16% "	32 0	75	0 90	1 11	(4)
Red Oak	0% moisture	38 0	75	0 98	1 02	(4)
	0% "	48 0	75	1 18	0 85	(4)
	8% "	38 0	75	1 03	0 97	(4)
	8% "	48 0	75	1 24	0.81	(4)
	16% "	38 0	75	1 07	0 94	(4)
	16% "	48 0	75	1 29	0 78	(4)
Shortleaf Yellow Pine	0% moisture	76 0	75	0 74	1 35	(4)
	0% "	36 0	75	0 91	1 10	(4)
	8% "	26 0	75	0 79	1.27	(4)
	8% "	36 0	75	0.97	1 03	(4)
	16% "	26 0	75	0 84	1 19	(4)
	16% "	36 0	75	1 04	0.96	(4)
Soft Elm	0% moisture	28 0	75	0.73	1.37	(4)
	0% "	34 0	75	0 88	1.14	(4)
	8% "	28 0	75	0 77	1.30	(4)
	8% "	34 0	75	0 93	1 08	(4)
	16% "	28 0	75	0 81	1 24	(4)
	16% "	34 0	75	0 97	1 03	(4)
Soft Maple	0% moisture	36 0	75	0 89	1 12	(4)
	0% "	42 0	75	0 95	1 05	(4)
	8% "	36 0	75	0 96	1 04	(4)
	8% "	42 0	75	1 02	0 98	(4)
	16% "	36 0	75	1.01	0 99	(4)
	16% "	42 0	75	1 09	0 92	(4)
Sugar Pine	0% moisture	22 0	75	0.54	1 85	(4)
	0% "	28 0	75	0 64	1 56	(4)
	8% "	22 0	75	0 59	1.70	(4)
	8% "	28 0	75	0 71	1 41	(4)
	16% "	22 0	75	0 65	1 54	(4)
	16% "	28 0	75	0 78	1 28	(4)
Virginia Pine		34 3	86	0 96	1 04	(1)
West Coast Hemlock	0% moisture	22 0	75	0 68	1 47	(4)
	0% "	30 0	75	0 79	1 27	(4)
	8% "	22 0	75	0 73	1 37	(4)
	8% "	30 0	75	0 85	1 18	(4)
	16% "	22 0	75	0 78	1 28	(4)
	16% "	30 0	75	0 91	1 10	(4)
White Pine		31 2	86	0 78	1 28	(1)
Yellow Pine				1 00	1 00	(3)
Yellow Pine or Fir				0.80*	1 25	

For notes see Page 107

TABLE 3. COEFFICIENTS OF TRANSMISSION (U) OF MASONRY WALLS[a]

Coefficients are expressed in Btu per hour per square foot per degree Fahrenheit difference in temperature between the air on the two sides, and are based on a wind velocity of 15 mph

TYPICAL CONSTRUCTION	TYPE OF WALL	THICKNESS OF MASONRY (INCHES)	WALL No
	**Solid Brick** Based on 4-in hard brick and the remainder common brick	8 12 16	1 2 3
	**Hollow Tile** Stucco Exterior Finish The 8-in and 10-in tile figures are based on two cells in the direction of flow of heat The 12-in tile is based on three cells in the direction of flow of heat The 16-in tile consists of one 10-in tile and one 6-in tile each having two cells in the direction of heat flow.	8 10 12 16	4 5 6 7
	**Limestone or Sandstone**	8 12 16 24	8 9 10 11
	**Concrete (Monolithic)** These figures may be used with sufficient accuracy for concrete walls with stucco exterior finish	6 10 16 20	12 13 14 15
	Cinder (Monolithic) Conductivity k = 4 36	6 10 16 20	16 17 18 19
	Haydite (Monolithic) Conductivity k = 3 96	6 10 16 20	20 21 22 23
	**Cinder Blocks** Cores filled with dry cinders, 69 7 lb per cu ft. Cores filled with granulated cork, 5.12 lb per cu ft Cores filled with rock wool, 14 2 lb per cu ft Based on one air cell in direction of heat flow Cores filled with granulated cork, 5 24 lb per cu ft.	8 8 8 8 12 12	24 25 26 27 28 29
	Concrete Blocks Cores filled with granulated cork, 5 14 lb per cu ft Based on one air cell in direction of heat flow	8 8 12	30 31 32
	Haydite Blocks Cores filled with granulated cork, 5 06 lb per cu ft.	8 8	33 34
	Haydite Blocks Cores filled with granulated cork, 5 6 lb per cu ft	12 12	35 36

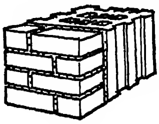
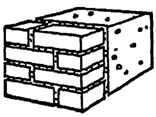
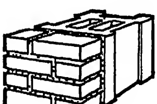
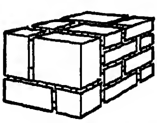
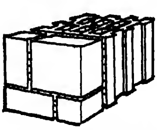
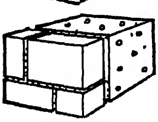
[a]Computed from factors marked by * in Table 2
[b]Based on the actual thickness of 2-in furring strips

INTERIOR FINISH											
UNINSULATED WALLS						INSULATED WALLS					
Plain walls—no interior finish	Plaster (½ in) on walls	Plaster on wood lath—furred	Plaster (¾ in) on metal lath—furred	Plaster (½ in) on plaster board (⅜ in)—furred	Decorated building board (½ in) without plaster—furred	Plaster (½ in) on rigid insulation (½ in)—furred	Plaster (½ in) on rigid insulation (1 in)—furred	Plaster (½ in) on corkboard (1½ in) set in cement mortar (½ in)	Plaster (¾ in) on metal lath attached to furring strips—furred space (over ¾-in wide) faced one side with bright aluminum foil	Plaster on metal lath attached to furring strips (2 in [b])—rock wool fill (1⅝ in [b])[c]	Plaster (¾ in) on metal lath attached to furring strips (2 in [b])—flexible insulation (½ in) between furring strips (one air space)
A	B	C	D	E	F	G	H	I	J	K	L
0 50	0 46	0 30	0 32	0 30	0 23	0 22	0 16	0 14	0 23	0 12	0 20
0 36	0 34	0 24	0 25	0 24	0 19	0 19	0 14	0 12	0 19	0 11	0 17
0 28	0 27	0 20	0 21	0 20	0 17	0 16	0 13	0 11	0 17	0 10	0 15
0 40	0 37	0 26	0 27	0 26	0 20	0 20	0 15	0 13	0 20	0 11	0 18
0 39	0 37	0 26	9.27	0 26	0 20	0 19	0 15	0 13	0 20	0 11	0 18
0 30	0 29	0 22	0 22	0 22	0 17	0 17	0 13	.0 12	0 17	0 10	0 16
0 25	0 24	0 19	0 19	0 19	0 15	0 15	0 12	0 11	0 15	0 097	0 14
0 71	0 64	0 37	0 39	0 37	0 26	0 25	0 18	0 15	0 26	0 13	0 23
0.58	0 53	0 33	0.34	0 33	0 24	0 23	0 17	0 14	0 24	0 13	0 21
0 49	0 45	0.30	0 31	0 30	0 22	0 22	0 16	0 14	0 22	0 12	0 20
0 37	0 35	0 25	0 26	0 25	0 20	0 19	0 15	0 13	0 20	0 11	0 18
0 79	0 70	0 39	0 42	0 39	0 27	0 26	0 19	0 16	0 27	0 13	0 23
0 62	0 57	0 34	0 37	0 34	0 25	0 24	0 18	0 15	0 25	0 13	0 22
0 48	0 44	0 29	0 31	0 29	0 22	0 21	0 16	0 14	0 22	0 12	0 20
0 41	0 39	0 27	0 28	0 27	0 21	0 20	0 15	0 13	0 21	0 12	0 18
0 46	0 43	0 29	0 30	0.29	0 22	0 21	0 16	0 14	0 22	0 12	0 19
0 33	0 31	0.23	0 24	0 23	0 18	0 18	0 14	0 12	0 18	0 11	0 16
0 22	0 22	0.17	0 18	0.17	0.15	0 14	0 12	0 10	0 15	0 09	0 13
0 19	0 18	0.15	0 15	0 15	0 13	0 13	0 11	0 09	0 13	0 09	0 12
0 44	0 41	0 28	0.29	0.28	0 21	0 21	0 16	0 13	0 21	0.12	0 19
0 30	0 29	0 22	0 23	0 22	0 17	0.17	0.14	0 12	0 18	0 10	0 16
0 21	0 20	0 16	0 17	0 16	0 14	0 14	0 11	0 10	0 14	0.09	0 13
0 17	0.17	0 14	0 14	0 14	0 12	0 12	0 10	0 09	0 12	0 08	0 11
0.42	0 39	0 27	0 28	0 27	0 21	0 20	0 16	0 13	0 21	0 12	0 19
0 31	0 29	0.23	0.23	0 22	0 18	0 17	0.14	0 12	0 18	0 11	0 16
0.22	0 21	0.17	0.18	0 17	0 14	0.14	0 12	0 11	0 14	0 09	0 13
0 23	0 22	0.19	0 18	0 18	0 15	0 14	0 12	0 10	0 15	0 09	0 14
0 37	0 35	0.25	0 26	0 25	0,19	0 19	0 15	0 13	0 19	0 11	0 17
0 20	0 19	0.17	0 16	0 16	0 13	0 13	0 11	0 10	0 14	0 09	0 13
0.56	0 52	0 32	0 34	0 32	0 24	0 23	0 17	0 14	0.24	0 12	0 21
0 41	0 39	0 27	0 28	0 27	0 21	0 20	0 15	0 13	0 21	0 12	0 18
0 49	0 46	0 30	0 32	0 30	0 23	0 22	0 16	0 14	0 23	0 12	0 20
0 36	0 34	0 26	0 26	0 24	0 19	0 19	0 15	0 13	0 19	0 11	0 17
0.18	0 17	0 15	0.15	0 14	0 13	0 12	0 10	0 09	0 13	0 08	0 12
0 34	0 32	0 25	0 25	0 24	0 19	0 18	0 14	0 12	0 19	0 11	0 17
0.15	0 14	0 13	0 13	0 12	0 11	0 11	0 09	0 08	0 11	0 08	0 10

[c]A waterproof membrane should be provided between the outer material and the insulation fill to prevent possible wetting by absorption and a subsequent lowering of efficiency

TABLE 4. COEFFICIENTS OF TRANSMISSION (U) OF MASONRY WALLS WITH VARIOUS TYPES OF VENEERS[a]

Coefficients are expressed in Btu per hour per square foot per degree Fahrenheit difference in temperature between the air on the two sides, and are based on a wind velocity of 15 mph

TYPICAL CONSTRUCTION	TYPE OF WALL		WALL No
	FACING	BACKING	
	4 in Brick Veneer[d]	6 in Hollow Tile[c] 8 in 10 in 12 in	37 38 39 40
	4 in Brick Veneer[d]	6 in Concrete 10 in 16 in	41 42 43
	4 in Brick Veneer[d]	8 in Cinder Blocks	44
		8 in Cinder Blocks — Cores filled with granulated cork, 5 12 lb per cu ft	45
		12 in Cinder Blocks	46
		12 in Cinder Blocks — Cores filled with granulated cork, 5 24 lb per cu ft	47
		8 in Concrete Blocks	48
		8 in Concrete Blocks — Cores filled with granulated cork, 5 14 lb per cu ft	49
		12 in Concrete Blocks	50
		8 in Haydite Block	51
		8 in. Haydite Block—Cores filled with granulated cork, 5 06 lb per cu ft	52
		12 in Haydite Block	53
		12 in Haydite Block Cores filled with granulated cork, 5 6 per cu ft.	54
	4 in Cut-Stone Veneer[d]	8 in Common Brick 12 in 16 in	55 56 57
	4 in Cut-Stone Veneer[d]	6 in Hollow Tile[c] 8 in 10 in 12 in	58 59 60 61
	4 in. Cut-Stone Veneer[d]	6 in Concrete 10 in 16 in	62 63 64

[a]Computed from factors marked by * in Table 2.
[b]Based on the actual thickness of 2-in furring strips.
[c]The 6-in , 8-in and 10-in tile figures are based on two cells in the direction of heat flow The 12-in tile is based on three cells in the direction of heat flow.

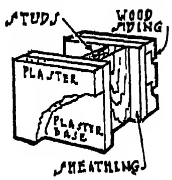
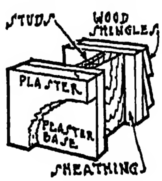
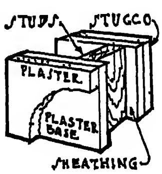
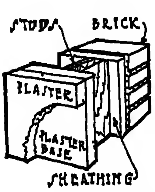
INTERIOR FINISH

Uninsulated Walls						Insulated Walls					
Plain walls—no interior finish	Plaster (½ in.) on walls	Plaster on wood lath—furred	Plaster (¾ in.) on metal lath—furred	Plaster (½ in.) on plaster board (⅜ in.)—furred	No plaster—decorated rigid or building board interior finish (½ in.)—furred	Plaster (½ in. on rigid insulation (½ in.)—furred	Plaster (½ in.) on rigid insulation (1 in.)—furred	Plaster on cork board (1½ in.) set in cement mortar (½ in.)	Plaster on metal lath (¾ in.) attached to furring strips—furred space (over ¾-in. wide) faced one side with bright aluminum foil	Plaster (¾ in.) on metal lath attached to furring strips (2 in.[b])—rock wool fill (1⅝ in.[b])[f]	Plaster (¾ in.) on metal lath attached to furring strips (2 in.[b])—flexible insulation (½ in.) between furring strips (one air space)
A	B	C	D	E	F	G	H	I	J	K	L
0.36	0.34	0.24	0.25	0.24	0.19	0.19	0.16	0.13	0.19	0.11	0.17
0.34	0.33	0.24	0.25	0.24	0.19	0.18	0.14	0.12	0.19	0.11	0.17
0.34	0.32	0.23	0.24	0.23	0.19	0.18	0.14	0.12	0.19	0.11	0.17
0.27	0.26	0.20	0.21	0.20	0.16	0.16	0.13	0.11	0.16	0.10	0.15
0.57	0.53	0.33	0.35	0.33	0.24	0.23	0.17	0.14	0.24	0.13	0.21
0.48	0.45	0.30	0.31	0.30	0.22	0.22	0.16	0.14	0.22	0.12	0.20
0.39	0.37	0.26	0.27	0.26	0.20	0.19	0.15	0.13	0.20	0.11	0.18
0.35	0.33	0.24	0.25	0.24	0.19	0.18	0.14	0.12	0.19	0.11	0.17
0.20	0.19	0.16	0.16	0.16	0.13	0.13	0.11	0.10	0.13	0.09	0.12
0.31	0.30	0.22	0.23	0.22	0.18	0.17	0.14	0.12	0.18	0.11	0.16
0.18	0.18	0.15	0.15	0.15	0.13	0.12	0.10	0.09	0.13	0.08	0.12
0.44	0.42	0.28	0.30	0.28	0.21	0.21	0.16	0.13	0.21	0.12	0.19
0.34	0.32	0.24	0.25	0.23	0.19	0.18	0.14	0.12	0.19	0.11	0.17
0.40	0.38	0.26	0.28	0.26	0.20	0.20	0.15	0.13	0.20	0.11	0.18
0.31	0.29	0.23	0.23	0.22	0.18	0.17	0.14	0.12	0.18	0.11	0.16
0.17	0.16	0.14	0.14	0.14	0.12	0.12	0.10	0.09	0.12	0.08	0.11
0.20	0.28	0.21	0.22	0.21	0.17	0.17	0.13	0.12	0.17	0.10	0.16
0.14	0.14	0.12	0.12	0.12	0.10	0.10	0.09	0.08	0.10	0.07	0.10
0.37	0.35	0.25	0.26	0.25	0.19	0.19	0.15	0.13	0.19	0.11	0.17
0.28	0.27	0.21	0.21	0.21	0.17	0.16	0.13	0.12	0.17	0.10	0.15
0.23	0.22	0.18	0.18	0.18	0.15	0.14	0.12	0.11	0.15	0.095	0.14
0.37	0.35	0.25	0.26	0.25	0.20	0.19	0.15	0.13	0.20	0.11	0.18
0.36	0.34	0.24	0.25	0.24	0.19	0.19	0.15	0.13	0.19	0.11	0.17
0.35	0.33	0.24	0.25	0.24	0.19	0.18	0.14	0.12	0.19	0.11	0.17
0.28	0.26	0.20	0.21	0.20	0.17	0.16	0.13	0.11	0.17	0.10	0.15
0.61	0.56	0.34	0.36	0.34	0.25	0.24	0.18	0.15	0.25	0.13	0.22
0.51	0.47	0.31	0.32	0.31	0.23	0.22	0.17	0.14	0.23	0.12	0.20
0.41	0.38	0.26	0.28	0.26	0.20	0.20	0.15	0.13	0.21	0.11	0.18

[d] Calculations include cement mortar (½ in.) between veneer or facing and backing.
[e] Based on one air cell in direction of heat flow.
[f] A waterproof membrane should be provided between the outer material and the insulation fill to prevent possible wetting by absorption and a subsequent lowering of efficiency.

Table 5. Coefficients of Transmission (U) of Various Types of Frame Construction[a]

These coefficients are expressed in Btu per hour per square foot per degree Fahrenheit difference in temperature between the air on the two sides, and are based on a wind velocity of 15 mph

Typical Construction	Exterior Finish	Type of Sheathing	Wall No
	**Wood Siding or Clapboard**	**1 in Wood**[d]	65
		½ in Rigid Insulation	66
		½ in Plaster Board	67
	**Wood Shingles**	**1 in Wood**[d].	68
		½ in Rigid Insulation[e]	69
		½ in Plaster Board[e]	70
	**Stucco**	**1 in Wood**[d]	71
		½ in. Rigid Insulation	72
		½ in Plaster Board	73
	**Brick Veneer**	**1 in Wood**[d]	74
		½ in Rigid Insulation	75
		½ in Plaster Board	76

[a]Computed from factors marked by * in Table 2.

[b]These coefficients may also be used with sufficient accuracy for plaster on wood lath or plaster on plaster board

[c]Based on the actual width of 2 by 4-in studding, namely, 3⅝ in

INTERIOR FINISH

No Insulation Between Studding							Insulation Between Studding		
Plaster on wood lath on studding	Plaster (¾ in) on metal lath on studding	Plaster (½ in) on plaster board (⅜ in) on studding	Plaster (½ in) on rigid insulation (½ in) on studding	Plaster (½ in) on rigid insulation (1 in) on studding	Plaster (½ in) on corkboard (1½ in) on studding	No plaster—decorated rigid or building board interior finish (½ in)	Plaster (¾ in) on metal lath—stud space faced one side with bright aluminum foil	Plaster (¾ in) on metal lath[b] on studding—rock wool fill (3⅝ in.[e]) between studding[g]	Plaster (¾ in) on metal lath[b] on studding—flexible insulation (½ in) between studding and in contact with sheathing
A	B	C	D	E	F	G	H	I	J
0 25	0 26	0 25	0 19	0 15	0 11	0 19	0 20	0 072	0 17
0 23	0 24	0 23	0 18	0.14	0 11	0 18	0 19	0 070	0 17
0 31	0 33	0 31	0 22	0 17	0 13	0.23	0 24	0 076	0.20
0 25	0 26	0 25	0 19	0 15	0 11	0 19	0.20	0 072	0 17
0.19	0.20	0 19	0.15	0 12	0 10	0 16	0.16	0 066	0 14
0 24	0 25	0.24	0 19	0 15	0 11	0 19	0 19	0 071	0 17
0 30	0 31	0 30	0.22	0 16	0 12	0 22	0 23	0 076	0 20
0.27	0 29	0 27	0 20	0 16	0 12	0 21	0 22	0 074	0 19
0 40	0 43	0 40	0 26	0 19	0 14	0 28	0 29	0 081	0 24
0 27	0 28	0 27	0.20	0 15	0 12	0 21	0 21	0 074	0 18
0 25	0 26	0 25	0.19	0.15	0 11	0 19	0.20	0 072	0 18
0 35	0 37	0.35	0 24	0.18	0.13	0 25	0 26	0 079	0 22

[d]Yellow pine or fir—actual thickness about 25/32 in

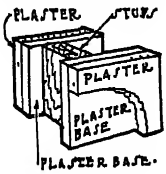
[e]Furring strips between wood shingles and sheathing

[f]Small air space and mortar between building paper and brick veneer neglected

[g]A waterproof membrane should be provided between the outer material and the insulation fill to prevent possible wetting by absorption and a subsequent lowering of efficiency.

Table 6 Coefficients of Transmission (U) of Frame Interior Walls and Partitions[a]

Coefficients are expressed in Btu per hour per square foot per degree Fahrenheit difference in temperature between the air on the two sides, and are based on still air (no wind) conditions on both sides

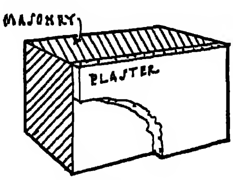


Typical Construction	Wall No	Single Partition (Finish on One Side of Studding)	Double Partition (Finished on Both Sides of Studding): Air Space Between Studding	Flaked Gypsum Fill[b] Between Studding	Rock Wool Fill[b] Between Studding	½-in Flexible Insulation Between Studding (One Air Space)	Stud Space Faced One Side with Bright Aluminum Foil
Type of Wall		A	B	C	D	E	F
Wood Lath and Plaster On Studding	77	0.62	0.34	0.11	0.076	0.21	0.24
Metal Lath and Plaster[c] On Studding	78	0.69	0.39	0.11	0.078	0.23	0.26
Plaster Board (⅜ in) and Plaster[d] On Studding	79	0.61	0.34	0.10	0.075	0.21	0.24
½ in Rigid Insulation and Plaster[d] On Studding	80	0.35	0.18	0.083	0.063	0.14	0.15
1 in Rigid Insulation and Plaster[d] On Studding	81	0.23	0.12	0.066	0.054	0.097	0.10
1⅜ in Corkboard and Plaster[d] On Studding	82	0.16	0.081	0.052	0.044	0.070	0.073
2 in Corkboard and Plaster[d] On Studding	83	0.12	0.063	0.045	0.038	0.057	0.050

[a]Computed from factors marked by * in Table 2
[b]Thickness assumed 3⅝ in.
[c]Plaster on metal lath assumed ¾-in thick.
[d]Plaster assumed ½-in thick

Table 7. Coefficients of Transmission (U) of Masonry Partitions[a]

Coefficients are expressed in Btu per hour per square foot per degree Fahrenheit difference in temperature between the air on the two sides, and are based on still air (no wind) conditions on both sides



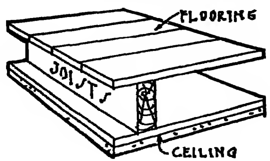
Typical Construction	No	Plain Walls (No Plaster)	Walls Plastered on One Side	Walls Plastered on Both Sides
Type of Wall		A	B	C
4-in Hollow Clay Tile	84	0.45	0.42	0.40
4-in Common Brick	85	0.50	0.46	0.43
4-in Hollow Gypsum Tile	86	0.30	0.28	0.27
2-in Solid Plaster	87	——	——	0.53

[a]Computed from factors marked by * in Table 2

TABLE 8 COEFFICIENTS OF TRANSMISSION (U) OF FRAME CONSTRUCTION FLOORS AND CEILINGS[a]

Coefficients are expressed in Btu per hour per square foot per degree Fahrenheit difference in temperature between the air on the two sides, and are based on still air (no wind) conditions on both sides

TYPICAL CONSTRUCTION



TYPE OF CEILING	INSULATION BETWEEN JOISTS	No	TYPE OF FLOORING: No Flooring	Yellow Pine Flooring[b] on Joists	Yellow Pine Flooring on Rigid Insulation (½ in) on Joists	Maple or Oak Flooring[c] on Yellow Pine Sub-Flooring[b] on Joists	¼-in Battleship Linoleum on Yellow Pine Flooring[b]
			A	B	C	D	E
No Ceiling	None	1	- ----	0.46	0.27	0.34	0.34
Metal Lath and Plaster (¾ in)	None	2	0.69	0.30	0.21	0.25	0.25
Wood Lath and Plaster	None	3	0.62	0.28	0.20	0.24	0.24
Plaster Board (⅜ in) and Plaster (½ in)	None	4	0.61	0.28	0.20	0.24	0.23
Rigid Insulation (½ in) and Plaster (½ in)	None	5	0.35	0.21	0.16	0.18	0.18
Metal Lath and Plaster	Flexible[d] Insulation (½ in)	6	0.24	0.16	0.13	0.15	0.15
Metal Lath and Plaster	Rigid Insulation[d] (½ in)	7	0.26	0.17	0.14	0.15	0.15
Metal Lath and Plaster	Bright Aluminum Foil[e]	8	0.59	0.22	0.17	0.19	0.19
Metal Lath and Plaster	Rock Wool Fill (3⅝ in)	9	0.079	0.068	0.063	0.066	0.066
Corkboard (1½ in) and Plaster (½ in)	None	10	0.16	0.12	0.10	0.11	0.11
Corkboard (2 in) and Plaster (½ in)	None	11	0.12	0.10	0.087	0.094	0.091

[a]Computed from factors marked by * in Table 2

[b]Thickness assumed to be 25/32 in.

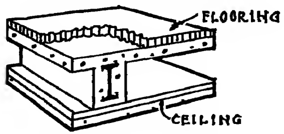
[c]Thickness assumed to be 13/16 in.

[d]Based on one air space with no flooring, and two air spaces with flooring. The value of U will be the same if insulation is applied to under side of joists and separated from lath and plaster ceiling by 1-in furring strips.

[e]Air space faced on one side with bright aluminum foil

Table 9. Coefficients of Transmission (U) of Concrete Construction Floors and Ceilings[a]

Coefficients are expressed in Btu per hour per square foot per degree Fahrenheit difference in temperature between the air on the two sides, and are based on still air (no wind) conditions on both sides



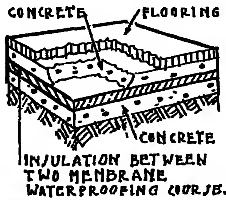
Typical Construction	Thickness of Concrete (Inches)	No	Type of Flooring				
			No Flooring (Concrete Bare)[b]	Yellow Pine Flooring[c] on Wood Sleepers Embedded in Concrete[d]	Maple or Oak Flooring[e] on Yellow Pine Sub-Flooring[c] on Wood Sleepers Embedded in Concrete	Tile or Terrazzo[f] Flooring on Concrete	¼-in Battleship Linoleum Directly on Concrete
Type of Ceiling			A	B	C	D	E
No Ceiling	4	1	0.65	0.40	0.31	0.61	0.44
	6	2	0.59	0.37	0.30	0.56	0.41
	8	3	0.53	0.35	0.28	0.51	0.38
	10	4	0.49	0.33	0.27	0.47	0.36
½ in Plaster Applied Directly to Under Side of Concrete	4	5	0.59	0.38	0.30	0.56	0.41
	6	6	0.54	0.35	0.28	0.52	0.38
	8	7	0.50	0.33	0.27	0.47	0.36
	10	8	0.45	0.32	0.26	0.44	0.34
Suspended or Furred Metal Lath and Plaster (¾ in) Ceiling	4	9	0.37	0.28	0.23	0.36	0.29
	6	10	0.35	0.26	0.22	0.34	0.28
	8	11	0.33	0.25	0.21	0.32	0.27
	10	12	0.32	0.24	0.21	0.31	0.25
Suspended or Furred Ceiling of Plaster Board (⅜ in) and Plaster (½ in)	4	13	0.35	0.26	0.22	0.34	0.28
	6	14	0.33	0.25	0.21	0.32	0.26
	8	15	0.31	0.24	0.21	0.30	0.25
	10	16	0.30	0.23	0.20	0.29	0.24
Suspended or Furred Ceiling of Rigid Insulation (½ in) and Plaster (½ in)	4	17	0.24	0.20	0.17	0.24	0.21
	6	18	0.23	0.19	0.17	0.23	0.20
	8	19	0.22	0.18	0.16	0.22	0.19
	10	20	0.22	0.18	0.16	0.21	0.19
Plaster (½ in) on Corkboard (1½ in) Set in Cement Mortar (½ in.) on Concrete	4	21	0.15	0.13	0.12	0.14	0.14
	6	22	0.14	0.13	0.12	0.14	0.13
	8	23	0.14	0.12	0.11	0.14	0.13
	10	24	0.14	0.12	0.11	0.14	0.13

[a]Computed from factors marked by * in Table 2.
[b]The figures in Column A may be used with sufficient accuracy for concrete floors covered with carpet.
[c]Thickness of yellow pine flooring assumed to be 25/32 in
[d]The figures in Column B may be used with sufficient accuracy for maple or oak flooring[e] applied directly over the concrete on wood sleepers
[e]Thickness of maple or oak flooring assumed to be 13/16 in.
[f]Thickness of tile or terrazzo assumed 1 in.

TABLE 10 COEFFICIENTS OF TRANSMISSION (U) OF CONCRETE FLOORS ON GROUND WITH VARIOUS TYPES OF FINISH FLOORING[a, e]

Coefficients are expressed in Btu per hour per square foot per degree Fahrenheit difference in temperature between the ground and the air over the floor, and are based on still air (no wind) conditions

TYPICAL CONSTRUCTION



Type and Thickness of Insulation	Thickness of Concrete (Inches)	No	No Flooring (Concrete Bare)	Yellow Pine Flooring[b] on Wood Sleepers Resting on Concrete	Maple or Oak Flooring[c] on Yellow Pine Sub-Flooring on Wood Sleepers Resting on Concrete	Tile or Terrazzo[d] on Concrete	¼-in Battleship Linoleum Directly on Concrete
			A	B	C	D	E
None	4	1	1.07	0.35	0.28	0.98	0.60
	6	2	0.90	0.33	0.27	0.84	0.54
	8	3	0.79	0.32	0.26	0.74	0.50
	10	4	0.70	0.30	0.25	0.66	0.46
None[e]	4	5	0.66	0.29	0.24	0.63	0.44
	8	6	0.54	0.27	0.23	0.52	0.39
1 in Rigid Insulation[e]	4	7	0.22	0.16	0.14	0.22	0.19
1 in Rigid Insulation[e]	8	8	0.21	0.15	0.13	0.20	0.18
2 in Corkboard[e]	4	9	0.12	0.099	0.093	0.12	0.11
2 in Corkboard[e]	8	10	0.12	0.096	0.090	0.12	0.11

[a]Computed from factors marked by * in Table 2

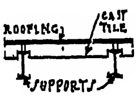
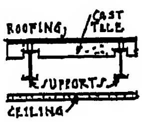
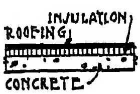
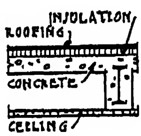
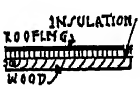
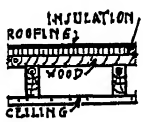
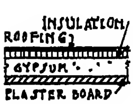
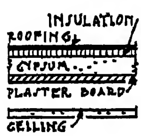
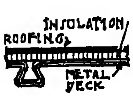
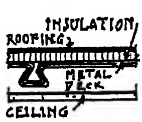
[b]Assumed 25/32 in thick

[c]Assumed 13/16 in thick.

[d]Assumed 1 in thick

[e]The figures for Nos. 5 to 10, inclusive, include 3-in cinder concrete placed directly on the ground The insulation is applied between the cinder concrete and the stone concrete Usually the insulation is protected on both sides by a waterproof membrane, but this is not considered in the calculations

Table 11 Coefficients of Transmission (U) of Various Types of Flat Roofs Covered with Built-Up Roofing[a]

Typical Construction: Without Ceilings	Typical Construction: With Metal Lath and Plaster Ceilings[d]	Type of Roof Deck	Thickness of Roof Deck (Inches)	No
		**Precast Cement Tile**	1⅝	1
		**Concrete** **Concrete** **Concrete**	2 4 6	2 3 4
		**Wood** **Wood** **Wood** **Wood**	1[b] 1½[b] 2[b] 4[b]	5 6 7 8
		**Gypsum Fiber Concrete[c] (2 in.) on Plaster Board (⅜ in.)** **Gypsum Fiber Concrete[c] (3 in.) on Plaster Board (⅜ in.)** **Gypsum Fiber Concrete[c] (2 in.) on Rigid Insulation Board (½ in.)** **Gypsum Fiber Concrete[c] (2 in.) on Rigid Insulation Board (1 in.)**	2⅜ 3⅜ 2½ 3	9 10 11 12
		**Flat Metal Roofs** Coefficient of transmission of bare corrugated iron (no roofing) is 1.50 Btu per hour per square foot of projected area per degree Fahrenheit difference in temperature, based on an outside wind velocity of 15 mph	...	13

[a]Computed from factors marked by * in Table 2.
[b]Nominal thicknesses specified—actual thicknesses used in calculations.
[c]Gypsum fiber concrete—87½ per cent gypsum, 12½ per cent wood fiber

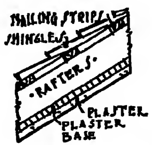
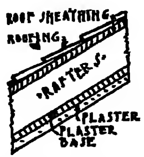
Coefficients are expressed in Btu per hour per square foot per degree Fahrenheit difference in temperature between the air on the two sides, and are based on an outside wind velocity of 15 mph

WITHOUT CEILING—UNDER SIDE OF ROOF EXPOSED								WITH METAL LATH AND PLASTER CEILINGS[d]							
No Insulation	Rigid Insulation (½ in)	Rigid Insulation (1 in)	Rigid Insulation (1½ in)	Rigid Insulation (2 in)	Corkboard (1 in)	Corkboard (1½ in)	Corkboard (2 in)	No Insulation	Rigid Insulation (½ in)	Rigid Insulation (1 in)	Rigid Insulation (1½ in)	Rigid Insulation (2 in)	Corkboard (1 in)	Corkboard (1½ in)	Corkboard (2 in)
A	B	C	D	E	F	G	H	I	J	K	L	M	N	O	P
0 84	0 37	0 24	0 18	0 14	0 22	0 16	0 13	0 43	0 26	0 19	0 15	0 12	0 18	0 14	0.11
0 82	0 37	0 24	0 17	0 14	0 22	0 16	0 13	0.42	0 26	0 19	0 15	0 12	0 18	0 14	0 11
0 72	0 34	0 23	0 17	0 13	0 21	0 16	0 12	0 40	0 25	0 18	0 14	0 12	0 17	0 13	0 11
0 64	0 33	0 22	0 16	0 13	0 21	0 15	0 12	0 37	0 24	0 18	0 14	0 11	0 17	0 13	0 11
0 49	0 28	0.20	0 15	0 12	0 19	0 14	0 12	0 32	0 21	0 16	0 13	0 11	0 15	0 12	0 10
0 37	0.24	0 18	0 14	0 11	0 17	0 13	0 11	0 26	0 19	0 15	Q 12	0 10	0 14	0 11	0 095
0 32	0 22	0 16	0 13	0 11	0 16	0 12	0 10	0 24	0 17	0 14	0 11	0 097	0 13	0 11	0 092
0 23	0 17	0 14	0 11	0 096	0.13	0 11	0.091	0 18	0 14	0 12	0 10	0 087	0 11	0 096	0 082
0 40	0 25	0.18	0 14	0 12	0 17	0.13	0 11	0 27	0 19	0.15	0 12	0 10	0 14	0 12	0 097
0 32	0 22	0 16	0 13	0.11	0 15	0 12	0 10	0 23	0 17	0.14	0 11	0 097	0 13	0 11	0 091
0 26	0 19	0 15	0 12	0 10	0 14	0 11	0 10	0.20	0 16	0 13	0 11	0 09	0.12	0 10	0 087
0 19	0 15	0 12	0.10	0 09	0.12	0 10	0 08	0 16	0 13	0 11	0 09	0.08	0 10	0 09	0 077
0 95	0 39	0 25	0 18	0 14	0.23	0 17	0 13	0.46	0 27	0 19	0 15	0 12	0 18	0 14	0 11

[d]These coefficients may be used with sufficient accuracy for wood lath and plaster, or plaster board and plaster ceilings It is assumed that there is an air space between the under side of the roof deck and the upper side of the ceiling

TABLE 12. COEFFICIENTS OF TRANSMISSION (*U*) OF PITCHED ROOFS[a]

Coefficients are expressed in Btu per hour per square foot per degree Fahrenheit difference in temperature between the air on the two sides, and are based on an outside wind velocity of 15 mph

TYPICAL CONSTRUCTION	TYPE OF ROOFING AND ROOF SHEATHING	INSULATION BETWEEN ROOF RAFTERS	No	TYPE OF CEILING (APPLIED DIRECTLY TO ROOF RAFTERS)								
				No Ceiling (Rafters Exposed)	Metal Lath and Plaster (¾ in.)	Plaster Board (⅜ in.) and Plaster (½ in.)	Wood Lath and Plaster	Rigid Insulation (½ in.)	Rigid Insulation (½ in.) and Plaster (½ in.)	Rigid Insulation (1 in.) and Plaster (½ in.)	Corkboard (1½ in.) and Plaster (½ in.)	Corkboard (2 in.) and Plaster (½ in.)
				A	B	C	D	E	F	G	H	I
	Wood Shingles on Wood Strips[b]	None	1	0.46	0.30	0.29	0.29	0.22	0.21	0.16	0.12	0.10
		½ in. Flexible[c]	2	----	0.17	0.16	0.16	0.14	0.13	0.11	0.091	0.079
		1 in. Flexible[c]	3	----	0.13	0.12	0.12	0.11	0.11	0.092	0.078	0.069
		Bright Aluminum Foil[g]	4	----	0.22	0.21	0.21	0.18	0.17	0.14	0.11	0.089
		3⅝ in. Rock Wool[e]	5	----	0.063	0.062	0.062	0.058	0.058	0.053	0.048	0.044
	Asphalt Shingles, Rigid Asbestos Shingles, Composition Roofing, or Slate or Tile Roofing[d] on Wood Sheathing[f]	None	6	0.56	0.34	0.32	0.32	0.24	0.23	0.17	0.13	0.11
		½ in. Flexible[c]	7	----	0.18	0.17	0.17	0.14	0.14	0.12	0.094	0.089
		1 in. Flexible[c]	8	----	0.13	0.13	0.13	0.11	0.11	0.095	0.080	0.071
		Bright Aluminum Foil[g]	9	----	0.24	0.24	0.24	0.19	0.18	0.14	0.11	0.098
		3⅝ in. Rock Wool[e]	10	----	0.065	0.064	0.064	0.060	0.059	0.054	0.049	0.045

[a]Computed from factors marked by * in Table 2. Nos. 6 to 10, inclusive, based on ½-in. thick slate.
[b]Based on 1 in. by 4 in. strips spaced 2 in.
[c]Figures based on two air spaces. Insulation may also be applied to under side of roof rafters with furring strips between.
[d]Roofing felt between roof sheathing and slate or tile neglected in calculations.
[e]Assumed 3⅝ in. thick based on the actual width of 2 in. by 4 in. rafters.
[f]Sheathing assumed 25/32 in. thick.
[g]Air space faced on one side with bright aluminum foil.

TABLE 13. COEFFICIENTS OF TRANSMISSION (U) OF DOORS, WINDOWS, SKYLIGHTS AND GLASS WALLS

Coefficients are based on a wind velocity of 15 mph, and are expressed in Btu per hour per square foot per degree Fahrenheit difference in temperature between the air inside and outside of the door, window, skylight or wall

A. Windows and Skylights

DESCRIPTION	U
Single	1.13[a, c]
Double	0.45[a]
Triple	0.281[a]

B. Solid Wood Doors[b, c]

NOMINAL THICKNESS INCHES	ACTUAL THICKNESS INCHES	U
1	25/32	0.69
1¼	1 1/16	0.59
1½	1 5/16	0.52
1¾	1⅜	0.51
2	1⅝	0.46
2½	2⅛	0.38
3	2⅝	0.33

C. Glass Walls

DESCRIPTION	U
Hollow glass tile wall, 6 x 6 x 2 in thick blocks	
Wind velocity 15 mph, outside surface, still air, inside surface	0.60
Still air, outside and inside surface	0.48

[a]See *Heating, Ventilating and Air Conditioning*, by Harding and Willard, revised edition, 1932.
[b]Computed using $C = 1.15$ for wood, $f_i = 1.65$ and $f_o = 6.0$
[c]It is sufficiently accurate to use the same coefficient of transmission for doors containing thin wood panels as that of single panes of glass, namely, 1.13 Btu per hour per square foot per degree difference between inside and outside air temperatures

The thicknesses upon which the coefficients in Tables 3 to 13, inclusive, are based are as follows:

Brick veneer	4 in.
Plaster and metal lath	¾ in
Plaster (on wood lath, plasterboard, rigid insulation, board form, or corkboard)	½ in.
Slate (roofing)	½ in.
Stucco on wire mesh reinforcing	1 in.
Tar and gravel or slag-surfaced built-up roofing	⅜ in.
1-in. lumber (S-2-S)	25/32 in.
1½-in lumber (S-2-S)	1 5/16 in.
2-in lumber (S-2-S)	1⅝ in.
2½-in. lumber (S-2-S)	2⅛ in
3-in. lumber (S-2-S)	2⅝ in.
4-in. lumber (S-2-S)	3⅝ in.
Finish flooring (maple or oak)	13/16 in.

Solid brick walls are based on 4-in. hard brick (high density) and the remainder common brick (low density). Stucco is assumed to be 1-in. thick on masonry walls. Where metal lath and plaster are specified, the metal lath is neglected.

The coefficients of transmission of the pitched roofs in Table 12 apply where the roof is over a heated attic or top floor so the heat passes directly through the roof structure including whatever finish is applied to the underside of the roof rafters.

Combined Coefficients of Transmission

If the attic is unheated, the roof structure and ceiling of the top floor must both be taken into consideration, and the combined coefficient of transmission determined. The formula for calculating the combined coefficient of transmission of a top floor ceiling, unheated attic space, and pitched roof, per square foot of ceiling area, is as follows.

$$U = \frac{U_r \times U_{ce}}{U_r + \frac{U_{ce}}{n}} \tag{6}$$

where

U = combined coefficient to be used with ceiling area
U_r = coefficient of transmission of the roof.
U_{ce} = coefficient of transmission of the ceiling.
n = the ratio of the area of the roof to the area of the ceiling

Stating the formula in terms of the total heat resistance of the ceiling and roof,

$$\frac{1}{U} = R = \frac{1}{U_{ce}} + \frac{1}{U_r \times n} \tag{7}$$

In selecting the values to be used for U_r and U_{ce} it should be noted that the under surface of the roof and the upper surface of the ceiling are more nearly equivalent to the boundary surfaces of an internal air space than they are to the external surfaces of a wall. It would be more nearly correct to use a value of 2 2 rather than the usual value of 1.65 as coefficients for these surfaces. In most cases this would make only a minor change in U. It should be noted that the over-all coefficient should be multiplied by the ceiling and not the roof area.

If the unheated attic space between the roof and ceiling has no dormers, windows or vertical wall spaces the combined coefficients may be used for determining the heat loss through the roof construction between the attic and top floor ceiling. If the unheated attic contains windows and vertical wall spaces these must be taken into consideration in calculating the roof area and also its coefficient U_r. In this case an approximate value of U_r may be obtained as the summation of the coefficient of each individual section such as the roof, vertical walls or windows times its percentage of total area This coefficient may be used with reasonable accuracy in the above formulae. If, however, there are roof ventilators such that the attic air is substantially at outside temperature, then the roof should be neglected and only the coefficient for the top floor ceiling construction used.

Basements and Unheated Rooms

The heat loss through floors into basements and into unheated rooms kept closed may be computed by assuming a temperature for these rooms

of 32 F Additional information on the inside and outside temperatures to be used in heat loss calculations is given in Chapter 7.

REFERENCES

Effects of Air Velocities on Surface Coefficients, by F B Rowley, A B Algren and J L Blackshaw (A S H V E TRANSACTIONS, Vol 36, 1930)

Wind Velocity Gradients Near a Surface and Their Effect on Film Conductance, by F C Houghten and Paul McDermott (A S H V E TRANSACTIONS, Vol 37, 1931)

Conductivity of Concrete, by F C Houghten and Carl Gutberlet (A S H V E TRANSACTIONS, Vol 37, 1931)

Surface Coefficients as Affected by Direction of Wind, by F B Rowley and W A Eckley (A S H V E TRANSACTIONS, Vol 37, 1931)

Insulating Effect of Successive Air Spaces Bounded by Bright Metallic Surfaces, by L W Schad (A S H V E TRANSACTIONS, Vol 37, 1931)

Importance of Radiation in Heat Transfer Through Air Spaces, by E R Queer (A S H V E TRANSACTIONS, Vol 38, 1932)

Properties of Metal Foil as an Insulating Material, by J L Gregg (*Refrigerating Engineering*, May, 1932)

Thermal Insulation with Aluminum Foil, by R B Mason (*Industrial and Engineering Chemistry*, March, 1933)

The Heat Conductivity of Wood at Climatic Temperature Differences, by F B Rowley (A S.H V E. Journal Section, *Heating, Piping and Air Conditioning*, June, 1933)

Thermal Insulation of Buildings, Technical Paper No 11 (*American Architect*, May, 1934).

Insulating Value of Bright Metallic Surfaces, by F B Rowley (A S H V E Journal Section, *Heating, Piping and Air Conditioning*, June, 1934)

Thermal Properties of Concrete Construction, by F. B Rowley, A B. Algren and Clifford Carlson (A S H V E Journal, *Heating, Piping and Air Conditioning*, January, 1936)

Thermal Properties of Concrete Construction, by F. B Rowley, A B Algren and Robert Lander (A S H V E Journal Section, *Heating, Piping and Air Conditioning*, November, 1936).

Heat Insulation as Applied to Buildings and Structures, by E A Allcut, University of Toronto, 1934

House Insulation, Its Economies and Application, by Russell E Backstrom (Report of the National Committee on Wood Utilization, *United States Government Printing Office*, 1931)

Heat Transmission Through Building Materials, by F B Rowley and A. B Algren, *University of Minnesota Engineering Experiment Station Bulletin No 8*

Calculation of Heat Transmission, by Margaret Fishenden and Owen A Saunders

Heating, Ventilating and Air Conditioning, by Harding and Willard, Revised Edition, 1932.

Heat Transmission, by W. H McAdams

Industrial Heat Transfer, by Shack

PROBLEMS IN PRACTICE

1 ● What is the coefficient *U* and how is it applied?

The coefficient *U* is the heat loss through walls, ceilings, and floors and the value depends upon the construction and material, expressed in Btu per hour per square foot per degree difference in temperature between the inside and outside. To determine the total heat loss, multiply *U* for each material by the square feet of surface and the temperature difference.

2 ● Find the value of U for a 6-in. concrete wall with plaster on metal lath attached to 2-in. furring strips with flanged ½-in. blanket insulation.

0.23 (Table 3, Wall 12L)

3 ● A wall is built with two layers of ½-in. insulating material spaced 1 in. apart; the air space is lined on one side with bright aluminum foil; mean temperature is 40 F; still air on both sides of wall; *k* for insulating material is 0.34. Calculate the value of *U*.

$$f_i = 1.65, f_o = 1.65, a = 0.46$$

$$R = \frac{1}{1.65} + \frac{0.5}{0.34} + \frac{1}{0.46} + \frac{0.5}{0.34} + \frac{1}{1.65} = 6.327$$

$$U = \frac{1}{R} = 0.158$$

4 ● What is the inside surface temperature of a 6-in. solid concrete wall? Inside air, 70 F; outside air, −20 F with 15 mph wind.

The temperature drop from point to point through a wall is directly proportional to the heat resistance

$$f_i = 1.65;\ k \text{ for concrete} = 12;\ f_o = 6.0$$

$$\text{Over-all resistance } R = \frac{1}{1.65} + \frac{6}{12} + \frac{1}{6.0} = 1.27$$

$$\frac{\text{Temperature drop, inside air to surface}}{\text{Temperature drop, air to air}} = \frac{\frac{1}{1.65}}{1.27}$$

$$\text{Temperature drop, inside air to surface} = \frac{90}{1.27 \times 1.65} = 43$$

$$70 - 43 = 27 \text{ F, inside surface temperature of wall}$$

5 ● How many inches of insulating material having a conductivity of 0.30 would be required, for the wall of Question 4, to raise the inside surface temperature to 60 F?

Temperature drop, air to inside surface = 10 F, temperature drop, inside surface to outside air = 80 F. Therefore, the heat resistance from inside wall surface to outside air must be eight times that from inside air to inside wall surface, or $8 \times \frac{1}{1.65} = 4.85$ The resistance for added material is, therefore,

$$4.85 - \left(\frac{6}{12} + \frac{1}{6}\right) = 4.19$$

$$4.19 \times 0.30 = 1.25 \text{ in. of insulation.}$$

6 ● An unheated attic space in a residence has an equivalent pitched roof area of 1560 sq ft and a ceiling area of 1200 sq ft. If 15 per cent of the roof area is composed of vertical wall spaces having a value of U = 0.52, determine the total heat loss per hour through the ceiling and roof for a temperature difference of 85 F, if U = 0.46 for the roof and U = 0.38 for the ceiling.

An approximate value of *U* for the roof is equivalent to the summation of coefficients for each individual section times its percentage of total area

$U_r = (0.52 \times 0.15) + (0.46 \times 0.85) = 0.47$

Ratio of roof area to ceiling = 1560 ÷ 1200 = 1.3.

Substituting in Formula 6.

$$U = \frac{0.47 \times 0.38}{0.47 + \frac{0.38}{1.3}} = 0.235$$

$H = A\,U\,(t_i - t_o) = 1200 \times 0.235 \times 85 = 23{,}900$ Btu per hour.

Chapter 6

AIR LEAKAGE

Nature of Air Infiltration, Infiltration Through Walls, Window Leakage, Door Leakage, Selection of Wind Velocity, Crack Length used for Computations, Multi-Story Buildings, Heat Equivalent of Air Infiltration

AIR leakage losses are those resulting from the displacement of heated air in a building by unheated outside air, the interchange taking place through various apertures in the building, such as cracks around doors and windows, fireplaces and chimneys This leakage of air must be considered in heating and cooling calculations. (See Chapters 7 and 8.)

NATURE OF AIR INFILTRATION

The natural movement of air through building construction is due to two causes. One is the pressure exerted by the wind; the other is the difference in density of outside and inside air because of differences in temperature.

The wind causes a pressure to be exerted on one or two sides of a building. As a result, air comes into the building on the windward side through cracks or porous construction, and a similar quantity of air leaves on the leeward side through like openings. In general the resistance to air movement is similar on the windward to that on the leeward side. This causes a building up of pressure within the building and a lesser air leakage than that experienced in single wall tests as determined in the laboratory. It is assumed that actual building leakages owing to this building up of pressure will be 80 per cent of laboratory test values. While there are cases where this is not true, tests in actual buildings substantiate the factor for the general case. Tests on mechanically ventilated classrooms of average construction have shown that air infiltration acts quite independently of the planned air supply. Accordingly, the heating or cooling load owing to air infiltration from natural causes should be considered in addition to the ventilating load.

The air exchange owing to temperature difference, inside to outside, is not appreciable in low buildings. In tall, single story buildings with openings near the ground level and near the ceiling, this loss must be considered. Also in multi-storied buildings it is a large item unless the sealing between various floors and rooms is quite perfect. This temperature effect is a *chimney action*, causing air to enter through openings at lower levels and to leave at higher levels.

A complete study of all of the factors involved in air movement through building constructions would be very complex. Some of the complicating factors are: the variations in wind velocity and direction; the exposure of the building with respect to air leakage openings and with respect to adjoining buildings; the variations in outside temperatures as influencing the chimney effect; the relative area and resistance of openings on the windward and leeward sides and on the lower floors and on the upper floors; the influence of a planned air supply and the related outlet vents; and the variation from the average of individual building units. A study of infiltration points to the need for care in the obtaining of good building construction, or unnecessarily large heat losses will result.

INFILTRATION THROUGH WALLS

Table 1 gives data on infiltration through brick and frame walls. The brick walls listed in this table are walls which show poor workmanship and which are constructed of porous brick and lime mortar. For good workmanship, the leakage through hard brick walls with cement-lime mortar does not exceed one-third the values given. These tests indicate that plastering reduces the leakage by about 96 per cent; a heavy coat of cold water paint, 50 per cent; and 3 coats of oil paint carefully applied, 28 per cent. The infiltration through walls ranges from 6 to 25 per cent of that through windows and doors in a 10-story office building, with imperfect sealing of plaster at the baseboards of the rooms. With perfect sealing the range is from 0.5 to 2.7 per cent or a practically negligible quantity, which indicates the importance of good workmanship in proper sealing at the baseboard. It will be noted from Table 1, that the infiltration through properly plastered walls can be neglected.

TABLE 1. INFILTRATION THROUGH WALLS

Expressed in cubic feet per square foot per hour[a]

Type of Wall		Wind Velocity, Miles per Hour					
		5	10	15	20	25	30
8½ in. Brick Wall	Plain	1.75	4.20	7.85	12 2	18.6	22.9
	Plastered	0.017	0.037	0.066	0.107	0.161	0.236
13 in. Brick Wall	Plain	1 44	3.92	7.48	11 6	16.3	21.2
	Plastered	0.005	0.013	0 025	0.043	0.067	0.097
Frame Wall, with lath and plaster[b]		0.03	0.07	0 13	0.18	0.23	0.26

[a]The values given in this table are 20 per cent less than test values to allow for building up of pressure in rooms and are based on test data reported in the papers listed p 140

[b]Wall construction Bevel siding painted or cedar shingles sheathing, building paper, wood lath and 3 coats gypsum plaster

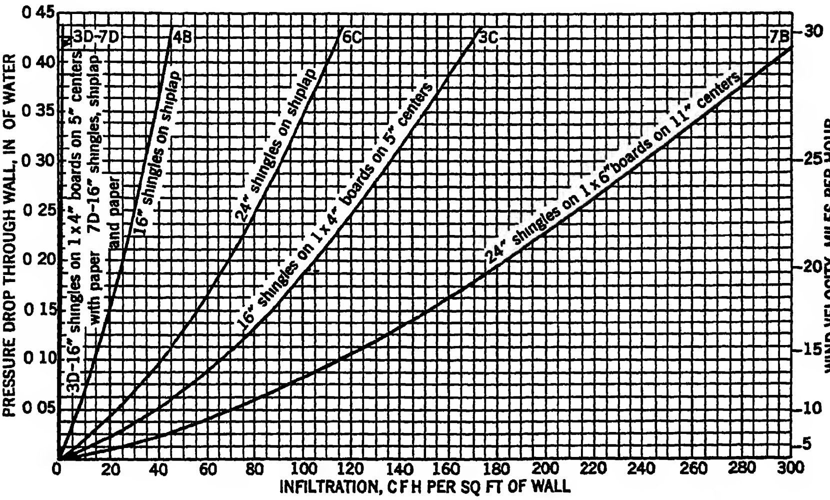


FIG. 1. INFILTRATION THROUGH VARIOUS TYPES OF SHINGLE CONSTRUCTION

The value of building paper when applied between sheathing and shingles is indicated by Fig. 1, which represents the effect on outside construction only, without lath and plaster. The effectiveness of plaster properly applied is no justification for the use of low grade building paper or of the poor construction of the wall containing it. Not only is it difficult to secure and maintain the full effectiveness of the plaster but also it is highly desirable to have two points of high resistance to air flow with an air space between them.

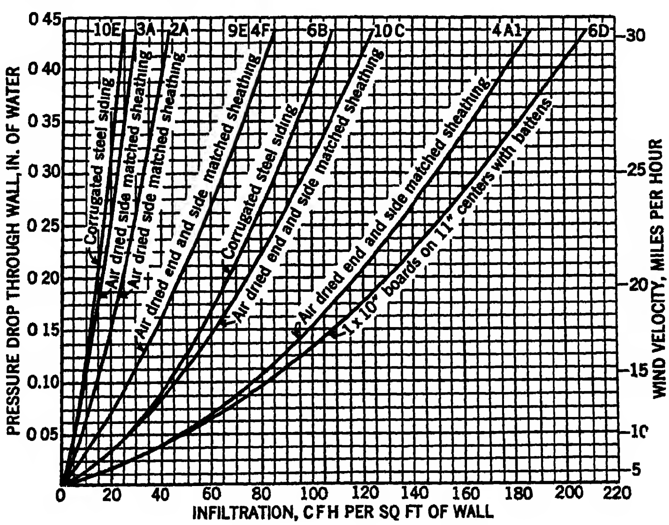


FIG. 2. INFILTRATION THROUGH SINGLE SURFACE WALLS USED IN FARM AND OTHER SHELTER BUILDINGS

The amount of infiltration that may be expected through simple walls used in farm and other shelter buildings, is shown in Fig. 2. The infiltration indicated in Figs. 1 and 2 is that determined in the laboratory and should be multiplied by the factor 0.80 to give proper working values.

WINDOW LEAKAGE

The amount of infiltration for various types of windows is given in Table 2. The fit of double-hung wood windows is determined by crack and clearance as illustrated in Fig. 3. The length of the perimeter opening or crack for a double-hung window is equal to three times the width plus two times the height, or in other words, it is the outer sash perimeter length plus the meeting rail length. Values of leakage shown in Table 2 for the average double-hung wood window were determined by setting the average measured crack and clearance found in a field survey of a large number of windows on nine windows tested in the laboratory. In

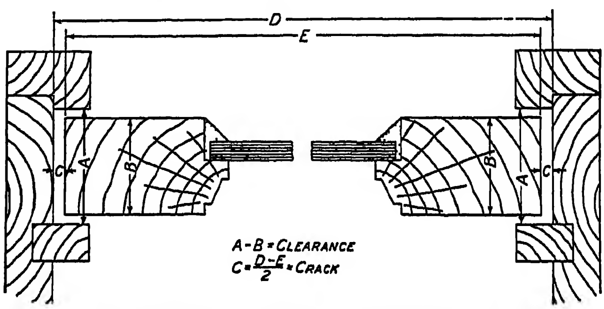


FIG 3. DIAGRAM ILLUSTRATING CRACK AND CLEARANCE

addition, the table gives figures for a poorly fitted window. All of the figures for double-hung wood windows are for the *unlocked* condition. Just how a window is closed, or fits when it is closed, has considerable influence on the leakage. The leakage will be high if the sash are short, if the meeting rail members are warped, or if the frame and sash are not fitted squarely to each other. It is possible to have a window with approximately the average crack and clearance that will have a leakage at least double that of the figures shown. Values for the average double-hung wood window in Table 2 are considered to be easily obtainable figures provided the workmanship on the window is good. Should it be known that the windows under consideration are poorly fitted, the larger leakage values should be used. Locking a window generally decreases its leakage, but in some cases may push the meeting rail members apart and increase the leakage. On windows with large clearances, locking will usually reduce the leakage.

Wood casement windows may be assumed to have the same unit leakage as for the average double-hung wood window when properly fitted. Locking, a normal operation in the closing of this type of window, maintains the crack at a low value.

For metal pivoted sash, the length of crack is the total perimeter of the movable or ventilating sections. Frame leakage on steel windows may be

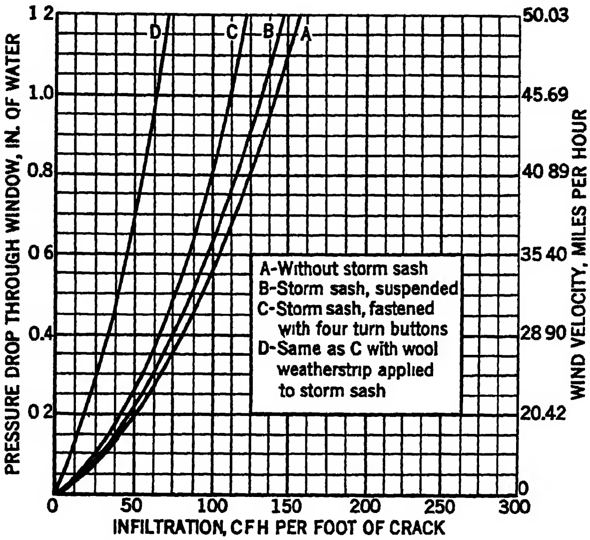


FIG. 4. INFILTRATION THROUGH SASH PERIMETER OF WINDOW WITH AND WITHOUT STORM SASH—5/64-IN. CRACK AND 3/32-IN CLEARANCE

neglected when they are properly grouted with cement mortar into brick work or concrete. When they are not properly sealed, the linear feet of sash section in contact with steel work at mullions should be figured at 25 per cent of the values for industrial pivoted windows as given in Table 2.

Leakage values for storm sash are given in Figs. 4 and 5. When storm

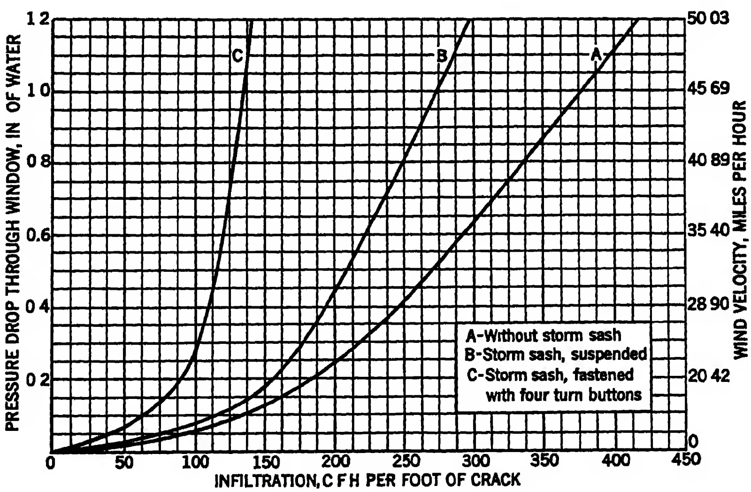


FIG. 5. INFILTRATION THROUGH SASH PERIMETER OF WINDOW WITH AND WITHOUT STORM SASH—1/8-IN. CRACK AND 1/8-IN. CLEARANCE

TABLE 2. INFILTRATION THROUGH WINDOWS

Expressed in Cubic Feet per Foot of Crack per Hour[a]

TYPE OF WINDOW	REMARKS	WIND VELOCITY, MILES PER HOUR					
		5	10	15	20	25	30
Double-Hung Wood Sash Windows (Unlocked)	Around frame in masonry wall—not calked[b]	3.3	8.2	14.0	20.2	27.2	34.6
	Around frame in masonry wall—calked[b]	0.5	1.5	2.6	3.8	4.8	5.8
	Around frame in wood frame construction[b]	2.2	6.2	10.8	16.6	23.0	30.3
	Total for average window, non-weather-stripped, 1/16-in crack and 3/64-in clearance[c] Includes wood frame leakage[d]	6.6	21.4	39.3	59.3	80.0	103.7
	Ditto, weatherstripped[d]	4.3	15.5	23.6	35.5	48.6	63.4
	Total for poorly fitted window, non-weather-stripped, 3/32-in crack and 3/32-in clearance[e] Includes wood frame leakage[d]	26.9	69.0	110.5	153.9	199.2	249.4
	Ditto, weatherstripped[d]	5.9	18.9	34.1	51.4	70.5	91.5
Double-Hung Metal Windows[f]	Non-weatherstripped, locked	20	45	70	96	125	154
	Non-weatherstripped, unlocked	20	47	74	104	137	170
	Weatherstripped, unlocked	6	19	32	46	60	76
Rolled Section Steel Sash Windows[k]	Industrial pivoted[g] 1/16-in crack	52	108	176	244	304	372
	Architectural projected, 1/32-in crack[h]	15	36	62	86	112	139
	Architectural projected, 3/64-in crack[h]	20	52	88	116	152	182
	Residential casement, 1/64-in crack[i]	6	18	33	47	60	74
	Residential casement, 1/32-in crack[i]	14	32	52	76	100	128
	Heavy casement section, projected, 1/64-in crack[j]	3	10	18	26	36	48
	Heavy casement section, projected 1/32-in crack[j]	8	24	38	54	72	92
Hollow Metal, vertically pivoted window[f]		30	88	145	186	221	242

[a]The values given in this table are 20 per cent less than test values to allow for building up of pressure in rooms, and are based on test data reported in the papers listed at the end of this chapter

[b]The values given for frame leakage are per foot of sash perimeter as determined for double-hung wood windows Some of the frame leakage in masonry walls originates in the brick wall itself and cannot be prevented by calking For the additional reason that calking is not done perfectly and deteriorates with time, it is considered advisable to choose the masonry frame leakage values for calked frames as the average determined by the calked and not-calked tests

[c]The fit of the average double-hung wood window was determined as 1/16-in crack and 3/64-in. clearance by measurements on approximately 600 windows under heating season conditions

[d]The values given are the totals for the window opening per foot of sash perimeter and include frame leakage and so-called *elsewhere* leakage The frame leakage values included are for wood frame construction but apply as well to masonry construction assuming a 50 per cent efficiency of frame calking

[e]A 3/32-in crack and clearance represents a poorly fitted window, much poorer than average

[f]Windows tested in place in building

[g]Industrial pivoted window generally used in industrial buildings Ventilators horizontally pivoted at center or slightly above, lower part swinging out

[h]Architectural projected made of same sections as industrial pivoted except that outside framing member is heavier, and refinements in weathering and hardware Used in semi-monumental buildings such as schools Ventilators swing in or out and are balanced on side arms 1/32-in crack is obtainable in the best practice of manufacture and installation, 3/64-in crack considered to represent average practice

[i]Of same design and section shapes as so-called *heavy section casement* but of lighter weight 1/64-in crack is obtainable in the best practice of manufacture and installation, 1/32-in crack considered to represent average practice

[j]Made of heavy sections Ventilators swing in or out and stay set at any degree of opening. 1/64-in crack is obtainable in the best practice of manufacture and installation, 1/32-in. crack considered to represent average practice.

[k]With reasonable care in installation, leakage at contacts where windows are attached to steel framework and at mullions is negligible With 3/64-in crack, representing poor installation, leakage at contact with steel framework is about one-third, and at mullions about one-sixth of that given for industrial pivoted windows in the table

sash are applied to well fitted windows, very little reduction in infiltration is secured, but the application of the sash does give an air space which reduces the heat transmission and helps prevent the frosting of the windows. When storm sash are applied to poorly fitted windows, a reduction in leakage of 50 per cent may be secured.

DOOR LEAKAGE

Doors vary greatly in fit because of their large size and tendency to warp. For a well fitted door, the leakage values for a poorly fitted double-hung wood window may be used. If poorly fitted, twice this figure should be used. If weatherstripped, the values may be reduced one-half. A single door which is frequently opened, such as might be found in a store, should have a value applied which is three times that for a well fitted door. This extra allowance is for opening and closing losses and is kept from being greater by the fact that doors are not used as much in the coldest and windiest weather.

In the case of infiltration through swinging and revolving doors engineers are not in full agreement at the present time regarding the allowances for cooling load determinations. Some references to recently published data are given at the end of this chapter.

SELECTION OF WIND VELOCITY

Although all authorities do not agree upon the value of the wind velocity that should be chosen for any given locality, it is common engineering practice to use the average wind velocity during the three coldest months of the year. Until this point is definitely established the practice of using average values will be followed. Average wind velocities for the months of December, January and February for various cities in the United States and Canada are given in Table 2, Chapter 7.

In considering both the transmission and infiltration losses, the more exact procedure would be to select the outside temperature and the wind velocity corresponding thereto, based on Weather Bureau records, which would result in the maximum heat demand. Since the proportion of transmission and infiltration losses varies with the construction and is different for every building, the proper combination of temperature and wind velocity to be selected would be different for every type of building, even in the same locality. Furthermore, such a procedure would necessitate a laborious cut-and-try process in every case in order to determine the worst combination of conditions for the building under consideration. It would also be necessary to consider heat lag due to heat capacity in the case of heavy masonry walls, and other factors, to arrive at the most accurate solution of the problem. Although heat capacity should be considered wherever possible, it is seldom possible to accurately determine the worst combination of outside temperature and wind velocity for a given building and locality. The usual procedure, as already explained, is to select an outside temperature based on the lowest on record and the average wind velocity during the months of December, January and February.

The direction of prevailing winds may usually be included within an

angle of about 90 deg. The windows that are to be figured for prevailing and non-prevailing winds will ordinarily each occupy about one-half the perimeter of the structure, the proportion varying to a considerable extent with the plan of the structure. (See discussion of wind movement in Chapter 4.)

CRACK LENGTH USED FOR COMPUTATIONS

In no case should the amount of crack used for computation be less than half of the total crack in the outside walls of the room. Thus, in a room with one exposed wall, take all the crack; with two exposed walls, take the wall having the most crack; and with three or four exposed walls, take the wall having the most crack; but in no case take less than half the total crack. For a building having no partitions, whatever wind enters through the cracks on the windward side must leave through the cracks on the leeward side. Therefore, take one-half the total crack for computing each side and end of the building.

Table 3 Air Changes Taking Place under Average Conditions Exclusive of Air Provided for Ventilation

Kind of Room or Building	Number of Air Changes Taking Place per Hour
Rooms, 1 side exposed	1
Rooms, 2 sides exposed	1½
Rooms, 3 sides exposed	2
Rooms, 4 sides exposed	2
Rooms with no windows or outside doors	½ to ¾
Entrance Halls	2 to 3
Reception Halls	2
Living Rooms	1 to 2
Dining Rooms	1 to 2
Bath Rooms	2
Drug Stores	2 to 3
Clothing Stores	1
Churches, Factories, Lofts, etc.	½ to 3

The amount of air leakage is sometimes roughly estimated by assuming a certain number of air changes per hour for each room, the number of changes assumed being dependent upon the type, use and location of the room, as indicated in Table 3. This method may be used to advantage as a check on the calculations made in the more exact manner.

MULTI-STORY BUILDINGS

In tall buildings, infiltration may be considerably influenced by temperature difference or chimney effect which will operate to produce a head that will add to the effect of the wind at lower levels and subtract from it at higher levels. On the other hand, the wind velocity at lower levels may be somewhat abated by surrounding obstructions. Furthermore, the chimney effect is reduced in multi-story buildings by the partial isolation of floors preventing free upward movement, so that wind and temperature difference may seldom cooperate to the fullest extent.

Making the rough assumption that the *neutral zone* is located at mid-height of a building, and that the temperature difference is 70 F, the following formulae may be used to determine an equivalent wind velocity to be used in connection with Tables 1 and 2 that will allow for both wind velocity and temperature difference:

$$M_e = \sqrt{M^2 - 1.75\,a} \tag{1}$$

$$M_e = \sqrt{M^2 + 1.75\,b} \tag{2}$$

where

M_e = equivalent wind velocity to be used in conjunction with Tables 1 and 2.

M = wind velocity upon which infiltration would be determined if temperature difference were disregarded.

a = distance of windows under consideration from mid-height of building if *above* mid-height.

b = distance if *below* mid-height.

The coefficient 1.75 allows for about one-half the temperature difference head.

For buildings of unusual height, Equation 1 would indicate negative infiltration at the highest stories, which condition may, at times, actually exist.

Sealing of Vertical Openings

In tall, multi-story buildings, every effort should be made to seal off vertical openings such as stair-wells and elevator shafts from the remainder of the building. Stair-wells should be equipped with self-closing doors, and in exceptionally high buildings, should be closed off into sections of not over 10 floors each. Plaster cracks should be filled. Elevator enclosures should be tight and solid doors should be used.

If the sealing of the vertical openings is made effective, no allowance need be made for the chimney effect. Instead, the greater wind movement at the high altitudes makes it advisable to install additional heating surface on the upper floors above the level of neighboring buildings, this additional surface being increased as the height is increased. One arbitrary rule is to increase the heating surface on floors above neighboring buildings by an amount ranging from 5 per cent to 20 per cent. This extra heating surface is required only on the windward side and on windy days, and hence automatic temperature control is especially desirable with such installations.

Heating Surface for Stair-Wells

In stair-wells that are open through many floor levels although closed off from the remainder of each floor by doors and partitions, the stratification of air makes it advisable to increase the amount of heating surface at the lower levels and to decrease the amount at higher levels even to the point of omitting all heating surface on the top several floor levels. One rule is to calculate the heating surface of the entire stair-well in the usual way and to place 50 per cent of this in the bottom third, the normal amount in the middle third and the balance in the top third.

HEAT EQUIVALENT OF AIR INFILTRATION

The heat required to warm cold outside air, which enters a room by infiltration, to the temperature of the room is given by the following equation:

$$H_1 = 0.24\ Q\ d\ (t - t_0) \tag{3}$$

where

H_1 = Btu per hour required for heating air leaking into building from outside temperature t_0 to inside temperature t.

Q = cubic feet of air entering per hour at inside temperature t.

d = density (pounds per cubic foot) of air at inside temperature t.

t = inside temperature at the proper level.

t_0 = outside air temperature for which heating system is designed.

0.24 = specific heat of air.

It is sufficiently accurate to take d = 0.075 lb, in which case the equation reduces to

$$H_1 = 0.018\ Q\ (t - t_0) \tag{4}$$

While a heating reserve must be provided to warm inleaking air on the windward side of a building, this does not necessarily mean that the heating plant must be provided with a reserve capacity, since the inleaking air, warmed at once by adequate heating surface in exposed rooms, will move transversely and upwardly through the building, thus relieving other radiators of a part of their load. The actual loss of heat of a building caused by infiltration is not to be confused with the necessity for providing additional heating capacity for a given space. Infiltration is a disturbing factor in the heating of a building, and its maximum effect (maximum in the sense of an average of wind velocity peaks during the heating season above some reasonably chosen minimum) must be met by a properly distributed reserve of heating capacity, which reserve, however, is not in use at all places at the same time, nor in any one place at all times.

REFERENCES

Air Leakage, by F. C. Houghten and C. C. Schrader (A.S.H.V.E. TRANSACTIONS, Vol. 30, 1924).

Air Leakage around Window Openings, by C. C. Schrader (A.S.H.V E. TRANSACTIONS, Vol. 30, 1924).

Neutral Zone in Ventilating, by J E. Emswiler (A.S.H V.E. TRANSACTIONS, Vol. 32, 1926)

Infiltration through Plastered and Unplastered Brick Walls, by F C. Houghten and Margaret Ingels (A.S.H.V.E. TRANSACTIONS, Vol. 33, 1927).

Effect of Frame Calking and Storm Sash on Infiltration around and through Windows, by W. M. Richtmann and C. Braatz (A S.H.V.E. TRANSACTIONS, Vol 34, 1928).

Air Leakage on Metal Windows in a Modern Office Building, by F. C. Houghten and M. E. O'Connell (A.S.H.V.E. TRANSACTIONS, Vol. 34, 1928).

The Weathertightness of Rolled Section Steel Windows, by J. E. Emswiler and W. C. Randall (A.S.H.V E. TRANSACTIONS, Vol 34, 1928).

Air Leakage through a Pivoted Metal Window, by F. C. Houghten and M E O'Connell (A.S.H.V.E. TRANSACTIONS, Vol. 34, 1928).

Air Infiltration through Various Types of Brick Wall Construction, by G. L. Larson, D. W. Nelson and C. Braatz (A S.H.V.E TRANSACTIONS, Vol. 35, 1929).

Pressure Difference across Windows in Relation to Wind Velocity, by J. E. Emswiler and W C. Randall (A.S H.V.E. TRANSACTIONS, Vol 36, 1930)

Air Infiltration Through Various Types of Wood Frame Construction, by G L. Larson, D W Nelson and C. Braatz (A.S H V E. TRANSACTIONS, Vol 36, 1930)

Air Infiltration Through Double-Hung Wood Windows, by G L. Larson, D W. Nelson and R. W. Kubasta (A S.H V.E. TRANSACTIONS, Vol. 37, 1931).

Flue Action in Tall Buildings, by H. L Alt (*Heating, Piping and Air Conditioning*, May, 1932)

Air Infiltration Through Steel Framed Windows, by D. O. Rusk, V H. Cherry and L Boelter (*Heating, Piping and Air Conditioning*, October, 1932)

Investigation of Air Outlets in Class Room Ventilation, by G. L. Larson, D. W. Nelson, and R. W. Kubasta (A S.H.V.E. TRANSACTIONS, Vol 38, 1932).

Influence of Stack Effect on the Heat Loss in Tall Buildings, by Axel Marin (A.S H. V.E. TRANSACTIONS, Vol. 40, 1934).

Wind Velocities Near a Building and Their Effect on Heat Loss, by F C. Houghten, J. L Blackshaw, and Carl Gutberlet (A S H.V E TRANSACTIONS, Vol. 40, 1934)

Fuel Saving Resulting from the Use of Storm Windows and Doors, by A P Kratz and S Konzo (A S H V.E JOURNAL SECTION, *Heating, Piping and Air Conditioning*, December, 1935)

The Infiltration Problem of Multiple Entrances, by A M. Simpson and K B Atkinson (A S H.V E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, June, 1936)

Infiltration Characteristics of Entrance Doors, by A. M. Simpson (*Refrigerating Engineering*, June, 1936).

PROBLEMS IN PRACTICE

1 ● Why is it important in exterior walls of air space construction to place a wind stop in the outer surface?

During the heating season, air within the space is at a higher temperature than outdoor air. If cracks are present in the outer surface, a chimney action takes place which causes a continuous change of air in the space This causes a greater heat loss through the inner wall surface for two reasons: (1) the temperature difference becomes greater and (2) the surface coefficient on the air space side is increased because of the higher air velocity. Another undesirable condition resulting from this leakage is a lowered temperature of the inside surface. The increased radiation from occupants to walls caused by this condition must be offset by higher air temperatures.

The wind stop in the outer wall surface is therefore important because:

a. It reduces the loss of heat through the wall.

b. It helps to maintain a high inner wall surface temperature.

2 ● Why is it essential to consider this in heating calculations?

The inflowing air displaces inside heated air and must be heated up to the internal temperature.

3 ● Where is it necessary to consider infiltration created by temperature difference?

In tall, single-story buildings and in multi-story buildings where the floors are not adequately isolated.

4 ● Why is the infiltration in a building less than that determined in laboratory tests?

In laboratory tests, the indicated wind velocity is measured by the difference in pressure on the two sides of a single wall, window, or object tested. In a building, an internal back pressure is built up between its walls to a point where outflow on the lee side is equal to inflow on the windward side and this back pressure reduces the actual inflow below that determined in the laboratory for a comparable wind.

5 ● Is heat loss by infiltration through walls of importance?

Only in the case of simple walls or poorly constructed compound walls.

6 ● What measurements are required to calculate the heat loss through double-hung wood windows?

Sash crack (equal to the sash perimeter plus the meeting rail) and frame crack (equal to the frame perimeter).

7 ● What is the basis for selecting the wind velocity and outside temperature to be used in making infiltration calculations?

Weather Bureau records. The wind velocity taken is the average during the three coldest months and the temperature used is the lowest on record for the given locality.

8 ● How does the temperature difference influence the heat loss in a tall building?

The chimney effect caused by the temperature difference operates to produce a head that will add to the effect of the wind at lower levels and subtract from it at higher levels.

9 ● For a wind velocity of 15 mph and a building 180 ft high, calculate the effective wind velocity at the ground floor and at a height of 150 ft.

a. At the ground floor the effective wind velocity would be

$$M_e = \sqrt{15^2 + 1.75 \times 90} = 19.6 \text{ mph}$$

b. At a floor 150 ft above the ground

$$M_e = \sqrt{15^2 - 1.75 \times 60} = 11.0 \text{ mph}$$

10 ● A room contains three 2 ft-8 in. by 5 ft-6 in. plain double-hung wood windows with 1/16-in. crack and 3/64-in. clearance. Assume a wind velocity of 15 mph and a temperature difference of 75 F. Neglecting chimney effect, what is the maximum heat loss due to infiltration?

From Table 2, infiltration per foot of crack is 39.3 cu ft. Length of crack for the three windows is 57 ft. The maximum heat loss, due to infiltration, is equal to 0.018 × 39.3 × 57 × 75 or 3020 Btu per hour.

11 ● Find the infiltration through a wall with 16-in. shingles on 1 in. by 4 in. boards with 20 mph wind velocity. Give the pressure drop through the wall.

Referring to Curve 3C, Fig. 1, the value on the horizontal scale corresponding to 20 mph is approximately 102 cfh per square foot of wall.

The pressure drop through the wall is 0.193 in. of water (see left hand vertical scale).

12 ● What will be the infiltration through air-dried end and side-matched sheathing for 15 mph wind velocity?

Referring to Curve 10C, Fig. 2, the value on the horizontal scale corresponding to 15 mph is 50 cfh per square foot of wall.

13 ● From Table 2, find the infiltration (cubic feet per hour per foot of crack) for an average double-hung window, not weather stripped, with a 20 mph wind velocity.

59.3 cu ft per foot of crack per hour.

14 ● Using the value found in Question 13, what will be the heat requirement in a building with a total crack (all windows and doors) of 180 ft if the wind velocity is 15 mph, the outside temperature is 0 F, and the inside temperature is 70 F?

Using one half of the total crack, the volume of air is:

$$90 \times 59.3 = 5337 \text{ cu ft}$$

$$H = 0.018 \times 5337 \times (70 - 0) = 6724.6 \text{ Btu.} \quad \text{(See Equation 4.)}$$

Chapter 7

HEATING LOAD

Heat Demand Design Factors, Method of Procedure, Inside and Outside Temperatures, Wind Velocity Effects, Auxiliary Heat Sources, Wall Surface Condensation, Heat Loss Computation Example

TO design any system of heating, the maximum probable heat demand must be accurately estimated in order that the apparatus installed shall be capable of maintaining the desired temperature at all times. The factors which govern this maximum heat demand—most of which are seldom, if ever, in equilibrium—include the following:

1. Outside temperature 2. Rain or snow. 3. Sunshine or cloudiness. 4. Wind velocity.	*Outside Conditions (The Weather)*
5. Heat transmission of exposed parts of building. 6. Infiltration of air through cracks, crevices and open doors and windows 7. Heat capacity of materials. 8. Rate of absorption of solar radiation by exposed materials.	*Building Construction*
9. Inside temperatures. 10. Stratification of air. 11. Type of heating system 12. Ventilation requirements. 13. Period and nature of occupancy. 14. Temperature regulation.	*Inside Conditions*

The *inside conditions* vary from time to time, the physical properties of the *building construction* may change with age, and the *outside conditions* are changing constantly. Just what the worst combination of all of these variable factors is likely to be in any particular case is therefore conjectural. Because of the nature of the problem, extreme precision in estimating heat losses at any time, while desirable, is hard of attainment.

The procedure to be followed in determining the heat loss from any building can be divided into seven consecutive steps, as follows:

1. Determine on the inside air temperature, at the breathing line or the 30-in. line, which is to be maintained in the building during the coldest weather. (See Table 1.)

2. Determine on an outside air temperature for design purposes, based on the minimum temperatures recorded in the locality in question, which will provide for all but the most severe weather conditions. Such conditions as may exist for only a few consecutive hours are readily taken care of by the heat capacity of the building itself (See Table 2)

3. Select or compute the heat transmission coefficients for outside walls and glass; also for inside walls, floors, or top-floor ceilings, if these are next to unheated space; include roof if next to heated space. (See Chapter 5.)

4. Measure up net outside wall, glass and roof next to heated spaces, as well as any cold walls, floors or ceilings next to unheated space. Such measurements are made from building plans, or from the actual building.

5. Compute the heat transmission losses for each kind of wall, glass, floor, ceiling and roof in the building by multiplying the heat transmission coefficient in each case by the area of the surface in square feet and the temperature difference between the inside and outside air. (See Items 1 and 2.)

6. Select unit values and compute the heat equivalent of the infiltration of cold air taking place around outside doors and windows. These unit values depend on the kind or width of crack and wind velocity, and when multiplied by the length of crack and the temperature difference between the inside and outside air, the result expresses the heat required to warm up the cold air leaking into the building per hour. (See Chapter 6.)

7. The sum of the heat losses by transmission (Item 5) through the outside wall and glass, as well as through any cold floors, ceilings or roof, plus the heat equivalent (Item 6) of the cold air entering by infiltration represents the total heat loss equivalent for any building

Item 7 represents the heat losses after the building is heated and under stable operating conditions in coldest weather. Additional heat is required for raising the temperature of the air, the building materials and the material contents of the building to the specified standard inside temperature.

The rate at which this additional heat is required depends upon the heat capacity of the structure and its material contents and upon the time in which these are to be heated.

This additional heat may be figured and allowed for as conditions re-

TABLE 1. WINTER INSIDE DRY-BULB TEMPERATURES USUALLY SPECIFIED[a]

Type of Building	Deg Fahr	Type of Building	Deg Fahr
SCHOOLS		THEATERS—	
Class rooms	70–72	Seating space	68–72
Assembly rooms	68–72	Lounge rooms	68–72
Gymnasiums	55–65	Toilets	68
Toilets and baths	70	HOTELS—	
Wardrobe and locker rooms	65–68	Bedrooms and baths	70
Kitchens	66	Dining rooms	70
Dining and lunch rooms	65–70	Kitchens and laundries	66
Playrooms	60–65	Ballrooms	65–68
Natatoriums	75	Toilets and service rooms	68
HOSPITALS—			
Private rooms	70–72	HOMES	70–72
Private rooms (surgical)	70–80	STORES	65–68
Operating rooms	70–95	PUBLIC BUILDINGS	68–72
Wards	68	WARM AIR BATHS	120
Kitchens and laundries	66	STEAM BATHS	110
Toilets	68	FACTORIES AND MACHINE SHOPS	60–65
Bathrooms	70–80	FOUNDRIES AND BOILER SHOPS	50–60
		PAINT SHOPS	80

[a]The most comfortable dry-bulb temperature to be maintained depends on the relative humidity and air motion These three factors considered together constitute what is termed the *effective temperature*. See Chapter 3.

quire, but inasmuch as the heating system proportioned for taking care of the heat losses will usually have a capacity about 100 per cent greater than that required for average winter weather, and inasmuch as most buildings may either be continuously heated or have more time allowed for heating-up during the few minimum temperature days, no allowance is made except in the size of boilers or furnaces.

INSIDE TEMPERATURES

The inside air temperature which must be maintained within a building and which should always be stated in the heating specifications is understood to be the dry-bulb temperature at the breathing line, 5 ft above the floor, or the 30-in. line, and not less than 3 ft from the outside walls. Inside air temperatures, usually specified, vary in accordance with the use to which the building is to be put and Table 1 presents values which conform with good practice.

The proper dry-bulb temperature to be maintained depends upon the relative humidity and air motion, as explained in Chapter 3. In other words, a person may feel warm or cool at the same dry-bulb temperature, depending on the relative humidity and air motion. The optimum winter *effective temperature* for sedentary persons, as determined at the A.S.H.V.E. Research Laboratory, is 66 deg.[1]

According to Fig. 6, Chapter 3, for so-called still air conditions, a relative humidity of approximately 50 per cent is required to produce an effective temperature of 66 deg when the dry-bulb temperature is 70 F. However, even where provision is made for artificial humidification, the relative humidity is seldom maintained higher than 40 per cent during the extremely cold weather, and where no provision is made for humidification, the relative humidity may be 20 per cent or less. Consequently, in using the figures given in Table 1, consideration should be given to whether provision is to be made for humidification, and if so, the actual relative humidity to be maintained.

Temperature at Proper Level: In making the actual heat-loss computations, however, for the various rooms in a building it is often necessary to modify the temperatures given in Table 1 so that the air temperature at the proper level will be used. By *air temperature at the proper level* is meant, in the case of walls, the air temperature at the mean height between floor and ceiling; in the case of glass, the air temperature at the mean height of the glass; in the case of roof or ceiling, the air temperature at the mean height of the roof or ceiling above the floor of the heated room; and in the case of floors, the air temperature at the floor level. In the case of heated spaces adjacent to unheated spaces, it will usually be sufficient to assume the temperature in such spaces as the mean between the temperature of the inside heated spaces and the outside air temperature, excepting where the combined heat transmission coefficient of the roof and ceiling can be used, in which case the usual inside and outside temperatures should be applied. (See discussion regarding the use of combined coefficients of pitched roofs, unheated attics and top-floor ceilings Chapter 5.)

[1]See Chapter 3, p 67

High Ceilings: Research data concerning stratification of air in buildings are lacking, but in general it may be said that where the increase in temperature is due to the natural tendency of the warmer or less dense air to rise, as where a direct radiation system is installed, the temperature of the air at the ceiling increases with the ceiling height. The relation, however, is not a straight-line function, as the amount of increase per foot of height apparently decreases as the height of the ceiling increases, according to present available information[2].

Where ceiling heights are under 20 ft, it is common engineering practice to consider that the Fahrenheit temperature increases 2 per cent for each foot of height above the breathing line. This rule, sufficiently accurate for most cases, will give the probable air temperature at any given level for a room heated by direct radiation. Thus, the probable temperature in a room at a point 3 ft above the breathing line, if the breathing line temperature is 70 F, will be

$$(1.00 + 3 \times .02)\ 70 = 74.2 \text{ F.}$$

With certain types of heating and ventilating systems, which tend to oppose the natural tendency of warm air to rise, the temperature differential between floor and ceiling can be greatly reduced. These include unit heaters, fan-furnace heaters, and the various types of mechanical ventilating systems. The amount of reduction is problematical in certain instances, as it depends upon many factors such as location of heaters, air temperature, and direction and velocity of air discharge. In some cases it has been possible to reduce the temperature between the floor and ceiling by a few degrees, whereas, in other cases, the temperature at the ceiling has actually been increased because of improper design, installation or operation of equipment. So much depends upon the factors enumerated that it is not advisable to allow less than 1 per cent per foot (and usually more) above the breathing line in arriving at the air temperature at any given level for any of these types of heating and ventilating systems, unless the manufacturers are willing to guarantee that the particular type of equipment under consideration will maintain a smaller temperature differential for the specific conditions involved.

Temperature at Floor Level: In determining mean air temperatures just above floors which are next to ground or unheated spaces, a temperature 5 deg lower than the breathing-line temperature may be used, provided the breathing-line temperature is not less than 55 F.

OUTSIDE TEMPERATURES

The outside temperature used in computing the heat loss from a building is seldom taken as the lowest temperature ever recorded in a given locality. Such temperatures are usually of short duration and are rarely repeated in successive years. It is therefore evident that a temperature somewhat higher than the lowest on record may be properly assumed in making the heat-loss computations.

---

[2]Temperature Gradient Observations in a Large Heated Space, by G L. Larson, D W Nelson and O C. Cromer (A.S H V E. Transactions, Vol 39, 1933)

Tests of Three Heating Systems in an Industrial Type of Building, by G L. Larson, D. W. Nelson and John James (A S H V E. Journal Section, *Heating, Piping and Air Conditioning*, November, 1934).

The outside temperature to be assumed in the design of any heating system is ordinarily not more than 15 deg above the lowest recorded temperature as reported by the Weather Bureau during the preceding 10 years for the locality in which the heating system is to be installed. In the case of massive and well insulated buildings in localities where the minimum does not prevail for more than a few hours, or where the lowest recorded temperature is extremely unusual, more than 15 deg above the minimum may be allowed, due primarily to the *fly-wheel* effect of the heat capacity of the structure. The outside temperature assumed and used in the design should always be stated in the heating specifications. Table 2 lists the coldest dry-bulb temperatures ever recorded by the Weather Bureau at the places listed.

If Weather Bureau reports are not available for the locality in question, then the reports for the station nearest to this locality are to be used, unless some other temperature is specifically stated in the specifications. In computing the average heat transmission losses for the heating season in the United States the average outside temperature from October 1 to May 1 should be used.

WIND VELOCITY EFFECTS

The effect of wind on the heating requirements of any building should be given consideration under two heads:

1. Wind movement increases the heat transmission of walls, glass, and roof, affecting poor walls to a much greater extent than good walls

2 Wind movement materially increases the infiltration (inleakage) of cold air through the cracks around doors and windows, and even through the building materials themselves, if such materials are at all porous.

Theoretically as a basis for design, the most unfavorable combination of temperature and wind velocity should be chosen. It is entirely possible that a building might require more heat on a windy day with a moderately low outside temperature than on a quiet day with a much lower outside temperature. However, the combination of wind and temperature which is the worst would differ with different buildings, because wind velocity has a greater effect on buildings which have relatively high infiltration losses. It would be possible to work out the heating load for a building for several different combinations of temperature and wind velocity which records show to have occurred and to select the worst combination; but designers generally do not feel that such a degree of refinement is justified. Therefore, pending further studies of actual buildings, it is recommended that the average wind movement in any locality during December, January and February be provided for in computing (1) the heat transmission of a building, and (2) the heat required to take care of the infiltration of outside air.

The first condition is readily taken care of, as explained in Chapter 5, by using a surface coefficient f_0 for the outside wall surface which is based on the proper wind velocity. In case specific data are lacking for any given locality, it is sufficiently accurate to use an average wind velocity of approximately 15 mph which is the velocity upon which the heat transmission coefficient tables in Chapter 5 are based.

In a similar manner, the heat allowance for infiltration through cracks

TABLE 2. CLIMATIC CONDITIONS COMPILED FROM WEATHER BUREAU RECORDS[a]

COL A	COL B	COL C	COL D	COL E	COL F
State	City	Average Temp, Oct 1st–May 1st	Lowest Temperature Ever Reported	Average Wind Velocity Dec, Jan, Feb, Miles per Hour	Direction of Prevailing Wind, Dec, Jan, Feb
Ala.	Mobile	58.9	−1	10 4	N
	Birmingham	53.8	−10	8 5	N
Ariz.	Phoenix	59.5	12	6.4	E
	Flagstaff	35.8	−25	7 8	SW
Ark.	Fort Smith	50.4	−15	8.1	E
	Little Rock	51 6	−12	8.7	NW
Calif.	San Francisco	54.2	27	7.6	N
	Los Angeles	58.5	28	6.3	NE
Colo.	Denver	38.9	−29	7.5	S
	Grand Junction	38.9	−21	5.3	NW
Conn	New Haven	38.4	−15	9 7	N
D C.	Washington	43.4	−15	7.1	NW
Fla.	Jacksonville	62.0	10	9.2	NE
Ga.	Atlanta	51.5	−8	12.1	NW
	Savannah	58.5	8	9 5	NW
Idaho	Lewiston	42.3	−23	5.3	E
	Pocatello	35.7	−28	9 6	SE
Ill.	Chicago	36.4	−23	12 5	W
	Springfield	39.8	−24	10.1	NW
Ind.	Indianapolis	40.3	−25	11.5	SW
	Evansville	45.1	−16	9 8	S
Iowa	Dubuque	33.9	−32	7.1	NW
	Sioux City	32.6	−35	11.6	NW
Kans.	Concordia	39.8	−25	8.1	S
	Dodge City	41.4	−26	9 8	NW
Ky.	Louisville	45.3	−20	9.9	SW
La.	New Orleans	61.6	7	8.8	N
	Shreveport	56.2	−5	8 9	SE
Me.	Eastport	31.5	−23	12 0	W
	Portland	33.8	−21	9.2	NW
Md.	Baltimore	43 8	−7	7.8	NW
Mass.	Boston	38.1	−18	11.2	W
Mich.	Alpena	29.6	−28	12.4	W
	Detroit	35.8	−24	12.7	SW
	Marquette	28 3	−27	11.1	NW
Minn	Duluth	24.3	−41	12.6	SW
	Minneapolis	29.4	−33	11.3	NW
Miss.	Vicksburg	56.8	−1	8.3	SE
Mo.	St. Joseph	40.7	−24	9.3	NW
	St Louis	43.6	−22	11.6	S
	Springfield	44.3	−29	10.8	SE
Mont.	Billings	34.0	−49	....	W
	Havre	27.6	−57	9.5	SW
Nebr.	Lincoln	37.0	−29	10.5	S
	North Platte	35.4	−35	8 5	W
Nev.	Tonopah	39.4	−10	10.0	SE
	Winnemucca	37.9	−28	8.7	NE
N. H.	Concord	33.3	−35	6.6	NW
N. J.	Atlantic City	41.6	−9	15.9	NW
N Y.	Albany	35.2	−24	8.1	S
	Buffalo	34.8	−20	17 2	W
	New York	40.7	−14	17.1	NW

[a]U S. data from U. S. Weather Bureau
Canadian data from Meteorological Service of Canada.

TABLE 2. CLIMATIC CONDITIONS COMPILED FROM WEATHER BUREAU RECORDS[a]—(Continued)

COL. A	COL. B	COL. C	COL. D	COL. E	COL. F
State or Province	City	Average Temp, Oct 1st–May 1st	Lowest Temperature Ever Reported	Average Wind Velocity Dec., Jan., Feb., Miles per Hour	Direction of Prevailing Wind, Dec., Jan., Feb
N M	Santa Fe	38 3	−13	7.8	NE
N C.	Raleigh	50.0	−2	8.2	SW
	Wilmington	54 2	5	8.5	SW
N. Dak	Bismarck	24.6	−45	9.1	NW
	Devils Lake	20 3	−44	10.6	W
Ohio	Cleveland	37.2	−17	13.0	SW
	Columbus	39 9	−20	12.0	SW
Okla.	Oklahoma City	47.9	−17	12.0	N
Oreg.	Baker	35 2	−24	6.9	SE
	Portland	46.1	−2	7.5	S
Pa	Philadelphia	42 7	−6	11.0	NW
	Pittsburgh	41.0	−20	11.7	W
R. I	Providence	37.2	−17	12.8	NW
S. C.	Charleston	57.4	7	10.6	SW
	Columbia	54 0	−2	8.1	NE
S Dak.	Huron	28.2	−43	10 6	NW
	Rapid City	33 4	−34	8.2	W
Tenn.	Knoxville	47.9	−16	7.8	SW
	Memphis	51.1	−9	9.7	S
Texas	El Paso	53.5	−5	10.4	NW
	Fort Worth	55.2	−8	10.4	NW
	San Antonio	60.6	4	8.0	NE
Utah	Modena	36 3	−24	8.8	W
	Salt Lake City	40.0	−20	6.7	SE
Vt.	Burlington	31.5	−29	11.8	S
Va	Norfolk	49 3	2	12.5	N
	Lynchburg	46.8	−7	7.1	NW
	Richmond	47.0	−3	7.9	SW
Wash.	Seattle	44.8	3	11.3	SE
	Spokane	37 7	−30	7.1	SW
W. Va.	Elkins	39.4	−28	6.6	W
	Parkersburg	42 6	−27	7 5	SW
Wis.	Green Bay	30 0	−36	10.4	SW
	La Crosse	31 7	−43	7 3	S
	Milwaukee	33 4	−25	11.5	W
Wyo	Sheridan	30.7	−41	6.0	NW
	Lander	30.0	−40	5 0	SW
Alta.	Edmonton	23.0	−57	6.5	SW
B. C.	Victoria	43 9	− 1.5	12.5	N
	Vancouver	42 0	2	4.5	E
Man	Winnipeg	17.5	−47	10.0	NW
N. B	Fredericton	27.0	−35	9.6	NW
N S.	Yarmouth	35.0	−12	14.2	NW
Ont	London	32.6	−27	10.3	SW
	Ottawa	26.5	−34	8.4	NW
	Port Arthur	22 4	−37	7.8	NW
	Toronto	32 9	−26.5	13.0	SW
P. E I.	Charlottetown	29.0	−27	9.4	SW
Que.	Montreal	27.8	−29	14.3	SW
	Quebec	24.2	−34	13.6	SW
Sask	Prince Albert	15.8	−70	5.1	W
Yukon	Dawson	2.1	−68	3.7	S

[a]U S data from U. S. Weather Bureau.
Canadian data from Meteorological Service of Canada.

and walls (Tables 1 and 2, Chapter 6) must be based on the proper wind velocity for a given locality. In the case of *tall buildings* special attention must be given to infiltration factors. (See Chapter 6).

In the past many designers have used empirical *exposure factors* which were arbitrarily chosen to increase the calculated heat loss on the side or sides of the building exposed to the prevailing winds. It is also possible to differentiate among the various exposures more accurately by calculating the infiltration and transmission losses separately for the different sides of the building, using different assumed wind velocities. Recent investigations indicate, however, that the wind direction indicated by Weather Bureau instruments does not always correspond with the direction of actual impact on the building walls, due to deflection by surrounding buildings.

The exposure factor, which is still in use by many engineers, is usually taken as 15 per cent, and is added to the calculated heat loss on the side or sides exposed to what is considered the prevailing winter wind. There is a need for actual test data on this point, and pending the time when it can be secured, the question must be left to the judgment of the designing engineer. It should be remembered that the values of U in the tables in Chapter 5 are based on a wind velocity of 15 mph and that the infiltration figures are supposed to be selected from the tables in Chapter 6 to correspond to the wind velocities given in Table 2 of the present chapter.

The *Heating, Piping and Air Conditioning Contractors National Association* has devised a method[8] for calculating the square feet of equivalent direct radiation required in a building. This method makes use of exposure factors which vary according to the geographical location and the angular situation of the construction in question in reference to prevailing winds and the velocity of them.

AUXILIARY HEAT SOURCES

The heat supplied by persons, lights, motors and machinery should always be ascertained in the case of theaters, assembly halls, and industrial plants, but allowances for such heat sources must be made only after careful consideration of all local conditions. In many cases, these heat sources should not be allowed to affect the size of the installation at all, although they may have a marked effect on the operation and control of the system. In general, it is safe to say that where audiences are involved, the heating installation must have sufficient capacity to bring the building up to the stipulated inside temperature before the audience arrives. In industrial plants, quite a different condition exists, and heat sources, if they are always available during the period of human occupancy, may be substituted for a portion of the heating installation. In no case should the actual heating installation (exclusive of heat sources) be reduced below that required to maintain at least 40 F in the building.

Electric Motors and Machinery

Motors and the machinery which they drive, if both are located in the room, convert all of the electrical energy supplied into heat, which is

[8]See Standards of *Heating, Piping and Air Conditioning Contractors National Association.*

retained in the room if the product being manufactured is not removed until its temperature is the same as the room temperature.

If power is transmitted to the machinery from the outside, then only the heat equivalent of the brake horsepower supplied is used. In the first case the Btu supplied per hour $= \frac{\text{Motor horsepower}}{\text{Efficiency of motor}} \times 2{,}546$, and in the second case Btu per hour = bhp $\times$ 2,546, in which 2,546 is the Btu equivalent of 1 hp-hour. In high-powered mills this is the chief source of heating and it is frequently sufficient to overheat the building even in zero weather, thus requiring cooling by ventilation the year round.

The heat (in Btu per hour) from electric lamps is obtained by multiplying the watts per lamp by the number of lamps and by 3.415. One cubic foot of producer gas gives off about 150 Btu per hour; one cubic foot of illuminating gas gives off about 535 Btu per hour; and one cubic foot of natural gas gives off about 1000 Btu per hour. A Welsbach burner averages 3 cu ft of gas per hour and a fish-tail burner, 5 cu ft per hour. For information concerning the heat supplied by persons, see Chapter 3.

In intermittently heated buildings, besides the capacity necessary to care for the normal heat loss which may be calculated according to customary rules, additional capacity should be provided to supply the heat necessary to warm up the cold material of the interior walls, floors, and furnishings. Tests have shown that when a cold building has had its temperature raised to about 60 F from an initial condition of about 0 F, the heat absorbed from the air by the material in the structure may vary from 50 per cent to 150 per cent of the normal heat loss of the building. It is therefore necessary, in order to heat up a cold building within a reasonable length of time, to provide such additional capacity. If the interior material is cold when people enter a building, the radiation of heat from the occupants to the cold material will be greater than is normal and discomfort will result. (See Chapter 3.)

WALL SURFACE CONDENSATION[4]

Condensation on the interior surfaces of buildings is often a serious problem. Water dripping from a ceiling may cause irreparable damage to manufactured articles and machinery. It often results in short-circuiting of electric power and lighting systems, necessitating shut-downs and incurring costly repairs. It also causes rotting of wood roof structures, corrosion of metal roofs, and spalling and disintegration of gypsum and other types of roof decks not properly protected.

Condensation is caused by the contact of the warm humid air in a building with surfaces below the dew-point temperature, and can be remedied in two ways, (1) by increasing the temperature of such surfaces above the dew-point temperature, or (2) by lowering the humidity.

Dehumidification, of course, is not advisable where a high relative humidity is necessary for manufacturing processes. Hence, the only alternative is to increase the surface temperature by decreasing the inside

[4]Preventing Condensation on Interior Building Surfaces, by Paul D Close (A S H.V E Transactions, Vol. 36, 1930).

surface resistance. This can be accomplished by increasing the velocity of air passing over the surface, or by increasing the over-all resistance of the wall or roof by installing a sufficient thickness of insulation.

The latter method is generally used, and the thickness of insulation is determined by ascertaining the amount of resistance to be added to increase the temperature of the interior surface above the dew-point temperature for the maximum conditions involved. This in turn is based on the fundamental principle that the drop in temperature is proportional to the resistance. See Question 12 at the end of this chapter.

HEAT LOSS COMPUTATION EXAMPLE

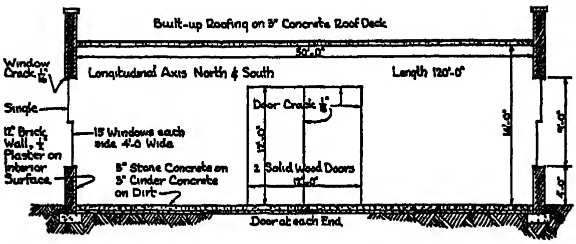


FIG. 1. ELEVATION OF FACTORY BUILDING

1. LOCATION...Philadelphia, Pa.

2. LOWEST OUTSIDE TEMPERATURE. (Table 2).. − 6 F

3. BASE TEMPERATURE: *In this example* a design temperature 10 F above lowest on record instead of 15 F is used. Hence the base temperature =

$$(-6 + 10) = +4 \text{ F.}$$

4. DIRECTION OF PREVAILING WIND (during Dec., Jan., Feb.)........................Northwest

5. BREATHING-LINE TEMPERATURE (5 ft from floor)......................................60 F

6. INSIDE AIR TEMPERATURE AT ROOF:

   The air temperature just below *roof* is higher than at the breathing line. Height of roof is 16 ft, or it is 16 − 5 = 11 ft above breathing line. Allowing 2 per cent per foot above 5 ft, or 2 × 11 = 22 per cent, makes the temperature of the air under the roof = 1.22 × 60 = 73.2 F.

7. INSIDE TEMPERATURE AT WALLS:

   The air temperature at the mean height of the *walls* is greater than at the breathing line. The mean height of the walls is 8 ft and allowing 2 per cent per foot above 5 ft, the average mean temperature of the walls is 1.06 × 60 = 63.6 F. By similar assumptions and calculations, the mean temperature of the *glass* will be found to be 64.2 F and that of the *doors* 61.2 F.

8. AVERAGE WIND VELOCITY (Table 2)..11.0 mph

9. OVER-ALL DIMENSIONS (See Fig. 1)...120 x 50 x 16 ft

10. CONSTRUCTION:

    Walls—12-in. brick, with ½-in. plaster applied directly to inside surface.

    Roof—3-in. stone concrete and built-up roofing.

Floor—5-in. stone concrete on 3-in. cinder concrete on dirt.

Doors—One 12 ft x 12 ft wood door (2 in. thick) at each end.

Windows—Fifteen, 9 ft x 4 ft single glass double-hung windows on each side.

11. Transmission coefficients:

Walls—(Table 3, Chapter 5, Wall 2B)........ $U = 0.34$
Roof—(Table 11, Chapter 5, Roofs 2A and 3A)........ $U = 0.77$
Floor—(Table 10, Chapter 5, Floors 5A and 6A)........ $U = 0.63$
Doors—(Table 13B, Chapter 5)........ $U = 0.46$
Windows—(Table 13A, Chapter 5)........ $U = 1.13$

12. Infiltration Coefficients:

Windows—Average windows, non-weatherstripped, $\frac{1}{16}$-in crack and $\frac{3}{64}$-in. clearance. The leakage per foot of crack for an 11-mile wind velocity is 25.0 cfh. (Determined by interpolation of Table 2, Chapter 6) The heat equivalent per hour per degree per foot of crack is taken from Chapter 6.

$$25.0 \times 0.018 = 0.45 \text{ Btu per deg Fahr per foot of crack.}$$

Doors—Assume infiltration loss through door crack twice that of windows or $2 \times 0.45 = 0.90$ Btu per deg Fahr per foot of crack.

Walls—As shown by Table 1, Chapter 6, a plastered wall allows so little infiltration that in this problem it may be neglected.

13. Calculations: See calculation sheet, Table 3.

Table 3. Calculation Sheet Showing Method of Estimating Heat Losses of Building Shown in Fig. 1

Part of Building	Width in Feet	Height in Feet	Net Surface Area or Crack Length	Coefficient	Temp. Diff.	Total Btu
North Wall						
Brick, ½-in plaster	50	16	656	0 34	59 6	13,293
Doors (2-in. wood)	12	12	144	0 46	57 2	3,789
⅛ in. Crack	1 pair doors		60	0 90	57 2	1,544[a]
West Wall:						
Brick, ½-in plaster	120	16	1380	0.34	59.6	27,964
Glass (Single)	15 x 4	9	540	1.13	60.2	36,734
⅛ in. Crack	Double Hung Windows (15)		450	0 45	60 2	6,095[a]
South Wall	Same as North Wall					18,626
East Wall	Same as West Wall					70,793
Roof, 3-in. concrete and slag-surfaced built-up roofing	50	120	6000	0.77	69.2	319,704
Floor, 5-in stone concrete on 3-in cinder concrete	50	120	6000	0 63	5[b]	18,900
Grand Total of heat required for building in Btu per hour						517,442

[a]This building has no partitions and whatever air enters through the cracks on the windward side must leave through the cracks on the leeward side. Therefore, only one-half of the total crack will be used in computing infiltration for each side and each end of building.

[b]A 5 F temperature differential is commonly assumed to exist between the air on one side of a large floor laid on the ground and the ground.

PROBLEMS IN PRACTICE

1 ● What is the relation between the sensible heat loss from a building and the heat required for humidification?

A house with a volume of 14,000 cu ft has a heat loss 120 Mbh for standard uninsulated frame construction and a 70 F temperature difference. Assuming a leakage rate of 1½ air changes per hour it would require about 10 Mbh to maintain a relative humidity of 45 per cent when the outside air is 0 F and 50 per cent relative humidity. By using an insulation such as rock wool, the sensible heat loss of this house may be reduced to approximately 77 Mbh The insulation does not affect the humidification load, which now assumes greater importance.

2 ● What inside dry-bulb temperatures are usually assumed for· (a) homes, (b) schools, (c) public buildings?

Referring to Table 1:

a. 70 to 72 F.

b. Temperature varies from 55 to 75 F, depending on the room. Classrooms, for instance, are usually specified as 70 to 72 F.

c. 68 to 72 F

3 ● How is the outside temperature selected for use in computing heat losses?

The outside temperature used in computing heat losses is generally taken from 10 to 15 F higher than the lowest recorded temperature as reported by the Weather Bureau during the preceding 10 years for the locality in which the heating system is to be installed In some cases where the lowest recorded temperature is extremely unusual, the design temperature is taken even higher than 15 F above the lowest recorded temperature

4 ● What are the effects of wind movement on the heating load?

a. Wind movement increases the heat transmission of walls, glass, and roof, it affects poor walls to a much greater extent than good walls.

b Wind movement materially increases the infiltration (inleakage) of cold air through the cracks around doors and windows, and even through the building materials themselves if such materials are at all porous.

5 ● Calculate the heat given off by eighteen 200-watt lamps.

200 × 18 × 3.415 = 12,294 Btu per hour.

6 ● A two-story, six-room, frame house, 28-ft by 30-ft foundation, has the following proportions:

Area of outside walls, 1992 sq ft.
Area of glass, 333 sq ft.
Area of outside doors, 54 sq ft.
Cracks around windows, 440 ft.
Cracks around doors, 54 ft.
Area of second floor ceiling, 783 sq ft.
Volume, first and second floors, 13,010 cu ft.
Ceilings, 9 ft high.

The minimum temperature for the heating season is −34 F, and the required inside temperature at the 30-in. level is 70 F. The average number of degree days for a heating season is 7851, and the average wind velocity is 10 mph, northwest.

The walls are constructed of 2-in. by 4-in. studs with wood sheathing, building paper, and wood siding on the outside, and wood lath and plaster on the inside. Windows are single glass, double-hung, wood, without weatherstrips. The second floor ceiling is metal lath and plaster, without an attic floor. The roof is of wood shingles on wood strips with rafters exposed. The area of the roof is 20 per cent greater than the area of the ceiling. Select values for the following: (a) U for walls; (b) U for glass; (c) U for second floor ceiling; (d) U for roof;

(e) U for ceiling and roof combined; (f) air leakage, cubic feet per hour per foot of window crack; (g) air leakage, cubic feet per hour per foot of door crack.

a. 0.25 (Table 5, Chapter 5).
b. 1.13 (Table 13, Chapter 5).
c. 0.69 (Table 8, Chapter 5)
d 0.46 (Table 12, Chapter 5).
e. 0.23 (Equation 6, Chapter 5).
f. 21.4 (Table 2, Chapter 6).
g. 42 8, which is double the window leakage.

7 ● Using the data of Question 6, calculate the maximum Btu loss per hour for the various constructions, and show the percentage of the total heat which is lost through each construction described.

Assume 2 per cent rise in temperature for each foot in height. The average temperature will be 72 8 F for walls, doors, and windows, and 79.1 F for the second floor ceiling.

a. Outside walls	46,200 Btu loss	37.2 per cent of total
b. Glass	34,950 Btu loss	28.1 per cent of total
c. Doors	5,670 Btu loss	4.6 per cent of total
d. Second floor ceiling	17,840 Btu loss	14.3 per cent of total
e. Air leakage, windows	15,750 Btu loss	12.7 per cent of total
f. Air leakage, doors	3,865 Btu loss	3.1 per cent of total
Total	124,275 Btu loss	100.0 per cent of total

8 ● For the house in Question 6, place 1-in. insulation in the outside walls and second floor ceiling; k for insulation = 0.34. Use weatherstrip on doors and windows, and double glass on the windows; C = 0.55. Calculate or select the following values: (a) U for walls; (b) U for glass; (c) U for second floor ceiling; (d) U for combination of ceiling and roof; (e) air leakage, cubic feet per hour per foot of door crack; (f) air leakage, cubic feet per hour per foot of window crack.

a. 0.144
b. 0.55
c. 0.23
d. 0.13
e. 15.5
f. 31.0

9 ● Calculate the maximum Btu loss per hour and show the percentage loss by each channel for the house as insulated in Question 8.

a. Outside walls	26,650 Btu loss	36.2 per cent of total
b. Glass	17,000 Btu loss	23 1 per cent of total
c. Doors	5,670 Btu loss	7.7 per cent of total
d. Ceiling	10,070 Btu loss	13.7 per cent of total
e. Air leakage, windows	11,400 Btu loss	15.5 per cent of total
f. Air leakage, doors	2,795 Btu loss	3.8 per cent of total
Total	73,585 Btu loss	100.0 per cent of total

10 ● From the results of Questions 7 and 9, calculate the Btu saved and the percentage saved by each change in construction.

	UNINSULATED	INSULATED	BTU SAVED	PER CENT SAVED
a. Outside walls	46,200	26,650	19,550	42.3
b. Glass	34,950	17,000	17,950	51.4
c. Doors	5,670	5,670	0	0
d. Ceiling	17,840	10,070	7,770	43.5
e. Air leakage, windows	15,750	11,400	4,350	27.6
f. Air leakage, doors	3,865	2,795	1,070	27.7

11 ● From the results of Questions 7 and 9, calculate the heat loads per heating season in Btu and note the savings by better construction.

The 7851 degree days for the heating season multiplied by 24 hours, times the Btu loss per hour for 1 F drop in temperature gives the Btu load per heating season.

Saving = 250,800,000 − 148,000,000 = 102,800,000 Btu

12 ● The dry-bulb temperature and the relative humidity at the ceiling of a mixing room in a bakery are 80 F and 60 per cent, respectively. The roof is a 4-in. concrete deck covered with built-up roofing. If the lowest outside temperature to be expected is —10 F, what thickness of rigid fiber insulation will be required to prevent condensation?

From Table 11, Chapter 5, U for the uninsulated roof = 0.72. From Table 2, Chapter 5, k for rigid fiber insulation = 0.33. From the psychrometric chart the dew point of air at 80 F and 60 per cent relative humidity is 65 F The ceiling temperature, therefore, must not drop below 65 F if condensation is to be prevented.

When equilibrium is established, the amount of heat flowing through any component part of a construction is the same for each square foot of area.

Therefore,

$$U\,[80 - (-10)\,] = 1.65\,(80 - 65)$$

where

U is the transmittance of the insulated roof.

Solving the equation, $U = 0.275$.

The resistance of the insulated roof $= \dfrac{1}{0.275} = 3.64$.

The resistance of the uninsulated roof $= \dfrac{1}{0.72} = 1.39$

The resistance of the insulation $= 3.64 - 1.39 = 2.25$

Resistance per inch of insulation $= \dfrac{1}{0.33} = 3.0$.

Since a resistance of 2.25 is required, and 1 in. of insulation has a resistance of 3, one inch will be sufficient to prevent condensation.

The same result might have been obtained by selecting an insulated 4-in. concrete slab having a U of less than 0.275 from Table 11, Chapter 5. This 4-in. concrete slab with 1-in. rigid insulation has a U of 0.23 which is safe.

Chapter 8

COOLING LOAD

Conditions of Comfort, Cooling Load, Sensible Heat Conducted, Design Outside Temperatures, Solar Radiation Through Walls and Roofs, Time Lag, Solar Radiation Transmitted Through Glass, Outside Air Heat and Moisture Leakage, Heat and Moisture Sources

THE method of calculating the cooling load is similar to that used in calculating the heating load. The direction of the flow of heat is reversed, however, and in most cases additional factors must be considered, such as solar radiation and the heat from occupants, lights, motors, and other sources. The character of the load depends on the type of building to be cooled as, for example, in auditoriums and other places of assemblage where the maximum load usually is that due to the heat and moisture given off by the occupants, or in office buildings and residences where solar radiation and the transmission and infiltration of heat through the building shell are most important.

While cooling is generally identified with the summer season, it is often necessary to cool in winter as well as in summer. In a crowded place of assemblage the heat given off by the occupants, together with that given off by the lighting and power equipment, may be more than the normal heat loss through the structure even in winter under cold climatic conditions.

Much of the basic information for the design of comfort conditioning installations has resulted from research conducted at the A.S.H.V.E. Research Laboratory and at institutions with which cooperative research investigations have been carried on. These data include the effective temperature index, and heat and moisture loss data given in Chapter 3.

CONDITIONS OF COMFORT

The conditions to be maintained in an enclosure are variable and depend on many factors, especially the season of the year and (during the summer) the outside dry-bulb temperature and the duration of the period of occupancy. Information concerning the proper effective temperatures to be maintained for various seasons is given in Chapter 3, where are also tabulated the most desirable indoor air conditions to be maintained in summer for exposures less than three hours. (See Table 2, Chapter 3.)

In installations for restaurants and theaters the requirements are different from those in offices, since there must be a considerable volume of air circulated in order to provide ventilation and cooling

TABLE 1. DESIGN DRY- AND WET-BULB TEMPERATURES, WIND VELOCITIES, AND WIND DIRECTIONS FOR JUNE, JULY, AUGUST, AND SEPTEMBER

STATE	CITY	DESIGN DRY-BULB	DESIGN WET-BULB	SUMMER WIND VELOCITY MPH	PREVAILING SUMMER WIND DIRECTION
Ala.	Birmingham	93	77	5 2	S
	Mobile	94	78	8 6	SW
Ariz.	Phoenix	110	77	6.0	W
Ark.	Little Rock	95	77	7.0	NE
Calif.	Los Angeles	88	70	6 0	SW
	San Francisco	85	68	11.0	SW
Colo	Denver	90	64	6 8	S
Conn.	New Haven	88	74	7.3	S
D. C.	Washington	95	78	6.2	S
Fla.	Jacksonville	94	78	8.7	SW
	Tampa	94	79	7.0	E
Ga.	Atlanta	91	75	7.3	NW
	Savannah	95	79	7.8	SW
Idaho	Boise	95	65	5.8	NW
Ill.	Chicago	95	75	10.2	NE
	Peoria	91	75	8.2	S
Ind	Indianapolis	90	73	9.0	SW
Iowa	Des Moines	92	74	6.6	SW
Ky.	Louisville	94	75	8.0	SW
La.	New Orleans	94	79	7.0	SW
Maine	Portland	85	71	7.3	S
Md.	Baltimore	93	76	6.9	SW
Mass.	Boston	88	73	9.2	SW
Mich.	Detroit	93	73	10.3	SW
Minn.	Minneapolis	93	72	8 4	SE
Miss.	Vicksburg	95	78	6.2	SW
Mo.	Kansas City	92	75	9 5	S
	St. Louis	95	78	9.4	SW
Mont.	Helena	87	63	7.3	SW
Nebr.	Lincoln	93	74	9.3	S
Nev	Reno	93	64	7.4	W
N. J.	Trenton	95	76	10.0	SW
N. Y.	Albany	90	74	7 1	S
	Buffalo	83	72	12.2	SW
	New York	95	75	12.9	SW
N. M.	Santa Fe	87	63	6.5	SE
N. C.	Asheville	87	72	5.6	SE
	Wilmington	93	79	7.8	SW
N Dak.	Bismarck	88	69	8.8	NW
Ohio	Cleveland	95	75	9.9	S
	Cincinnati	95	78	6.6	SW
Okla	Oklahoma City	96	76	10.1	S
Oreg.	Portland	83	65	6.6	NW
Pa.	Philadelphia	95	78	9.7	SW
	Pittsburgh	91	73	9.0	NW
R. I.	Providence	85	73	10.0	NW
S. C.	Charleston	94	80	9.9	SW
	Greenville	93	76	6.8	NE
Tenn.	Chattanooga	94	76	6.5	SW
	Memphis	93	77	7.5	SW
Texas	Dallas	99	76	9.4	S
	Galveston	93	79	9 7	S
	San Antonio	100	78	7.4	SE
	Houston	93	79	7.7	S
	El Paso	98	69	6.9	E

TABLE 1. DESIGN DRY- AND WET-BULB TEMPERATURES, WIND VELOCITIES, AND WIND DIRECTIONS FOR JUNE, JULY, AUGUST, AND SEPTEMBER (Continued)

STATE	CITY	DESIGN DRY-BULB	DESIGN WET-BULB	SUMMER WIND VELOCITY MPH	PREVAILING SUMMER WIND DIRECTION
Utah	Salt Lake City	95	67	8.2	SE
Vt.	Burlington	85	71	8.9	S
Va.	Norfolk	91	76	10.9	S
	Richmond	95	78	6.2	SW
Wash.	Seattle	83	61	7.9	S
	Spokane	89	63	6.5	SW
W. Va.	Parkersburg	90	74	5.3	SE
Wis.	Madison	89	73	8.1	SW
	Milwaukee	93	74	10.4	S
Wyo.	Cheyenne	85	62	9.2	S

COOLING LOAD

The cooling load may be divided into the following parts:

1. Transmission of heat through walls, roof, and glass with allowances for sun-exposed surfaces and heat capacity
2. Transmission of solar radiation through glass and absorption by interior furnishings.
3. Heat and moisture from infiltration and from outside air introduced.
4. Heat and moisture from occupants and heat from lights, machinery and other sources.

Sensible Heat Conducted

The transmission load *for surfaces not exposed to the sun* is calculated in a manner similar to that described in Chapter 7, by means of the following formula:

$$H_t = AU(t_o - t) \tag{1}$$

where

H_t = heat transmitted through the material of the wall, glass, roof, or floor, Btu per hour.

A = net inside area of wall, glass, roof, or floor, square feet.

t = inside temperature, degrees Fahrenheit.

t_o = outside temperature, degrees Fahrenheit.

U = coefficient of transmission of wall, floor, roof or glass, Btu per hour per square foot per degree Fahrenheit difference in temperature (Tables 3 to 13, Chapter 5.)

Design Outside Temperatures

Summer dry-bulb and wet-bulb temperatures for various cities are given in Table 1. It will be noted that the temperatures are not the maximums but the design temperatures which should be used in air conditioning calculations. The maximum outside wet-bulb temperatures as given in Weather Bureau reports usually occur only from 1 per cent to 4 per cent of the time, and they are therefore of such short duration that it is not practical to design a cooling system covering this range. The temperatures shown in Table 1 are based on a study of hourly tempera-

tures in New York City from which factors were derived and applied to the average maximum dry- and wet-bulb temperatures for other cities. This study covered a twenty-year record of Weather Bureau temperatures. The design temperatures given are not exceeded more than 5 to 8 per cent of the time during a cooling season of 1200 hours in June, July, August, and September for an average year.

Solar Radiation Through Walls and Roofs

Fig. 1[1] shows the total amount of solar energy in Btu per square foot per hour received during the day by a surface normal to the rays of the sun, by a horizontal surface, and by east, west, and south walls. The curves are drawn from A.S.H.V.E. Laboratory data obtained by pyrheliometer, are based on sun time, and are for a perfectly clear day on August 1 at a north latitude of 40 deg. Data from these curves may be used with little error for most United States latitudes and for all of the hotter months of the year.

The absorption of solar radiation by a surface depends upon the character of the surface and the angle of the surface with respect to the direction of the radiation. The heat absorption by a black oilcloth surface perpendicular to the sun's rays was found to be as high as 273 Btu per square foot per hour, based on tests conducted by the A.S.H.V.E. Research Laboratory in Pittsburgh[2]. Lamp black, red brick dust, and aluminum bronze painted surfaces perpendicular to the sun's rays showed, respectively, 94.0, 63.4, and 28.2 per cent as high a rate of absorption as the black oilcloth.

TABLE 2. ALLOWANCE FOR SOLAR RADIATION ON ROOFS AND WALLS

APPROXIMATE NUMBER OF DEGREES TO ADD TO DRY-BULB TEMPERATURE FOR DIFFERENT TYPES OF SURFACES

TYPE OF SURFACE	BLACK	RED BRICK OR TILE	ALUMINUM PAINT
Roof, horizontal	45	30	15
East or west wall	30	20	10
South wall	15	10	5

Solar radiation is an important factor in the mechanism of heat flow into buildings. Research conducted at the A.S.H.V.E. Research Laboratory[3] has shown that a large error may be introduced into the calculations by failure to consider the periodical character of heat flow resulting from the diurnal movement of the sun and the heat capacity of the structure, which determine the timing and magnitude of the heat wave flowing through the wall into a building on a hot, sunny day.

Unfortunately, the calculations for the transmission of heat from solar

[1]Heat Transmission as Influenced by Heat Capacity and Solar Radiation, by F. C Houghten, J L. Blackshaw, E M. Pugh and Paul McDermott (A.S H V.E TRANSACTIONS, Vol 38, 1932).

[2]Absorption of Solar Radiation in Relation to the Temperature, Color, Angle, and Other Characteristics of the Absorbing Surface, by F C Houghten and Carl Gutberlet (A S H V E TRANSACTIONS, Vol. 36, 1930).

[3]Coefficients of Heat Transfer as Measured under Natural Weather Conditions, by F. C Houghten and C. G F Zobel (A S H V.E TRANSACTIONS, Vol 34, 1928); Absorption of Solar Radiation in Its Relation to the Temperature, Color, Angle and Other Characteristics of the Absorbing Surface, by F. C Houghten and Carl Gutberlet (A.S.H.V E. TRANSACTIONS, Vol 36, 1930)

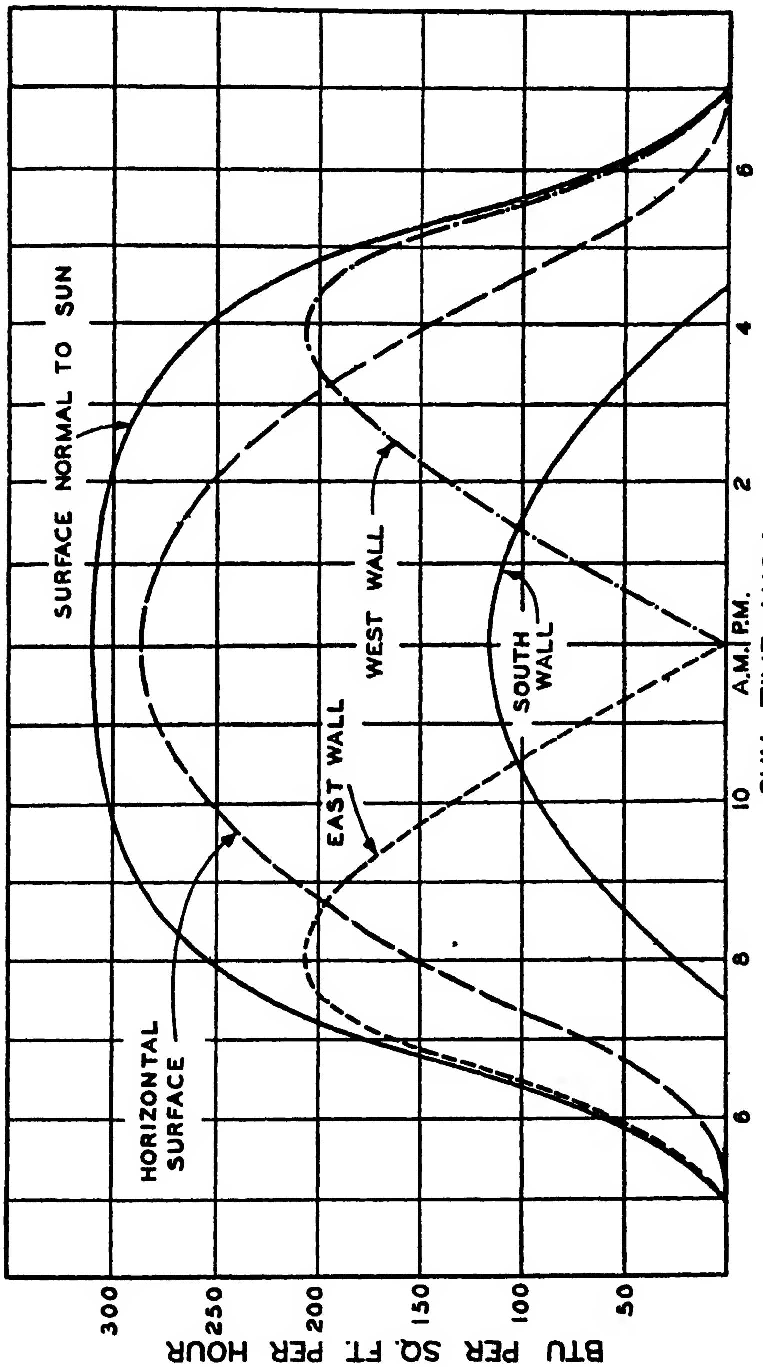


FIG. 1. CURVES GIVING SOLAR INTENSITY NORMAL TO SUN, ON HORIZONTAL SURFACE AND ON WALLS FOR AUGUST 1

radiation through building walls are too complicated to be of much practical value to the heating and ventilating engineer. Approximate results may be obtained by adding the number of degrees given in Table 2 to the outside design dry-bulb temperature in calculating the heat transmission through a wall or roof which may be exposed to the sun for an appreciable length of time. Table 2 was obtained from a study of the data in A.S.H.V.E. research papers on solar radiation[4]. Black and aluminum painted surfaces represent the extremes which are likely to occur. For other types of surfaces, values intermediate between those given in the table can be used.

Time Lag

The calculation of heat transmitted through walls and roofs does not take into consideration the heat capacity of the structure and the consequent time lag in the transmission of heat. In the thick walls used in modern office buildings the time lag may amount to 10 hours or more[5]. Thus in many cases the wall transmission cannot be added directly to the cooling load from other sources because the peak of the wall transmission load may not coincide with the peak of the total cooling load and may even occur after the cooling system has been shut down for the day. The data in Table 3 were taken from A.S.H.V.E. research papers[6] and while they result principally from a study of experimental slabs, they give an idea of the time lag to be expected in various structures.

Table 3. Time Lag in Transmission of Solar Radiation through Walls and Roofs

Type and Thickness of Wall or Roof	Time Lag, Hours
2-in pine	1½
6-in. concrete	3
4-in. gypsum	2½
3-in concrete and 1-in. cork	2
2-in. iron and cork (equivalent to ¾-in. concrete and 2.15-in cork)	2½
4-in. iron and cork (equivalent to 5½-in. concrete and 1.94-in. cork)	7¼
8-in. iron and cork (equivalent to 16-in. concrete and 1.53-in cork)	19
22-in. brick and tile wall	10

In intermittently cooled buildings the cooling capacity must be sufficient to care for the load imposed by the necessity to cool down the furnishings and the material of the interior construction to the point of maintained temperatures.

Solar Radiation Transmitted Through Glass

In considering the transmission through glass several factors must be considered. As the sun's rays impinge against a pane of glass, most of the radiation passes through to the other side, a small amount is reflected, and the balance is absorbed by the glass. The amount absorbed depends upon the character and thickness of the glass and the angle between the sun's rays and the glass. The temperature of the glass is raised by the absorbed

[4]Loc Cit Notes 1 and 2.

[5]Field Studies of Office Building Cooling, by J H Walker, S S Sanford and E P Wells (A S H.V.E. Transactions, Vol 38, 1932)

[6]Loc Cit Notes 1, 2 and 5

heat and this heat is then delivered to the air on the two sides of the glass in proportion to the difference between glass and air temperatures[7].

The A.S.H.V.E. tests[8] indicated that a single pane of double strength glass 0.127 in. thick absorbs approximately 11 per cent of the solar radiation passing through it when the impingement is normal. For smaller angles of impingement, the glass retards percentages of the total radiant energy approximately in proportion to the sine of the angle. Other experiments[9] indicate a glass absorption of 16.7 per cent for one pane of glass and 37.5 per cent for two ¼-in. panes separated by a 1¾-in. air space.

The amount of solar radiation delivered to an unshaded glass surface may be obtained from the curves in Fig. 1. For surfaces other than those given, the solar radiation normal to the glass must be calculated. Hendrickson and Walker[10] have shown how this may be done if the wall faces some direction other than east, west, or south. They have also shown how to calculate the net glass area on which the solar radiation impinges when the glass is partly shaded by the frame or wall. The values from Fig. 1 must be used only for the net glass area on which the sun shines. Tests at the A S.H.V.E. Research Laboratory[11] have determined the percentages of heat from solar radiation actually delivered to a room with bare windows and with various types of outdoor and indoor shading. The data in Table 4 are taken from these tests.

TABLE 4. SOLAR RADIATION TRANSMITTED THROUGH BARE AND SHADED WINDOWS

TYPE OF APPURTENANCE	FINISH FACING SUN	PER CENT DELIVERED TO ROOM
Bare window glass		97
Canvas awning	Plain	28
Canvas awning	Aluminum	22
Inside shade, fully drawn	Aluminum	45
Inside shade, one-half drawn	Buff	68
Inside Venetian blind, fully covering window, slats at 45 deg	Aluminum	58
Outside Venetian blind, fully covering window, slats at 45 deg	Aluminum	22

The percentage figures in this table were obtained by dividing the total amount of heat actually entering through the shaded window by the total amount of heat calculated to enter through a bare window (solar radiation plus glass transmission based on observed outside glass temperature). For bare windows on which the sun shines, the transmission of heat from outside air to glass is small or negative as the glass temperature is raised by the solar radiation absorbed. Therefore, in calculating the total heat gain through windows on the sunny sides of buildings, it is sufficiently accurate to figure the total cooling load due to the window, as the solar radiation times the proper factor from Table 4, and

[7]Heat Absorbing Glass Windows, by W W Shaver (A S H V.E Journal Section, *Heating, Piping and Air Conditioning*, September, 1935)

[8]Radiation of Energy Through Glass, by J L Blackshaw and F C Houghten (A S H.V E TRANSACTIONS, Vol. 40, 1934). Studies of Solar Radiation Through Bare and Shaded Windows, by F. C. Houghten, Carl Gutberlet, and J L Blackshaw (A S H.V E TRANSACTIONS, Vol 40, 1934).

[9]Loc Cit Note 5

[10]Summer Cooling for Comfort as Affected by Solar Radiation, by G. A Hendrickson and J H Walker, *Heating and Ventilating*, November, 1932, and The Determination of Sun Effect on Summer Cooling Loads, by G A Hendrickson and J H Walker, *Heating and Ventilating*, June, 1933

[11]Loc Cit Note 8

to neglect the heat transmission through the glass caused by the difference between the temperatures of the inside and outside air. Another reason for neglecting this glass transmission load is that the curves in Fig. 1 were based on the maximum intensity of solar radiation observed at the A.S.H.V.E. Laboratory during a three-year study, so results based on these curves will be amply high. It will be noted that Table 4 gives the amount of heat delivered through the window as 97 per cent of the solar radiation, which is greater than is indicated by the figures for absorption in the preceding paragraph. The explanation is that much of the radiation absorbed by the glass is delivered to the room.

Although 97 per cent of the heat from solar radiation is delivered to a room through bare window glass, more recent tests[12] have indicated that in the case of buildings having floors of high heat capacity such as concrete floors on which the solar radiation falls, approximately one-half of the heat entering a bare window is absorbed by the floor and does not immediately become a part of the cooling load but is delivered back to the air in the building at a slow rate over a period of 24 hours or longer.

Fig. 1 shows that the maximum solar intensity on any surface is of limited duration. In the case of windows the total energy impinging on the glass before and after the time of maximum intensity is further reduced by increased shading of the glass from the frame, or wall. The cooling load due to solar radiation therefore does not have to be figured as a steady load. Another point which should be noted is that the maximum solar radiation load on an east wall occurs early in the morning when the outside temperature is low.

In a paper by the A.S.H.V.E. Research Laboratory[13] it was shown that ordinary double strength window glass transmits no measurable amount of energy radiated from a source at 500 F or lower; that it transmits only 6.0 and 12.3 per cent of the total radiation from surfaces at 700 F and 1000 F, respectively; and that it transmits 65.7 per cent of the radiation from an arc lamp, 76.3 per cent of the radiation from an incandescent tungsten lamp, and 89.9 per cent of the radiation from the sun. Thus, glass windows in a room constitute heat traps, which allow rather free transmission of radiant energy into the room from the sun to warm objects in it, but do not allow the transmission of re-radiated heat from these same objects.

Tests[14] have been made which indicated that sunshine through window glass is the most important factor to contend with in the cooling of an office building. At times it was shown to account for as much as 75 per cent of the total cooling necessary. Because of the importance of the sunshine load, cooling systems should be zoned so that the side of the building on which the sun is shining can be controlled separately from the other sides of the building. If buildings are provided with awnings so that the window glass is shielded from sunshine, the amount of cooling required will be reduced and there will also be less difference in the cooling requirements of different sides of the building. The total cooling load for a building exposed to the sun on more than one side is of course less than

[12]Cooling Requirements of Single Rooms in a Modern Office Building, by F. C Houghten, Carl Gutberlet, and Albert J. Wahl (A S H.V.E Journal Section, *Heating, Piping and Air Conditioning*, April, 1935)

[13]Loc Cit. Note 8.

[14]Loc Cit Note 5

the sum of the maximum cooling loads in the individual rooms since the maximum solar radiation load on the different sides occurs at different times In determining the total cooling load for a building if the time when the maximum load occurs is not obvious, the load should be calculated for various times of day to determine the time at which the sum of the loads on the different sides of the building is a maximum.

Outside Air Heat and Moisture Leakage

An allowance must be made for the heat and moisture in the outside air introduced for ventilating purposes or entering the building through cracks, crevices, doors, and other places where infiltration might occur.

The volume of air entering due to infiltration may be estimated from data given in Chapter 6 using wind velocities from Table 1, Chapter 8. Information on the amount of outside air required for ventilation will be found in Chapter 3.

The heat gain resulting from the outside air introduced may be estimated from the following formula:

$$H_i = 60\, Q d_o\, (\Theta_o - \Theta) \tag{2}$$

where

H_i = heat to be removed from outside air entering the building, Btu per hour.
Q = volume of outside air entering the building, cubic feet per minute.
d_o = density of outside air, pounds of dry air per cubic foot of outside air, at the temperature t_o.
Θ_o = heat content of mixture of outside dry air (at temperature t_o) and water vapor, Btu per pound of dry air.
Θ = heat content of mixture of inside dry air (at temperature t) and water vapor, Btu per pound of dry air.

Heat and Moisture Sources

Figs. 8 to 11, Chapter 3, show the heat and moisture given off by human beings under various conditions of activity. For average conditions where

Table 5. Heat Gain Due to Various Devices, Btu per Hour

Lights and electrical appliances	3,413 per kw
Motors with connected load in same room[a]	
Nameplate rating, 1/8 to 1/2 hp	4,250 per hp
Nameplate rating, 1/2 to 3 hp	3,700 per hp
Nameplate rating, 3 to 20 hp	2,950 per hp
Restaurant coffee urns, 10-gal capacity	16,000
Dish warmers per 10 sq ft of shelf	6,000
Restaurant range—4 burners and oven	100,000
Residence gas range	
Giant burner	12,000
Medium burner	9,000
Oven	1,000 per cu ft of space
Pilot	250
Electric range	
Small burner, 1000 to 1350 watts	3,413 per kw
Large burner, 1700 to 2200 watts	3,413 per kw
Oven, 2000 to 3000 watts	3,413 per kw
Appliance connection, 660 watts	2,250
Warming compartment, 300 watts	1,025

[a]Deduct 2545 Btu per hp if connected load is outside of room.

a person is normally at rest, as in a theater, or doing very light work, as in a restaurant or residence, the total amount of heat given off will average about 400 Btu per hour. Part of this is latent heat due to the evaporation of 700 to 1200 grains of moisture per hour. Examples illustrating heat and moisture loss calculations for human beings are given in Chapter 10.

All sources of heat must of course be considered in designing the conditioning system. The heat gain due to various devices is given in Table 5.

Moisture evaporated by appliances must be included in the total latent heat load. In some cases only a small part of the heat from lights immediately affects the cooling load. Tests[15] show that with lights placed near the ceiling under some conditions the electricity used for illumination has little effect on the cooling load in an office cooled during the usual short period. The air heated by the lights stratifies closely to the ceiling and the temperature of the lower layers of air is raised only a small amount after a considerable lapse of time.

An example of cooling load calculation is given in Chapter 10. Another method of determining the heat gain of air conditioning loads, is given in a paper[16] presented before the Society which outlines solar effect and absorption coefficients to apply to walls facing several directions at various latitudes and at different times of the day.

---

[15]Loc. Cit Note 11

[16]A Rational Heat Gain Method for the Determination of Air Conditioning Cooling Loads, by F H Faust, L Levine and F O Urban (A S H V E. Journal Section, *Heating, Piping and Air Conditioning*, August, 1935).

PROBLEMS IN PRACTICE

1 ● In buildings such as an office building which is cooled intermittently, will the maximum cooling load occur coincidently with the maximum dry-bulb temperature?

Not necessarily Tests indicate that particularly for east and south exposures the maximum rate of cooling occurs shortly after the equipment is started in the morning. This extra heat is that which was absorbed by the building during the preceding day and also during the *off* period. The rapid lowering of the room temperature after the cooling equipment is started causes this stored up heat to flow quickly from wall and floor surfaces to the room air.

2 ● The outdoor and indoor temperatures are 90 F and 78 F, respectively. What is the amount of heat transmitted per hour through a 7 ft by 4 ft north window?

$H_t = 28 \times 1.13\ (90 - 78) = 380$ Btu per hour.

(Equation 1, Chapter 8 and Table 13, Chapter 5.)

3 ● What are the proper design temperatures for a Detroit store?

Outdoor dry-bulb, 93 F; wet-bulb, 73 F. (Table 1, Chapter 8.)

Indoor dry-bulb, 79.2 F; wet-bulb, 64 8 F. (Table 2, Chapter 3.)

4 ● a. What is the maximum heat transmission for a flat roof exposed to the sun with the outdoor and indoor temperature 95 F and 80 F, respectively? The roof is of uninsulated 6-in. concrete, with its underside exposed, and with a black upper surface.

b. If the temperatures specified were the maximum for the day and occurred at 12 o'clock, at what time would the maximum cooling load due to the roof exist?

a $H_t = 1 \times 0\ 64\ (95 + 45 - 80) = 38\ 4$ Btu per hour per square foot

(Equation 1 and Table 2, Chapter 8, and Table 11, Chapter 5)

b At 3 p.m. (Table 3, Chapter 8.)

5 ● For south windows equipped with plain canvas awnings, what is the maximum amount of heat delivered to a room when the outdoor temperature is 90 F and the indoor temperature is 78 F?

$115 \times 0\ 28 = 32\ 2$ Btu per square foot of glass (Fig. 1 and Table 4, Chapter 8, note that glass transmission can be neglected)

6 ● What is the heat gain per cubic foot of outside air introduced, under the following conditions if the barometric pressure is 29.50 in. Hg:

Outdoor temperatures, 90 F dry-bulb and 75 F wet-bulb.

Inside temperatures, 78 F dry-bulb and 65 F wet-bulb.

The relative humidity of the outdoor air is 50 per cent (Psychrometric Chart)

Pressure of saturated vapor at 90 F $= e_t = 1\ 4211$ in Hg. (Table 6, Chapter 1)

Pressure of vapor in the mixture $= 1\ 4211 \times 0\ 5 = 0\ 71055$ in. Hg.

Pressure of dry air in the mixture $= 29\ 50 - 0\ 71055 = 28\ 7895$ in Hg.

From Equation 4a, Chapter 1,

$d_a = \dfrac{29\ 50 - 0\ 7105}{0\ 753\ (90 + 460)} = \dfrac{28.79}{0\ 753 \times 550} = 0\ 06945$ = weight of dry air in 1 cu ft of the mixture.

$d_v = \dfrac{0\ 71055}{1.21\ (90 + 460)} = \dfrac{0\ 71055}{1.21 \times 550} = 0\ 001067$ = weight of vapor in 1 cu ft of the mixture at 50 per cent relative humidity.

Weight of 1 cu ft of the mixture $= 0\ 06945 + 0\ 001067 = 0\ 0705$ lb.

$H_l = 60\ Qd_o\ (\Theta_o - \Theta)$. (From Equation 2, Chapter 8).

$\Theta_o = 38\ 46$ and $\Theta = 29\ 96$ (Table 6, Chapter 1). The total heat of any air-vapor mixture may be obtained from the last column in Table 6, Chapter 1, by considering the temperatures to be wet-bulb readings, since the total heat of a mixture is constant for a given wet-bulb temperature.

$H_l = 0\ 0705\ (38.46 - 29\ 96) = 0\ 598$ Btu per cubic foot.

7 ● If there are twenty 200-watt lights in use in a room, what is the cooling load due to lights?

$200 \times 20 = 4000$ watts $= 4$ kw.

$3413 \times 4 = 13{,}652$ Btu per hour (Table 5, Chapter 8).

8 ● a. If a restaurant has two 10-gal coffee urns, what is the cooling load due to them?

b. What is the cooling load due to four 1350-watt burners on an electric range?

a. $16{,}000 \times 2 = 32{,}000$ Btu per hour (Table 5, Chapter 8)

b. $4 \times 1350 = 5400$ watts $= 5.4$ kw.

$5\ 4 \times 3413 = 18{,}430$ Btu per hour (Table 5, Chapter 8).

9 ● Why may the heat gain by transmission through glass on the sunny exposures be neglected?

Solar radiation on a glass surface during peak hours will cause the outer surface of the

glass to rise up to or slightly above the simultaneous outside air temperature, hence no heat can flow from outside air to the glass at this time

10 ● What should be the dry- and wet-bulb temperatures in a restaurant when the outdoor dry-bulb temperature is 95 F?

Dry-bulb, 80 F; wet-bulb, 65 F (Table 2, Chapter 3).

11 ● In an office building in which the occupants remain indoors most of the day, what is the most desirable indoor dry-bulb temperature and relative humidity in summer?

76.5 F and 50 per cent relative humidity (Fig. 6, Chapter 3).

12 ● An office with western exposure in a building having concrete floors is cooled only during office hours. If the windows are bare and unshaded, what is the maximum cooling load in the office due to solar radiation through the windows?

$215 \times 0.97 \times \frac{1}{2} = 104$ Btu per square foot of glass per hour (Fig 1 and Table 4, Chapter 8; note that only one-half of the heat need be considered because of absorption by the floor).

13 ● A 7 × 4 ft west window is equipped with an inside aluminum finished Venetian blind which is adjusted to fully cover the window when the sun shines. The net glass area if 75 per cent of the total area of the window. What is the cooling load due to the window at 10 a.m. and 4 p.m. assuming solar radiation as shown in Fig. 1 and dry-bulb temperatures as follows: 10 a.m., outside 85 F and inside 77 F; 4 p.m., outside 95 F and inside 80 F?

10 a.m.: $H_t = 28 \times 1.13\ (85 - 77) = 253$ Btu per hour (Equation 1, Chapter 8 and Table 13, Chapter 5).

4 p.m.: $28 \times 0.75 = 21$ sq ft net glass area.

$21 \times 215 \times 0.58 = 2619$ Btu per hour (Fig. 1 and Table 4, Chapter 8).

14 ● What is the cooling load of an ice cream cabinet in a restaurant due to a ¼ hp refrigerating machine having an air-cooled condenser?

$4250 \times \frac{1}{4} = 1062.5$ Btu per hour (Table 5, Chapter 8).

Chapter 9

CENTRAL SYSTEMS FOR HEATING AND HUMIDIFYING

Types of Systems, Blow-Through, Draw-Through, Heating Units, Design, Temperatures, Weight of Air to be Circulated, Temperature Loss in Ducts, Heat Supplied, Heating Units and Washer, Grate Area, Boiler Selection, Weight of Condensate, Static Pressure, Fans and Control

A FAN system of heating depends upon fans and blowers to distribute air through ducts from one centrally located plant. This chapter considers heating and humidifying systems of this type whereas similar systems arranged for cooling and dehumidifying are discussed in Chapter 10. A special type of central fan system, the mechanical warm air or fan furnace system, which is especially adapted to residences, churches, halls, and other small buildings, is covered in Chapter 24.

TYPES OF SYSTEMS

In the indirect type of central fan heating and air conditioning systems, steam is usually the medium by which heat is transferred from the boiler, or other source of heat, to the heating units. If the system is intended solely for heating, the air is passed over one or more stacks or batteries of heating units and then conveyed to the spaces for which it is intended through a system of ducts. In some cases, a predetermined amount of outside air is introduced for ventilating purposes, whereas in others the moisture content is controlled by passing the air through a washer or humidifier. If the apparatus is designed to control simultaneously the temperature, humidity, air motion, and distribution, it is known as an air conditioning system.

In the *split system*, the heating is accomplished by means of radiators or convectors, and the ventilating or air conditioning by means of the central fan apparatus. In the *combined system*, the entire operation of heating, ventilating, and air conditioning is handled by the central fan system.

A common arrangement of the central fan system of heating is illustrated by Fig. 1 and consists of a fan, a heating unit (heater) enclosed by a sheet metal casing connected with the suction side of the fan, a sheet metal casing connected to the heating unit casing run to the outside of the building and provided with an adjustable opening inside the building for recirculation of the air when desired, and a duct system attached to the fan outlet to convey and distribute the air to various parts of the building to be warmed by the apparatus. The fan is ordinarily motor-driven; there are, however, many cases when a direct-connected steam engine may be used to advantage. In this event the exhaust from the engine can be con-

nected to one or more sections of the heater, depending upon the condensation rate of the engine. The recirculation duct connected with the opening in the suction duct should be extended to a point as near the floor as possible.

When ventilation is not a requirement or is considered relatively unimportant, as in shop and factory heating, and the number of persons vitiating the air is small compared with the cubical contents of the building, or the process does not generate obnoxious gas or vapors, the air may be recirculated, sufficient outside air for ventilation being supplied by infiltra-

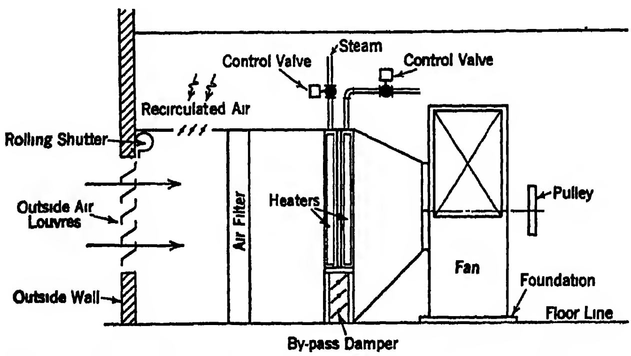


FIG 1 ARRANGEMENT OF A CENTRAL FAN HEATING SYSTEM (DRAW-THROUGH)

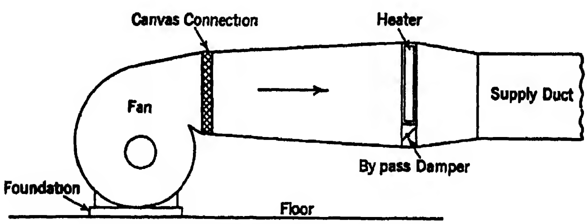


FIG. 2. ARRANGEMENT FOR HEATING UNIT (BLOW-THROUGH)

tion. The amount of heat to be supplied the heating unit in this case is the same as would be required for a direct radiation installation.

When ventilation is a requirement to be met, an arrangement similar to that shown by Fig. 1 may be employed. Since the amount of air necessary for heating is generally in excess of the amount required for ventilation, considerable fuel economy may be effected by recirculating a portion of the air. In this case only sufficient outside air is drawn into the system to meet the ventilation requirement and the remainder of the air, required for heating, is recirculated. This may be readily effected by an arrangement of ducts and dampers on the suction side of the fan as previously mentioned. If the outside air introduced is to be washed or conditioned the washer or humidifier and tempering coil may be added between the inlet for the recirculated air and the fresh air intake.

Blow-Through, Draw-Through

When the heating unit is located on the suction side of the fan, the system is known as *draw-through*. (See Fig. 1.) When the heating unit is located in the discharge from the fan, the system is known as *blow-through*. (See Fig. 2.) The draw-through combination is used for factory and toilet room installations because a more compact arrangement of the apparatus usually is possible. In addition, air leakage will be inward. The blow-through combination is used principally in schools and public buildings, and for all booster coil arrangements where different temperatures and independent temperature regulation are required for different heated spaces. In public building installations, the fan frequently blows the heated air into a plenum chamber from which the air ducts radiate to the various rooms of the building; this arrangement is sometimes called the *plenum* system.

HEATING UNITS

The heating units for central fan systems using steam as the heating medium may be classified as (1) tempering coils, (2) preheater coils, (3) reheater coils, (4) booster coils, and (5) water heaters, either open or closed. *Tempering coils* are used with ventilating and air conditioning systems for raising the temperature of the outside cold air to above freezing, or 32 F. They are not required for heating systems where all of the air is recirculated, since the temperature of the recirculated air will be above freezing. *Preheater coils* are used with air conditioning systems to raise the temperature of the air from that leaving the tempering coils to such a temperature that in passing through the water sprays of the washer (without water heater) the air will become partially saturated (adiabatically) having a moisture content corresponding to the required dew-point temperature. Preheater coils therefore supply heat as necessary to control the dew-point temperature. The *reheater coils* are used to raise the temperature of the air leaving the tempering coils (in the case of a heating or ventilating system) or the air leaving the washer (in the case of an air conditioning system) to that necessary to maintain the desired temperature in the rooms or spaces to be heated or conditioned, except where booster coils are used, in which case the reheater coils raise the air temperature to approximately room temperature, or slightly higher. *Booster coils* are installed in the duct branches to control the temperature of the air entering the rooms or spaces for which it is intended. *Water heaters* are used on an air conditioning system to control the dew-point temperature. They are used mainly for industrial work, seldom for comfort conditioning. They are not used where preheater coils are employed. The open type supplies steam directly to the spray water, while the closed type utilizes a heat interchanger by which the steam imparts its heat to the spray water. Where water heaters are required for comfort conditioning, the closed type is used.

The heating units for central fan systems in use at the present time consist either of pipe coils, finned tubes of steel, copper, brass or other metal, cast-iron sections with extended surfaces, or the cellular type. Steam is passed through these heating units and the air to be heated is passed over their exterior surfaces.

In selecting a heating unit for any particular service, the choice should be based on the desired requirements as follows:

1. Final temperature desired.
2. Loss in pressure (or friction) of air passing over the heating unit
3. Air velocity over the heating unit
4. Free area or face area of heating unit.
5. Ratio of heating surface to net free (or face) area.
6. Air volume required.
7. Number of rows of pipes, tubes, or sections.
8. Amount of heating surface
9. Steam pressure drop through the heating unit.
10 Weight of heating unit

Final Temperature Desired. The choice of a heating unit is largely influenced by the final temperature desired, when the entering air temperature and steam pressure available at the heating unit are specified. These data are obtainable from manufacturers' catalogs.

Loss in Air Pressure (or Friction). The allowable friction through the heating unit is one of the first factors to be determined in the selection of the apparatus. The velocities of air through various types of heating units will not necessarily be the same, but for any particular job the velocity through the heating unit should be a secondary consideration and the allowable friction or air pressure loss should be fixed approximately before proceeding with the selection of the heating unit. The loss in air pressure (or friction) through the heating unit should not exceed a predetermined maximum allowable amount for economical operation and for moderate size and first cost of installation.

In public building work, the maximum allowable friction through both tempering coil and reheater coils should never exceed ⅝ in. of water and it is advisable that the friction be kept considerably lower than this figure if possible. A tempering coil friction ranging from 0.10 to 0.20 in. of water is considered satisfactory. The air pressure loss for reheaters ordinarily ranges from 0.20 to 0.40 in. of water. In factory work, the maximum friction through the heater should never exceed 0.8 in. or 1 in. of water and it is advisable to figure the heaters at lower frictions if possible.

Velocity through Heating Unit. This velocity has generally been given in manufacturers' tables as being measured at 70 F and in most cases refers to the velocity through the net free area of the heating unit, or through the net space between the pipes, tubes or sections. Although most manufacturers give suitable velocities measured at 70 F, certain manufacturers show velocities measured at 65 F and others indicate velocities measured at the average air temperature through the heating unit. Many new heating units, however, specify net face areas with corresponding velocities instead of velocities through net free areas. In either case, manufacturers publish the corresponding friction or air-pressure loss in tables. The velocity through the net free area of the heating unit averages about 1000 fpm and that through the net face area about 500 fpm.

The *volume of air* to be heated in any particular case is determined after consideration of the ventilation requirements, heat losses, and quantity of air required for proper circulation, as explained in Chapters 3 and 7.

The number of *rows of pipes, tubes, or sections* or the amount of *heating surface* to be used may be selected from manufacturers' catalogs after the quantity of air handled and the heat load are known. Savings in operating expense or cost of installation should result from a proper selection of heater and by-pass areas. For example, instead of having the entire air quantity go through a one-row heating unit, it may be advantageous to use a two-row heating unit and a properly sized by-pass. Thus, when no heating is being done, a suitable by-pass damper may be opened to place a lighter load on the fan.

The *steam pressure drop through the heating unit* is also tabulated in manufacturers' data tables. The sizing of steam supply and return piping, allowing for drops through heating units, is explained in Chapter 32.

Weight of Heating Unit. In the design of a heating system, the weight limitations of heating units are determined by the location of the units. Obviously, if there is no loading limitation imposed, any type of heating unit may be selected. On the other hand if the heating unit is to be hung from the ceiling, it may be desirable to use the lightest unit which will accomplish the work required.

DESIGNING THE SYSTEM

The general procedure for the design of central fan systems is as follows:

1. Calculate the heat loss for each room or space to be heated
2. Determine volume of outside air to be introduced.
3. Assume or calculate temperature of air leaving registers or supply outlets.
4. Calculate weight of air to be circulated
5. Estimate temperature loss in duct system
6. Calculate heat to be supplied the heating units and washer.
7. Select heating units and washer from manufacturers' data and performance curves.
8. Calculate total heat to be supplied.
9. Calculate grate area and select boiler
10. Design duct system.
11. Calculate total static pressure of system.
12. Select fan, motor, and drive.

The heat losses (H) should be calculated in accordance with the procedure outlined in Chapter 7. If a positive pressure is maintained by the central fan system in the room or space to be ventilated or conditioned, there will ordinarily be very little infiltration of cold outside air through the cracks and crevices of the space. Consequently, the volume of air introduced into the space at the assumed or calculated outlet temperature need only be sufficient to provide for the transmission losses, plus about one-third of the infiltration losses. The exfiltration of heated or conditioned air through the cracks and crevices of the space should be provided for by making the usual allowance for the infiltration losses in arriving at the total heat loss of the space. The air required to make up for this exfiltration of heated or conditioned air will be brought in at the outside air intake and may be included as a part of the outside air neces-

sary for the ventilating requirements. The heat required to raise this air to the conditions maintained in the room must be provided by the tempering coils, preheater coils, and reheater coils. If a positive pressure is not maintained in the room or space to be conditioned, the normal infiltration of outside cold air will take place in this room, and the outlet temperature, together with the required air volume at this temperature, must be sufficient to provide for both infiltration and transmission losses.

Volume of Outside Air

The volume of outside air required for ventilation or air conditioning purposes may be determined from data in Chapter 3. In no case shall less than 10 cfm per person be introduced.

The heat required to warm the outside air introduced for ventilation purposes (H_o) may be determined by means of the following formula:

$$H_o = 0.24\,(t - t_o)\,M_o \tag{1}$$

where

0.24 = specific heat of air at constant pressure.
t = room temperature, degrees Fahrenheit.
t_o = outside temperature, degrees Fahrenheit.
M_o = weight of outside air to be introduced per hour, in pounds = $60\,d_oQ_o$
Q_o = volume of outside air to be introduced, cubic feet per minute.
d_o = density of air at t_o, pounds per cubic foot

Example 1. A building in which the temperature to be maintained at 70 F requires 10,000 cfm. If the outside temperature is 20 F, how much heat will be required to warm the air introduced for ventilation purposes to the room temperature?

Solution. 10,000 × 60 = 600,000 cfh; d_o = 0.08273 (Table 1, Chapter 1); M_o = 0.08273 × 600,000 = 49,656 lb; t = 70 F, t_o = 20 F, H_o = 0.24 × (70 − 20) × 49,656 = 595,872 Btu per hour

Temperature of Air Leaving Registers

If the system is to function only as a heating system, that is, entirely as a recirculating one, the temperature of the air leaving the register outlets must be assumed. For public buildings, these temperatures may range from 100 to 120 F, whereas for factories and industrial buildings the outlet or register temperature may be as high as 140 F. In no case should the outlet temperature exceed these values.

For ventilating or conditioning systems, the temperature of the air leaving the supply outlets may be estimated by means of the following formula:

$$t_r = \frac{H}{60\,d\,Q \times 0.24} + t \tag{2}$$

where

t_r = outlet temperature, degrees Fahrenheit.
H = heat loss of room or space to conditioned, Btu per hour.
Q = *total* volume of air to be introduced at the temperature t, cubic feet per minute
d = density of air, pounds per cubic foot.

If the outlet temperature (t_r) as determined from Equation 2 exceeds 120 F for public buildings, or 140 F for factories or industrial buildings,

these respective outlet temperatures should be used as factors in the following equation to determine the volume of air to be introduced into the room or space

$$Q = \frac{H}{60\, d \times 0.24\, (t_y - t)} \tag{3}$$

Example 2 The heat loss of a certain auditorium to be conditioned is 100,000 Btu per hour The ventilating requirements are 1,500 cfm and the room temperature 70 F Determine the outlet temperature

Solution Substituting in Formula 2,

$$t_y = \frac{100{,}000}{60 \times 0.07492 \times 1500 \times 0.24} + 70 = 131.7 \text{ F}$$

Inasmuch as this temperature is excessive, it will be necessary to assume an outlet temperature, which will be taken as 120 F, and to calculate the amount of air to be introduced into the room at this temperature to provide for the heat loss Substituting in Equation 3,

$$Q = \frac{100{,}000}{60 \times 0.07492 \times 0.24\,(120 - 70)} = 1850 \text{ cfm (at temperature } t)$$

Weight of Air to be Circulated

The total weight of air (M) to be introduced into the room or space to be heated or conditioned is given by the following formulae:

$$M = \frac{H}{0.24(t_y - t)} = 60\, dQ \tag{4}$$

$$M = M_o + M_r \tag{5}$$

$$M_o = 60\, d_o Q_o \tag{6}$$

where

d = density of air at temperature t, pounds per cubic foot
d_o = density of air at temperature t_o, pounds per cubic foot.
Q_o = volume of outside air at temperature t_o, cubic feet per minute
M_o = weight of outside air, pounds per hour
M_r = weight of recirculated air, pounds per hour

Example 3 Using the data of Example 2 and an outside temperature of 20 F, what will be the values of M, M_o and M_r?

Solution $d = 0.07492$, $d_o = 0.08273$, $Q = 1850$, $Q_o = 1500$, $H = 100{,}000$

$$M = \frac{100{,}000}{0.24 \times (120 - 70)} = 8{,}333 \text{ lb}$$

$$M_o = 0.08273 \times 60 \times 1500 = 7{,}448 \text{ lb}$$

$$M_r = M - M_o = 8{,}333 - 7{,}448 = 885 \text{ lb}$$

Temperature Loss in Ducts

The allowances (t_z) to be made for temperature drop through the duct system are as follows:

1. When the duct system is located in the enclosure to which the air is being delivered, as in a factory, it may be assumed that there is no loss between the reheater coil and the point or points of discharge into the enclosure

2. For ducts run underground an allowance shall be made based on the estimated heat loss of the duct, assuming the average temperature of the ground to be 55 F

3 For galvanized ducts with the usual ranges of air temperature and velocity, the coefficient of heat transmission may be taken as 1 7 Btu per hour per degree difference between the mean temperature of the air in the duct and that surrounding the duct.

The heat loss may then be expressed by

$$H = 1.7\ \pi DL \left[\left(\frac{t_1 + t_y}{2}\right) - t_o\right] \tag{7}$$

and also by

$$H = 0.24\ M\ (t_1 - t_y) = 60\ \frac{\pi}{4}\ D^2 Vd \times 0\ 24\ (t_1 - t_y) \tag{8}$$

Equating (7) and (8)

$$1.7\ \pi\ DL \left[\left(\frac{t_1 + t_y}{2}\right) - t_o\right] = 3.6\ \pi\ D^2 Vd\ (t_1 - t_y)$$

$$\frac{t_1 + t_y - 2t_o}{t_1 - t_y} = \frac{4\ 235\ DVd}{L} \tag{9}$$

where

H = heat loss from the duct, Btu per hour
D = diameter of duct, feet
L = length of duct, feet.
t_1 = temperature of air entering the duct
t_y = temperature of air leaving the duct.
t_o = temperature of air surrounding the duct
M = weight of air passing a given cross section of the duct per hour.
V = velocity of air in the duct, feet per minute, at specified temperature
d = density of the air at the specified temperature at which V is measured.

Usually all the values in Formula 9 may be approximated except t_1, the initial or entering temperature, which can be readily found where the others are known or assumed.

Example 4. Determine the temperature drop in a galvanized duct 20 in diameter and 60 ft long carrying air at a velocity of 1200 fpm measured at 70 F, to be delivered at a temperature of 140 F when the air surrounding the duct is at a temperature of 50 F.

Solution. Substituting in Formula 9,

$$\frac{t_1 + 140 - (2 \times 50)}{t_1 - 140} = \frac{4\ 235 \times 1\ 666 \times 1200 \times 0\ 07492}{60}$$

$$t_1 = 158.7\ \text{F}$$

$$\text{temperature drop} = 158.7 - 140 = 18.7\ \text{F}$$

Example 5. An uninsulated 12 in. diameter galvanized duct extends 50 ft through an unconditioned room to supply 80 F cool air to an adjacent space. If the average temperature of the air surrounding the duct in the unconditioned room is 100 F and the velocity of the air through the duct is 1000 fpm, measured at 70 F, determine the temperature gain which must be allowed in passing the air through the unconditioned room.

Solution Substituting in Formula 9,

$$\frac{t_1 + 80 - (2 \times 100)}{t_1 - 80} = \frac{4.235 \times 1.0 \times 1000 \times 0\ 07492}{50}$$

$$t_1 - 120 = 6.34\ (t_1 - 80)$$

$$-5\ 34\ t_1 = 120 - 507$$

$$t_1 = 72.3 \text{ F}$$

$$\text{temperature gain} = 80 - 72.3 = 7.7 \text{ F}$$

Therefore, air having a temperature of 72.3 F must be introduced at the end of the 50 ft duct in order to supply 80 F air to the conditioned space

Heat Supplied Heating Units and Washer

The following cases may arise in practice:

A. The heating of the building is done entirely by means of a central fan system, all of the air being drawn from the outside.

B Similar to *A*, except that all of the air is recirculated.

C A portion of the air is recirculated, and the remainder is drawn in from the outside

D Air at the same temperature is to be delivered to all the rooms A constant relative humidity is maintained in the building and all of the air circulated is drawn from outside the building (Not applicable to the heating of various rooms where individual control of each room is desired)

E. Outside air, return air, and by-pass air are used with the reheater located in by-pass air chamber.

F. Arrangement of apparatus where individual control of the temperature for each room is required in conjunction with air washer equipment to maintain a constant relative humidity in the rooms. The air washer is provided with a water heater for the spray water, capable of fully saturating the air. A section of preheater may be used for this purpose in place of the water heater With this arrangement and with a uniform temperature of air entering the rooms, it is impossible to maintain the same room temperature throughout the building because the weight of air to be delivered to each room is determined and fixed by the ventilating requirements

In analyzing these cases, the following symbols will be used.

H = heat loss of the room or building, Btu per hour

H_1 = heat to be supplied to the reheater coil, Btu per hour

H_2 = heat supplied tempering coil, or tempering coil and preheater, Btu per hour.

H_3 = heat supplied air washer by water heater, Btu per hour.

H_4 = heat to be supplied booster coil, Btu per hour

M = weight of air to be introduced into the room or building, pounds per hour.

M_r = weight of recirculated air, pounds per hour

M_b = weight of air by-passing washer, pounds per hour.

M_o = weight of air drawn in from outside, pounds per hour.

t_o = mean temperature of outside air, degrees Fahrenheit.

t = mean air temperature to be maintained in the room or building, degrees Fahrenheit

t_1 = mean temperature of the air entering the reheater coil.

t_2 = mean temperature of the air leaving the reheater coil

t_z = temperature loss in the duct system.

t_y = temperature of the air leaving the duct outlets

t_x = average temperature of air entering tempering coil.

t_w = temperature of air entering washer.

0.24 = specific heat of air at constant pressure.

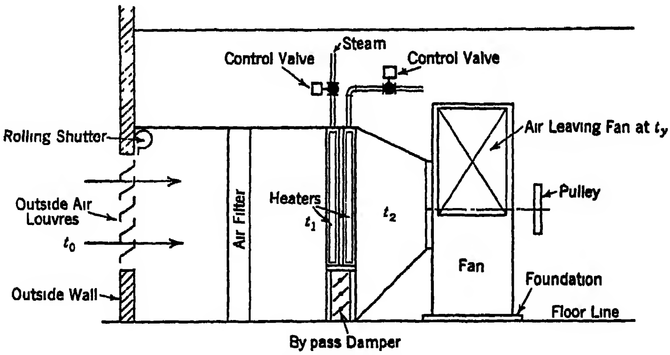


FIG. 3 HEATING UNIT AND FAN ARRANGED FOR OUTSIDE AIR CIRCULATION (*Case A*)

Case A (Fig 3) All of the air circulated to be drawn from outside the building, in which case $t_x = t_0$

$$H_2 = 0.24\ (t_1 - t_0)\ M_0 \tag{10}$$

$$H_1 = 0.24\ (t_2 - t_1)\ M_0 \tag{11}$$

Example 6. The heat loss H for a certain factory building is 700,000 Btu per hour. The mean inside temperature t to be maintained is 65 F. The assumed outside air temperature t_0 is 0 F, $t_x = 0$, $t_y = t_2$ and is assumed to be 140 F. The temperature leaving the tempering coil is assumed to be 35 F. Required, H_1 and H_2 From Equation 4,

$$M = \frac{700{,}000}{0.24\ (140 - 65)} = 38{,}889 \text{ lb per hour.}$$

$H_2 = 0.24 \times (35 - 0) \times 38{,}889 = 326{,}667$ Btu per hour
$H_1 = 0.24 \times (140 - 35) \times 38{,}889 = 980{,}003$ Btu per hour
$H_2 + H_1 = 326{,}667 + 980{,}003 = 1{,}306{,}670$ Btu per hour

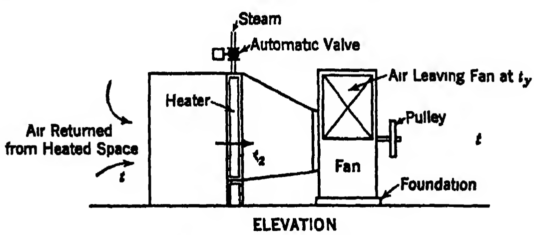


FIG. 4 ARRANGEMENT FOR RECIRCULATION (*Case B*)

Case B (Fig. 4) All of the air is to be recirculated, in which case $t_1 = t$

$M_r = 38{,}889$ lb
$H_1 = 0.24\ (t_2 - t_1)\ M_r$
$H_1 = 0.24\ (140 - 65) \times 38{,}889 = 700{,}000$ Btu per hour.

This example illustrates the saving in fuel consumption by the recirculation of the air. The heat to be supplied the apparatus is the same as that required for a direct system of heating and is equal to the heat loss of the building ($H_1 = H$); in the example 700,000 Btu per hour as compared with 1,306,670 for *Case A*.

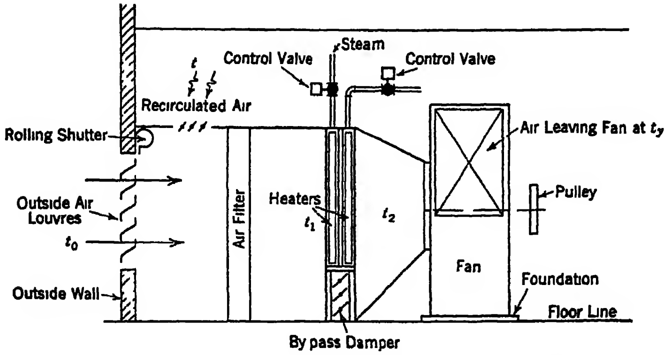


FIG 5. COMBINATION OF RECIRCULATED AIR AND OUTSIDE AIR (*Case C*)

Case C (Fig 5) A portion of the air circulated is recirculated air and the remainder, as may be required for ventilating purposes, is drawn in from the outside According to Equations 4 and 5,

$$M = M_o + M_r = \frac{H}{0.24\ (t_y - t)}$$

The temperature of the resulting mixture of outside and recirculated air entering the tempering coil is

$$t_x = \frac{M_o t_o + M_r t}{M} \tag{12}$$

Example 7. Assuming that a positive supply of outside air (d_o = 0.08633) is required for ventilation at the rate of 90,000 cu ft per hour in the preceding example, then M_o = 0 08633 × 90,000 = 7776 lb per hour are required, measured at 65 F

$$M_r = M - M_o = 38{,}889 - 7776 = 31{,}113 \text{ lb}$$

$$t_x = \frac{7776 \times 0 + 31{,}113 \times 65}{38{,}889} = 52 \text{ F}$$

$$H_1 = 38{,}889 \times 0.24\ (140 - 52) = 821{,}336 \text{ Btu.}$$

This amount of work may be accomplished with one or more banks of heating units, that is, either a single reheater or a tempering coil and reheater.

The three preceding cases refer to installations in which conditioning the air to maintain certain relative humidity requirements does not enter into the problem, as for example, certain types of industrial installations. In practically all modern public buildings, theaters, schools, and in many industrial installations, the ventilating requirements include the provision for washing and humidifying the air delivered to the various rooms of the structure.

In the following cases it is assumed that in addition to maintaining a mean room temperature t, the heating and ventilating apparatus is required to maintain a constant relative humidity in the rooms.

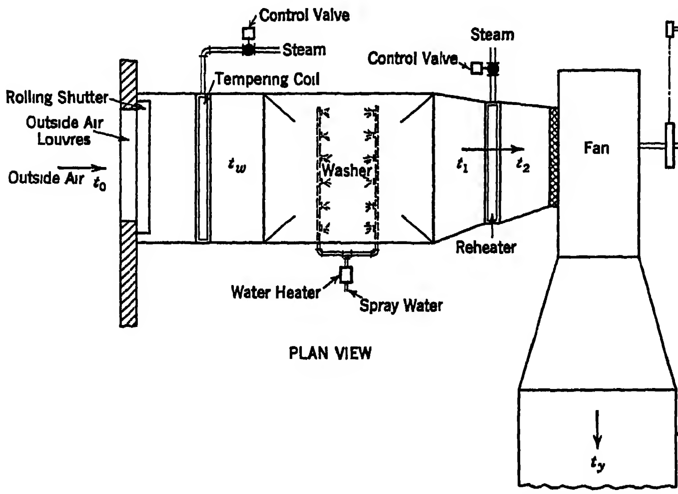


FIG. 6. OUTSIDE AIR CIRCULATED: CONSTANT RELATIVE HUMIDITY IN ROOM (*Case D*)

Case D (Fig 6) The maximum relative humidity that may be maintained within the building without the precipitation of moisture on single glazed sash when the outside temperature is 30 F is approximately 35 per cent If the inside temperature t is 70 F, 35 per cent relative humidity corresponds to a dew-point temperature of 41 F. (See psychrometric chart)

The installation shown in Fig 6 contemplates the use of a tempering coil, an air washer provided with a water heater, and a reheater The tempering coil, one section in depth, warms the incoming air to approximately 35 F to prevent freezing any of the spray water. The air passing through the spray chamber is saturated and leaves at a temperature of $t_1 = 41$ F.

The heat to be supplied the reheater is

$$H_1 = 0.24\,(t_2 - 41)M \text{ Btu per hour.}$$

The heat to be supplied the tempering coil is

$$H_2 = 0.24\,(35 - t_0)M \text{ Btu per hour.}$$

The amount of heat, per pound of air circulated, to be supplied the humidifying washer or humidifier is the difference between the heat content of the assumed dry air entering the washer at a temperature of $t_w = 35$ F and that of the leaving saturated air at $t_1 = 41$ F (Table 6, Chapter 1), or.

$$15.657 - 8.397 = 7.26 \text{ Btu per pound of dry air.}$$

The amount of heat required for the washer is.

$$H_3 = 7.26\ M \text{ Btu per hour.}$$

The total amount of heat required by the apparatus is, therefore:

$$H_1 + H_2 + H_3 \text{ Btu per hour.}$$

If a washer having a ***humidifying efficiency*** of 67 per cent ***without water heater*** is employed it will be necessary to heat the outside air drawn into the apparatus by means of the tempering and preheater coils to such a temperature that the air in passing through

the water sprays will become partially saturated (adiabatically) having a moisture content per pound of air equal to saturated air at 41 F If the incoming air is warmed to t_w = 88 F (requiring a two-section-depth heating unit) it will be cooled in the washer to 64 F, with a temperature drop of 88 − 64 = 24 deg.

If the *humidifying efficiency* of the washer were 100 per cent, the air would become adiabatically saturated at 52 F after a temperature drop of 88 − 52 = 36 F The efficiency of the washer is, however, only 67 per cent, so that the actual temperature drop will be 0.67 × 36 deg or 24 deg, as used

The heat to be supplied the reheater is in this case $H_1 = 0\,24\ (t_2 - 64)\ M$ Btu per hour, and the heat to be supplied to the tempering coil and preheater is $H_2 = 0\,24\ (88 - t_0)\ M$ The total heat required by the apparatus is $H_1 + H_2$, no heat being supplied to the washer

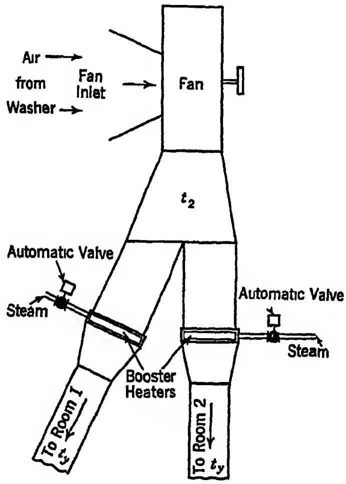


Fig 7. Outside Air Circulated, Constant Temperature and Relative Humidity Maintained in Each Room (*Case E*)

Case E. (Fig. 7) The temperature t_y will ordinarily be different for each room. With H and M fixed, $0\,24\ (t_y - t)M = H$, or

$$t_y = \frac{H}{0\,24\ M} + t$$

In order to provide the proper temperature for each room, a booster coil is generally installed in each supply duct near the outlet to control the outlet temperature t_y. The amount of steam supplied to these booster units is usually controlled automatically by individual thermostats. The heat required by the booster coils depends on the temperature range through which the air is heated and the quantity of air, or

$$H_4 = 0\,24\ (t_y - t_1 - t_z)M \tag{13}$$

Heat to be Supplied

The amount of heat to be supplied (H^1) is equal to the sum of the heat requirements of the various heating units and the water heater of the washer, if any, plus the allowance for piping tax. (See preceding *Cases A* to *E*.)

Grate Area, Boiler Selection

The required grate area may be determined by the following formula.

$$G = \frac{H}{F \times E \times C} \tag{14}$$

where

G = required grate area, square feet
F = calorific value of fuel, Btu per pound.
C = combustion rate, pounds per square foot of grate per hour
E = boiler and grate efficiency, per cent.

Example 8. Using the data in Example 5, and assuming coal having a calorific value of 12,000 Btu per pound, a combustion rate of 7 lb per square foot, and a performance efficiency of 0.60, and neglecting the piping tax,

$$G = \frac{1{,}306{,}670}{12{,}000 \times 0.60 \times 7} = 26 \text{ sq ft}$$

Weight of Condensate

The normal weight of condensate to be handled from central fan systems may be estimated by means of the following formula:

$$W = \frac{60\, dQ \times 0.24 \times \Delta t}{h_{fg}} \tag{15}$$

where

W = weight of condensate, pounds per hour.
Q = total volume of air, cubic feet per minute
Δt = temperature rise of air, degrees Fahrenheit.
h_{fg} = latent heat of steam in the system, Btu per pound.

Ducts and Outlets, Air Filters, Air Washers

The design of the duct system should be based on data contained in Chapter 20. Air washers and humidifiers are described in Chapter 12. For information on air filters, see Chapter 16.

Static Pressure

The total static pressure against which the system must operate may be found by summing up the static losses through the complete system from the outside air intake to the discharge outlets or nozzles. This means that the loss due to friction must be determined for each piece of apparatus involved. Most of these values may be obtained from manufacturers' data tables. For a simple system, the following static pressure drops may be assumed:

1. Outside air inlet, comprised of screen, louver and short duct, may have a loss of 0.2 in. of water.
2. A typical oil filter at rated capacity and velocity has a drop of 0.25 in. of water
3. The loss of one row of a standard make tempering stack equals 0.09 in. water.
4. The loss of one row of a standard make preheater equals 0 10 in. water.
5. A standard humidifier at rated velocity may have a loss of about 0.35 in. water.
6. The loss through one row of a standard make reheater equals 0.12 in. water.
7. A fair assumption for duct losses on a simple system is 0 25 in. water
8. The static pressure for a nozzle type outlet may be taken as 0 1 in water.

The sum of these values equals 0.2 + 0.25 + 0.09 + 0.10 + 0.35 + 0.12 + 0.25 + 0.1 = 1.46 in. which is the static pressure against which the system must operate.

Fans and Control

The selection of fans may be based on data contained in Chapter 17 and for motors in Chapter 42. Because centrifugal fans reach their maximum efficiency when working against the resistance offered by the average central fan heating system, they are well adapted to such systems and are generally used. Information on temperature control for central fan systems is given in Chapter 14.

PROBLEMS IN PRACTICE

1 ● Consider a blast heating system handling 10,000 cfm. The resistance to air flow offered by one coil arrangement is 0.9 in. of water and by another coil arrangement is 0.2 in. of water. The fan operates 4000 hours per year and the combined efficiency of motor and fan is 60 per cent. Determine the annual energy saving if the second coil is used.

Difference in system resistance = 0.9 − 0.2 = 0.7 in. of water

$$\text{Reduction in power input} = \frac{10,000 \times 0.7}{6356 \times 0.60} = 1.83 \text{ hp}$$

Annual energy saving = 1.83 × 0.764 × 4000 = 5480 kwhr.

2 ● What saving results from recirculating some of the room air and reducing the amount of outside air?

Because outside air must be heated to room temperature, reducing the amount of outside air produces a proportionate saving in heat or fuel.

3 ● What items make up the total heating load in a central fan heating system?

1. The net heat loss from the conditioned space.
2. The heat required for evaporation of water for humidification.
3. The heat required to raise the temperature of outside air to room temperature.
4. Heat losses from pipes and ducts.

4 ● Why is it necessary to determine the total static pressure of a central fan heating system?

To select a fan of maximum efficiency and to determine the power required to operate the fan.

5 ● A group of three drafting rooms, having a total volume of 27,000 cu ft, a transmission loss of 110,100 Btu per hour, and an infiltration loss of 34,200 Btu per hour on the basis of 0 F outdoors and 70 F room temperature, is to be heated by a recirculating hot blast heating system with air entering the rooms at 116 F. How many cubic feet per minute, measured at 70 F, will be required?

Substitute in Equation 3. H = 110,100 + 34,200 = 144,300 Btu per hour, t_y = 116 F;

$$t = 70 \text{ F}, Q = \frac{144,300}{60 \times 0.07492 \times 0.24 \,(116 - 70)} = 2900 \text{ cfm.}$$

6 ● In the preceding question, if the hot air loses 4 F between heater and rooms, how many pounds of steam per hour at 1-lb gage will the heating sections condense?

Substitute in Equation 15 Q = 2900 cfm, from solution of Question 5, $\Delta t = 116 + 4 - 70 = 50$ F, h_{fg} = 968 Btu, from steam table in Chapter 1.

$$W = \frac{60\, dQ \times 0\,24 \times \Delta t}{h_{fg}} = \frac{60 \times 0\,07492 \times 2900 \times 0\,24 \times 50}{968} = 161\,8 \text{ lb per hour.}$$

7 ● The same rooms are converted to chemical laboratories, requiring the introduction of 12 changes of outside air, measured at 70 F, per hour to permit the exhaust fans connected to the chemical hoods to maintain only a slight negative pressure in the rooms. At what temperature must the air enter the rooms to maintain 70 F with 0 F outside?

Substitute in Equation 2. $H = 110{,}100 + 34{,}200 = 144{,}300$ Btu per hour, $Q = \frac{12}{60} \times 27{,}000 = 5400$ cfm, $t = 70$ F, $t_y = \frac{H}{60\, dQ \times 0\,24} + t = \frac{144{,}300}{60 \times 0\,07492 \times 5400 \times 0\,24} + 70 = 94.7$ F.

8 ● In the preceding question, if the air drops 2 F between the heater and the rooms, how many pounds of steam per hour at 1-lb gage will the heating system condense?

Substitute in Equation 15. Q = 5400 cfm, $\Delta t = 94\,7 + 2 = 96.7$ F, from solution of Question 7; h_{fg} = 968 Btu, from steam table in Chapter 1.

$$W = \frac{60\, dQ \times 0\,24 \times \Delta t}{h_{fg}} = \frac{60 \times 0.07492 \times 5400 \times 0.24 \times 96\,7}{968} = 583 \text{ lb per hour.}$$

9 ● The combination hot blast heating and ventilating system for the dining rooms of a hotel is to heat the rooms to 70 F with 0 F outside, and permit the exhaust fan from the adjoining kitchen to draw 5000 cfm from the dining rooms. The transmission losses from the dining rooms total 240,000 Btu per hour. The infiltration into the dining rooms amounts to 1000 cfm from outdoors and 1000 cfm from heater rooms. How many cubic feet per minute, measured at 70 F, must be supplied the dining rooms if the air enters at 112 F?

First find the infiltration loss by substituting in Equation 1

$t = 70$ F, $t_o = 0$; $M_o = d \times Q = 0\,07492 \times 60 \times 1000 = 4497$ lb per hour. In this case d and Q are figured at 70 F. $H_o = 0.24\,(t - t_o)\, M_o = 0\,24\,(70 - 0) \times 4497 = 75{,}550$ Btu per hour

Next by substituting in Equation 3, find the cubic feet per hour to be circulated H = sum of transmission and infiltration losses in room = 240,000 + 75,550 = 315,550 Btu per hour; $t_y = 112$ F, $t = 70$ F,

$$Q = \frac{H}{60\, d \times 0.24\,(t_y - t)} = \frac{315{,}500}{60 \times 0.07492 \times 0.24\,(112 - 70)} = 6950 \text{ cfm.}$$

10 ● In Question 9, 3000 cfm of outside air will be drawn in by the supply fan and 3950 cfm will be recirculated. What will be the output of the heating sections in Btu per hour if there is a loss of 2 F between the heaters and the room?

The average temperature of the mixture of outdoor and recirculated air entering the heater $= \frac{3000 \times 0 + 3950 \times 70}{6950} = 39.8$ F. Air leaves the heater at $112 + 2 = 114$ F.

Referring to Equation 15, $W \times h_{fg}$ = amount of heat required per hour $= 60\, dQ \times 0.24 \times \Delta t = H$ Q = 6950 cfm; $\Delta t = 114 - 39\,8 = 74\,2$ F $H = 60 \times 0\,07492 \times 6950 \times 0\,24 \times 74.2 = 557{,}000$ Btu per hour.

Chapter 10

CENTRAL SYSTEMS FOR COOLING AND DEHUMIDIFYING

Types of Systems, Dehumidifiers, Designing the System, Zoning, Location of Apparatus, Temperature of the Air Leaving Room Inlets, Air Quantity Required, Heat to be Removed by Cooling and Dehumidifying Apparatus, Size of Reheaters, Surface Cooling Problem, Auxiliary Equipment

CENTRAL systems, equipped for cooling and dehumidifying, are used principally in the air conditioning of theaters, restaurants, office buildings, or other places where people gather, and in manufacturing establishments where air conditions have an important influence on the quality of product or rate of production. A central cooling and dehumidifying plant is one in which the fans, dehumidifiers, and other related apparatus are assembled in suitable apparatus rooms from which supply and return ducts lead to the conditioned spaces. The design of such systems is considered in this chapter, while in Chapter 9 central systems for heating and humidifying are described. Air conditioning for industrial processes is considered in Chapter 40.

TYPES OF SYSTEMS

Dehumidification or cooling of air may be accomplished by several methods and by use of many heat transfer mediums. Most comfort-conditioning, central station, air-conditioning systems employ cold water or the direct expansion of a refrigerant in either spray type or surface type equipment to accomplish the required cooling and dehumidification. Among the several other methods that may be employed are: passing the air through or over a dehydrating agent and then lowering the dry-bulb temperature to the proper level, and evaporative cooling. With the former method the excess sensible heat may be removed with cold water and where this is not available, mechanical refrigeration must be used after the air has been dehydrated. The latter method is applicable to comfort conditioning only in regions where the summer wet-bulb temperature is low.

If the system is intended solely for summer conditioning, the apparatus will consist essentially of a dehumidifier of the surface type or spray type; filters; fan and motor, reheater; outside air, return air, and supply air duct work; air inlets and outlets; spray pump for spray dehumidifier; refrigera-

tion equipment, and suitable controls. Generally, however, a central station air conditioning system is designed for year-round service. This means that properly sized heaters and humidifiers, with their respective controls, must be added. With few exceptions, systems designed to meet summer capacity requirements will have ample capacity for winter and intermediate season conditioning.

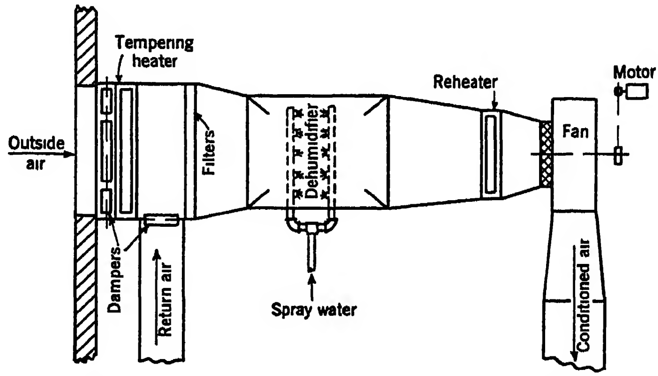


Fig 1. Spray Type Air Conditioning Apparatus

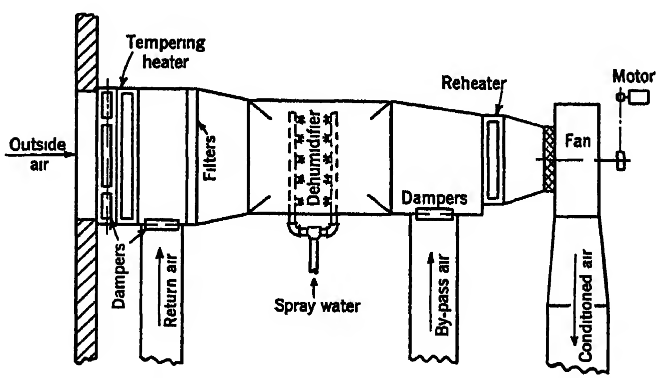


Fig 2 Spray Type Air Conditioning Apparatus with By-Pass

A common arrangement of a central station spray type system for cooling and dehumidifying is illustrated in Fig. 1. The plant may be designed to condition 100 per cent outside air, 100 per cent return air, or a mixture of outside and return air. Further, part of the air returned from the conditioned space may be by-passed[1] around the conditioner as illustrated in Fig. 2. The reheater may be installed in the fan inlet chamber as shown, in the by-pass air duct, or in the fan discharge duct,

[1]Patents exist covering the use of the by-pass for cooling and dehumidifying systems

depending upon apparatus space and other design conditions. Still another arrangement of equipment will result if the dehumidified air fan delivers the conditioned air to several other fans rather than to the conditioned space directly. These booster fan equipments may use part by-pass air as illustrated in Fig. 3 or 100 per cent dehumidified air and reheaters. The main apparatus, in either case, may or may not have a by-pass connection, depending on load conditions and other design factors.

The systems illustrated in Figs. 1 and 2 may be converted into the surface cooling type by merely replacing the dehumidifiers with surface cooling coils which use cold water or direct expansion of refrigerant to accomplish the required cooling and dehumidifying. The coils may also be installed within the spray chamber, either in series with the sprays or below them.

DEHUMIDIFIERS

Information on spray type dehumidifiers is given in Chapter 12.

Surface cooling type dehumidifiers generally consist of extended-surface

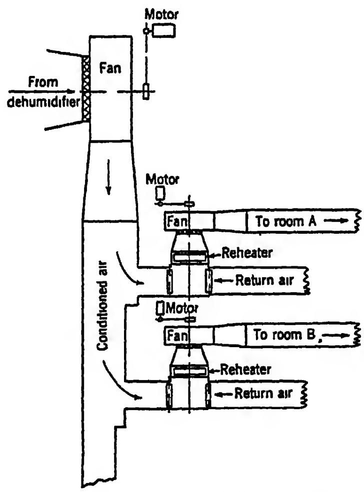


Fig. 3. Central Dehumidifying Plant and Local Recirculating Fans

coils within which the water or refrigerant is circulated or the refrigerant is expanded. The air to be cooled and dehumidified is drawn or blown over the coils. This system is generally comparatively low in initial cost and has low operating costs. Some localities have refrigeration codes which restrict the use, in comfort conditioning applications, of refrigerants acting by direct expansion in coils exposed to the air stream. Therefore, local codes should be consulted by the designer before he plans a system employing direct-expansion methods. Close humidity control cannot be maintained during the cooling season by the surface cooling type of equipment. Winter humidification may be accomplished by use of

evaporating pans or spray nozzles. The cooling coils serve no purpose during the intermediate or heating seasons, so in this respect the spray type equipment is often preferred, in that during certain seasons evaporative cooling will be sufficient to produce the cooling desired. Effective cooling and dehumidification accomplished by surface units are dependent upon many variable factors. The air velocity through the unit, air temperature, moisture content of the air, water or refrigerant temperature, and velocity of the water or refrigerant through the tubes must be considered in selecting the proper unit for a given design load. If any of these factors vary without a corresponding variation of the other factors, the effective work of the coil will increase or decrease, as the case may be.

DESIGNING THE SYSTEM

The general procedure for the design of a central cooling and dehumidifying system is as follows.

1 Calculate the heat gain for each room or space to be conditioned (See Chapters 6 and 8)

2 Establish the temperature of air leaving the supply inlets

3 Calculate the quantity of air to be circulated

4. Estimate the temperature rise in the duct system

5 Determine the volume of outside air to be introduced. (See Chapter 3)

6 Calculate the heat to be removed by the cooling and dehumidifying apparatus.

7 Calculate the size of the reheating equipment

8 Select cooling equipment and heating equipment from manufacturers' data and performance curves

9 Design the air distribution system and the air outlets and inlets (See Chapters 19 and 20)

10 Calculate the total static pressure of the system.

11 Select the fan, motor, and drive (See Chapters 17 and 42)

12 Select the pump and motor

13. Design the control system (See Chapter 14.)

ZONING

The above general outline of procedure will prove satisfactory for the smaller and less complex installations. However, when dealing with air-conditioning systems for large buildings, after a proper analysis has been made of the conditions to be maintained and the heat loads encountered, it is generally considered best practice to divide the complete job into a number of suitably sized units. In some cases a unit per floor or group of floors may complete the design satisfactorily, whereas in others it may be advantageous to have separate units for each of the various outside exposures of the building. Where the floor area is large in relation to the outside wall exposure, it is obvious that provision must be made for the variable load to which the outside exposures are subjected. The heat loads on inside rooms are apt to be less variable since the fluctuations of the outside weather conditions are not directly involved. Such conditions often result in the natural zoning or segregation of rooms having similar exposures and internal heat loads. Variation in the hours of occupancy in different portions of a building also frequently require careful zoning for successful operation.

LOCATION OF APPARATUS

Availability of space for apparatus and duct work is of primary importance when selecting the type of system for a given design. In general, for large installations, the refrigeration equipment, because of its size, weight, and operating characteristics, is located in the basement along with the boilers, fire pumps, and other equipment. The air conditioning apparatus is generally located where clean outdoor air is readily available, the designer bearing in mind that supply and return air ducts, steam connections, water and drain connections, and electrical connections must be made to the equipment proper.

TEMPERATURE OF AIR LEAVING ROOM INLETS

In comfort conditioning applications, air has been distributed from properly designed inlets without producing drafts at temperatures varying from approximately 5 to 30 deg below the required room temperature. Factors influencing the design and selection of air inlets are: ceiling height, contour of ceiling, length of blow, and temperature and quantity of air to be distributed. Most summer conditioning installations are designed to supply the air to the conditioned space at from 8 to 18 deg below room temperature. Recently the use of specially designed nozzles has indicated the possibility of reducing the air quantity necessary to dissipate a given heat load by introducing the air into the room as much as 30 deg below room temperature. Directional flow inlets which spread the air fanwise permit lower inlet temperatures than single direction inlets. Comfort conditioning systems employing differentials greater than 18 deg require special consideration and design experience because high pressure inlets or nozzles are usually used. Further, care must be taken to allow a sufficient air quantity under all load conditions to insure good distribution. If winter heating, as well as summer conditioning, is to be accomplished by the same distributing system, the design of the inlets will be influenced as discussed in Chapter 9. Industrial systems in which drafts are not objectionable usually employ a temperature differential equal to the dew-point depression

AIR QUANTITY REQUIRED

For calculating the quantity of air required to absorb a given heat gain, the following approximate formulae may be used

$$M = \frac{H_s}{60 \times 0.24 \times (t - t_y)} = dQ \tag{1}$$

or, assuming a constant value of 0 075 lb for d,

$$Q = \frac{H_s \times 55\ 2}{60 \times (t - t_y)} \tag{2}$$

where

Q = volume of air required, cubic feet per minute

H_s = total sensible heat gain, Btu per hour.

t = room temperature, degrees Fahrenheit.

t_y = inlet temperature, degrees Fahrenheit

M = weight of air required, pounds per minute.

d = density of air at the temperature and relative humidity of the room, pounds per cubic foot

Example 1 The total sensible heat gain in a restaurant when held at 80 F is 199,736 Btu per hour. Assuming a 12 deg F temperature differential between the entering air and the room temperatures, which is the same as assuming the dry-bulb temperature of the entering air to be 68 F, calculate the required air capacity of the system

Solution

$$Q = \frac{199{,}736 \times 55\ 2}{60 \times 12} = 15{,}313 \text{ cfm} = 1146 \text{ lb per minute.}$$

If a system similar to the one shown in Fig. 1 is used, 1146 lb per minute will be the capacity of the dehumidifier as well as of the fan equipment.

Example 2. If in addition to the 199,736 Btu per hour sensible heat load, the conditioned space has a moisture gain of 384,000 grains per hour, calculate the apparatus dew point required to give maintained conditions of 80 F dry-bulb and 65 F wet-bulb, with a corresponding 56½ F dew point.

Solution With 384,000 grains of moisture per hour to be picked up, the entering dew-point temperature should be low enough so that the addition of this moisture will not increase the dew point above 56½ F

Grains per pound of air saturated at 56½ F (Table 6, Chapter 1)		68.0
Less: Grains per pound to be picked up,	$\frac{384{,}000}{1146 \times 60}$,	5 6
Grains per pound allowable in entering air		62 4

This corresponds to an apparatus dew-point temperature of 54 17 F.

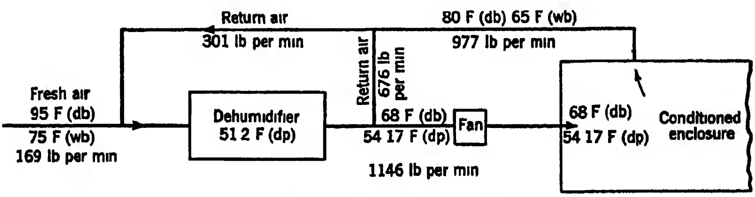


FIG. 4. DIAGRAM OF BY-PASS METHOD

Example 3. Illustration of the by-pass system (See Fig. 4.)

Assume the same data as for Example 2. Instead of passing all of the air through the dehumidifier for cooling and dehumidifying, a portion may be passed through and the balance be mixed with the conditioned air at the leaving end of the dehumidifier, the mixture being proportioned so that the resultant conditions will be those required to give proper conditions in the area considered.

Solution. The quantity of air to be dehumidified, the quantity to be by-passed, and the apparatus dew-point temperature may be approximately calculated as follows

Let

X = percentage of air to be by-passed.
Y = percentage of air to be passed through the dehumidifier.
t_d = apparatus dew-point temperature, degrees Fahrenheit.

The quantity X of 80 F air must mix with the quantity Y of dehumidified air to produce air with a resultant 68 F dry-bulb temperature. Also, X quantity of air at 56½ F dew point must be mixed with Y quantity of dehumidified air to give a resultant dew-point temperature of the mixture of 54.17 F. It is assumed that the air passing through the dehumidifier is saturated

Solving simultaneous equations,

$$
\begin{array}{rcll}
80.0X + Yt_d & = & 68.00 & (3) \\
56.5X + Yt_d & = & 54.17 & (4) \\
\hline
23.5X + 0 & = & 13.83 &
\end{array}
$$

$$X = \frac{13.83 \times 100}{23.5} = 59 \text{ per cent, air by-passed}$$

$$Y = 100 - X = 41 \text{ per cent, air passed through washer}$$

The second step is to determine the apparatus dew-point temperature. Substitute X in either Equation 3 or Equation 4, and solve for t_d

$$80 \times 0.59 + t_d \times 0.41 = 68$$

$$t_d = \frac{68 - 47}{0.41} = 51.2 \text{ F, the apparatus dew point}$$

HEAT TO BE REMOVED BY COOLING AND DEHUMIDIFYING APPARATUS

Example 4. Assume the same data as for Example 3. If the amount of outside air, at 95 F dry-bulb and 75 F wet-bulb, required for ventilation has been found to be 169 lb per minute, determine the refrigeration capacity required.

Solution As the total weight of the air introduced per minute is 1146 lb, and 41 per cent of it goes through the dehumidifier, the total work to be done may be computed as follows

Air passing through dehumidifier, 1146 × 0.41	470 lb
Less. Outside air for ventilation	169 lb
Return air	301 lb

The refrigeration required for the return air is.

Total heat per pound at 65 F	29.96 Btu
Less: Total heat per pound at 51.2 F	20.92 Btu
Requirement for cooling 1 lb of return air	9.04 Btu

301 lb × 9.04 Btu = 2721 Btu per minute required to cool the return air.

The refrigeration required for the outside air is

Total heat per pound of outside air	38.46 Btu
Less. Total heat per pound at 51.2 F	20.92 Btu
Requirement to cool 1 lb of outside air	17.54 Btu

169 lb × 17.54 Btu = 2964 Btu per minute required to cool the outside air.

Thus, the total refrigeration required is:

2721 Btu + 2964 Btu = 5685 Btu per minute, which is equivalent to a load of 28.4 tons of refrigeration.

SIZE OF REHEATERS

A properly designed air-conditioning system will have reheaters of sufficient capacity to heat the conditioned air from the apparatus dew-point temperature to the inlet delivery temperature. If winter heating is to be accomplished, consult Chapter 9.

The following general formula may be used to determine the amount of heat necessary to reheat a given quantity of air:

$$H_l = 0.24\ (t_y - t_d)\ M \qquad (5)$$

where

H_l = heat to be supplied to reheater coil, Btu per hour.

Example 5. Assume the same data as for Example 1, and find the amount of reheating required.

Solution.

$$H_l = 0.24\ (68 - 54.17)\ 1146 \times 60 = 228{,}200 \text{ Btu per hour.}$$

SURFACE COOLING PROBLEM

The amount of coil surface required for a given amount of work is dependent upon factors previously listed. Obviously, the various types of surfaces made available by different manufacturers will have different transmission values. It is recommended that the designer consult the latest manufacturers' catalogs because more accurate ratings are being issued from time to time.

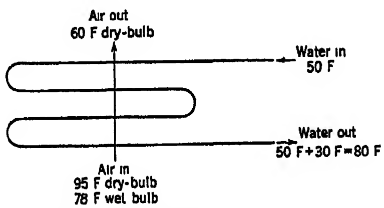


FIG 5. COUNTER-FLOW SURFACE COOLING DIAGRAM

Example 6 It is desired to cool and dehumidify 30,000 cfm of air at 95 F dry-bulb, 78 F wet-bulb, and 72 F dew point, to a 60 F dew point. Cooling water is available at 50 F in a quantity which will allow a 30 F rise in temperature to be used. The counter-flow surface cooling used is sketched in Fig. 5

Solution. The pounds of partially saturated air cooled and dehumidified per hour equal 60 times the cubic feet of air at 95 F dry-bulb and 78 F wet-bulb brought past the coil surface per minute, multiplied by the pounds per cubic foot of the air as determined from Table 4, Chapter 1.

$$30{,}000 \times 60 \times 0.0708 = 127{,}440 \text{ lb per hour.}$$

The total heat H_t to be removed per hour by the surface coil is found to be equal to the pounds of partially saturated air passed over the coil per hour times the difference between the total heat of air at 78 F wet-bulb and at 60 F wet-bulb.

$$H_t = 127{,}440\ (41.42 - 26.37) = 1{,}918{,}000 \text{ Btu per hour}$$

The latent heat L to be removed per hour will be found by multiplying the pounds of partially saturated air passed over the coils per hour by the difference in the latent heat of the air per pound at the initial and final dew points

$$L = 127{,}440\ (1091.6 - 1086.2) = 688{,}000 \text{ Btu per hour.}$$

The sensible heat H_s to be removed per hour is equal to the total heat of the air less its latent heat.

$$H_s = H_t - L = 1{,}918{,}000 - 688{,}000 = 1{,}230{,}000 \text{ Btu per hour.}$$

Manufacturers' standard ratings for surface coolers are usually based on the cubic feet of air passed through their equipment per minute, reduced to the conditions of saturated air measured at a temperature of 70 F. In the present example, to convert the 127,440 lb of air cooled per hour to a basis which will permit the use of such standard ratings, it is necessary to multiply the pounds of air cooled per hour by the specific volume of the air, and to divide by 60.

$$\frac{127{,}440 \times 13.68}{60} = 29{,}100 \text{ cfm of 70 F saturated air.}$$

The amount of cooling water necessary when a 30 F rise in its temperature is to be used is:

$$\frac{1{,}918{,}000}{30 \times 8.34 \times 60} = 128 \text{ gpm.}$$

With counter flow of air and water, it is necessary to determine the mean temperature difference between the air and the water in order to properly use the transmission coefficients given in apparatus rating tables.

$$\text{Mean temperature difference} = \frac{D_1 - D_2}{\log_e \dfrac{D_1}{D_2}} \tag{6}$$

where

D_1 = the difference between the temperatures of inlet air and outlet water, degrees Fahrenheit

D_2 = the difference between the temperatures of outlet air and inlet water, degrees Fahrenheit.

$$\frac{(95 - 80) - (60 - 50)}{\log_e \dfrac{(95 - 80)}{(60 - 50)}} = 12.33 \text{ F.}$$

If from apparatus rating tables based on air velocities over the coils and water velocities through the coils, it has been found that the transmission coefficient is equal to 8.0 Btu per square foot per degree difference in mean temperature between the air and the water, the area of cooling coil surface necessary will be equal to the sensible heat divided by the transmission coefficient and also by the mean temperature difference.

$$\frac{1{,}230{,}000}{8.0 \times 12.33} = 12{,}450 \text{ sq ft of cooling coil surface necessary.}$$

The latent heat is taken out at the same time the sensible heat is extracted, but no extra surface is required unless the latent heat exceeds approximately 40 per cent of the total heat. This is because the wetted surface has a much higher coefficient of transmission. Approximately 10 per cent more surface should be added if the latent heat exceeds 40 per cent of the total heat.

AUXILIARY EQUIPMENT

Consult Chapters 9, 14, 17, 19, 20, and 42 for information on the air distribution system; air outlets and inlets; static pressure on fan; fans; motor and drive; and the control system.

PROBLEMS IN PRACTICE

1 ● What is meant by the term evaporative cooling?

Evaporative cooling, or adiabatic saturation of the air, is only effective when the air to be cooled is very dry It is accomplished by passing the air in an unsaturated condition through a water spray which evaporates a part of the water at the expense of the sensible heat. In this adiabatic transfer the total heat content of the air remains constant while the dew point rises and the dry-bulb falls until the air is saturated

2 ● In central systems for cooling and dehumidifying what factors fix the quantity of air required?

The weight or volume of air required depends wholly on the sensible heat gain in the room conditioned and on the difference between the dry-bulb temperature of the air at the room inlets and the dry-bulb temperature maintained in the room

3 ● In central systems for cooling and dehumidifying can the dry-bulb temperature change be fixed arbitrarily?

No, because the change depends on factors at both the conditioner and the room At the conditioner, temperature of the available water supply may limit the dry-bulb temperature of the leaving air. At the room, the dry-bulb temperature of the entering air may be further limited by 1 The duct and supply grille arrangement permitted by architectural and structural requirements for the particular space, *e.g.*, ceiling height and obstructions on ceilings, such as beams 2 The state of activity of the occupants 3 The velocity at the inlet grille, as limited by noise level requirements 4 The direction of the jet relative to the occupants.

4 ● What factors determine the dew point of the air entering the space?

The maximum dew point desired in the conditioned space, and the moisture gain in the space per unit weight of air supplied.

5 ● Why must the air leaving a dehumidifying type air washer often have its dry-bulb temperature raised before delivery to the occupied zone of room?

The air leaves the dehumidifying air washer saturated at a relatively low temperature which in most cases is lower than the allowable delivery dry-bulb temperature as fixed by factors outlined under Question 3 Also, the air may possibly be carrying a small amount of entrained water which might settle out in the ducts near the washer and cause corrosion difficulties

6 ● What methods may be used to raise the dry-bulb temperature of the air after it leaves the dehumidifying air washer and before it enters the room?

a Sensible heat may be added by a reheating method from a source outside the air stream This method passes all or part of the cold, dehumidified air over steam or hot water coils at the central conditioner or in the ducts, or over electric grids or similar devices. Any available source of sensible heat can be used

b A mixing method using sensible heat already in the air stream. In this method the cold, dehumidified air is mixed with air at a higher dry-bulb temperature and the dry-bulb temperature of the resulting mixture is higher than that of the air when it left the conditioner The air at high dry-bulb temperature is obtained by not passing it through the dehumidifying washer The mixing may take place at a central conditioner or in the rooms themselves

c. Combinations of these methods.

7 ● What are the advantages of using counter flow of air and water in surface coolers?

Counter flow results in a higher mean temperature difference than does parallel flow for the same range of air and water temperatures, which means that less cooling surface is required. Counter flow permits higher initial water temperatures and also allows a greater temperature rise for the water These factors combine to reduce the cost of circulating and refrigerating the cooling water

Chapter 11

COOLING METHODS

Air Cooling Processes, Evaporative Cooling; Dehumidification, Refrigeration, Silica Gel System, Alumina System, Lithium Chloride System, System Design, Operating Methods, Steam Jet System, Refrigerating Compressors, Refrigerants, Evaporators and Coolers, Condensers

BY using any of the following four methods, or any combination of them, *effective temperature* (see Chapter 3) may be reduced.

a Sensible cooling· Lowering of the dry-bulb temperature by the removal of sensible heat without change of the dew-point temperature.

b. Dehumidifying: Lowering of the dew-point temperature by the removal of moisture without change of the dry-bulb temperature.

c. Evaporative cooling· Lowering of the dry-bulb temperature through the evaporation of moisture without the addition or the subtraction of heat.

d. Air motion· Increasing the air motion over the body.

As an example, let the condition be considered of 92 F dry-bulb, with a 40 per cent relative humidity, corresponding to a wet-bulb temperature of 72.8 F, and an effective temperature for still air of 81.1 F. This *effective temperature* may be reduced 3.1 F by any of the four basic methods mentioned, as follows:

First, by lowering the dry-bulb temperature to 85 5 F without changing the dew-point of 64.2, this gives an effective temperature of 78 F.

Second, by reducing the moisture content of the air to 46 grains per pound of dry air without changing the dry-bulb temperature; this gives an effective temperature of 78 F

Third, by reducing the dry-bulb temperature to 83.8 F without changing the total heat of the air This requires the evaporation of 14 grains of moisture per pound of dry air, and the effective temperature will become 78 F.

Fourth, by increasing the air movement from still air to 460 fpm, a velocity which will reduce the effective temperature 3 1 F from 81 1 F to 78 F

AIR COOLING PROCESSES

The best method of reducing the effective temperature in any specific case will depend on the accompanying circumstances and can be determined only by a thorough analysis made by a competent engineer. Generally speaking, the removal from the air of the sensible heat, or moisture, or both, by sensible cooling or dehumidifying is the most satisfactory method. Adequate results by the utilization of air motion or by evaporative cooling are difficult to obtain because of the dependence of both methods upon climatic conditions beyond the engineers' control although these methods are much less expensive than the first two

Table 1 Average Maximum Water Main Temperatures[a]

State	City	Temp F	State	City	Temp F
Ala	Birmingham	84	Mass	Boston	80
	Mobile	73		Cambridge	70
Ariz	Phoenix	81		Fall River	76
	Tucson	80		Lowell	50
Calif	Anaheim	60		Lynn	68
	Berkeley	69		New Bedford	70
	Fresno	72		Salem	68
	Fullerton	75		Worcester	76
	Glendale	68	Mich	Detroit	77
	Los Angeles	75		Flint	70
	Oakland	69		Grand Rapids	84
	Ontario	70		Highland Park	77
	Pasadena	82		Jackson	56
	Pomona	75		Kalamazoo	53
	Riverside	78		Lansing	64
	Sacramento	72		Saginaw	82
	San Bernardino	65	Minn	Duluth	55
	San Diego	82		Minneapolis	80
	San Francisco	62		St Paul	77
	Whittier	75	Mo	Jefferson City	82
Colo	Denver	75		Kansas City	84
Conn	Bridgeport	66		St Joseph	84
	Hartford	73		St. Louis	85
	New Haven	76		Springfield	70
	Waterbury	72	Nebr	Lincoln	87
D. C	Washington	84		Omaha	87
Del	Wilmington	83	Nev	Reno	61
Fla	Jacksonville	80	N H	Manchester	76
	Miami	80	N J	Jersey City	63
	Tampa	77		Newark	74
Ga.	Atlanta	87		Paterson	78
	Macon	80		Trenton	79
Ill	Chicago	76	N. Y	Albany	68
	Cicero	76		Buffalo	75
	Evanston	73		Jamaica	56
	Peoria	67		Mt Vernon	74
	Rockford	59		New Rochelle	75
	Springfield	82		New York	72
Ind	Evansville	86		Rochester	70
	Gary	75		Schenectady	60
	Indianapolis	80		Syracuse	71
	South Bend	61		Utica	69
	Terre Haute	82		Yonkers	70
Iowa	Cedar Rapids	78	N. C.	Asheville	74
	Des Moines	77		Charlotte	85
	Sioux City	62		Winston-Salem	82
Kans	Concordia	57	N M	Albuquerque	65
	Kansas City	86	Ohio	Akron	76
	Topeka	88		Canton	50
	Wichita	72		Cincinnati	84
Ky	Louisville	85		Cleveland	74
La	Baton Rouge	85		Columbus	82
	New Orleans	85		Dayton	60
Me	Augusta	60		Lakewood	82
				Springfield	72
Md	Baltimore	67		Toledo	83

[a]These averages taken from various city water main locations, with some actual values slightly higher and some lower than values shown

Table 1 Average Maximum Water Main Temperature[a] (Continued)

State	City	Temp F	State	City	Temp F
Okla	Oklahoma City	82	Utah	Logan	44
	Tulsa	85		Salt Lake City	60
Oreg	Eugene	60	Va	Fredericksburg	75
	Portland	64		Lynchburg	73
Pa	Altoona	74		Norfolk	80
	Erie	75	Wash	Olympia	58
	Johnstown	74		Seattle	62
	McKeesport	82		Spokane	51
	Philadelphia	83		Tacoma	57
	Pittsburgh	67	W Va	Charleston	85
R I	Providence	68		Huntington	78
S C	Charleston	80		Wheeling	78
	Greenville	81	Wis	LaCrosse	54
	Spartanburg	78		Madison	58
S Dak	Rapid City	55		Milwaukee	70
Tenn	Chattanooga	84		Racine	68
	Knoxville	89			
	Memphis	70			
	Nashville	90			
Texas	Amarillo	65	Province		
	Austin	90			
	Beaumont	86			
	Dallas	86	Alta	Calgary	64
	Fort Worth	84	B C	Vancouver	60
	Galveston	90	Ont	London	50
	Houston	84		Toronto	63
	Port Arthur	83	P E I	Charlottetown	48
	San Antonio	76	Que	Montreal	78
	Wichita Falls	85		Quebec	68

[a]These averages taken from various city water main locations, with some actual values slightly higher and some lower than values shown.

mentioned. Cooling by evaporation is satisfactory only when the air to be cooled is very dry; air motion as a means of producing cooling effect is never entirely adequate in the range of high temperatures. Of the two, evaporative cooling, or adiabatic saturation of the air, is a much more dependable method which will make more reduction in the effective temperature than will an increasing air motion within permissible limits.

As an example of this, consider an outdoor condition of 96 F dry-bulb and 80 F wet-bulb. The effective temperature is 85.7 F and, if the still air is moved with a velocity of 300 fpm, the effective temperature will be reduced only 2 0 F while saturation at the wet-bulb temperature would reduce the effective temperature 5.7 F. At 300 fpm velocity this saturated air would reduce the effective temperature to 75.6 F, thus making a total improvement of 10.1 F.

Frequently the temperature of the city water main supply is low enough during the summer to permit an appreciable cooling effect. Table 1 lists the maximum city water main temperatures for various localities in this country and Canada.

EVAPORATIVE COOLING

Evaporative cooling is accomplished by passing air through a water

spray in which the water is being continually recirculated. The air, entering in an unsaturated condition, evaporates a part of the water at the expense of the sensible heat. As this is an adiabatic transfer, the total heat content of the air remains constant, while the dew point rises and the dry-bulb falls until the air is saturated. A system[1] of ducts and a propelling fan are used to distribute the air in a proper manner.

It will be seen that the reduction in dry-bulb temperature is a direct function of the wet-bulb depression of the air entering the dehumidifier and that the resulting air temperature is governed entirely by the entering wet-bulb temperature of the outside air.

DEHUMIDIFICATION

Dehumidification may be accomplished in three ways:

1. By cooling the air below the dew point and causing a part of the moisture contained to precipitate.
2. By extracting all or part of the moisture by adsorption.
3. By extracting all or part of the moisture by absorption.

As used in this discussion, the term *adsorption* pertains to the action of a substance in condensing a gas or vapor and holding the condensate on its surface without any change in the chemical or physical structure of the substance and with the release of sensible heat. The term, *absorption*, implies a change in the chemical or physical structure of a substance in the process of dehydrating air. Adsorption is distinctly a surface action and is thereby distinct from absorption. Adsorbers include lithium chloride, calcium chloride; silica gel, lamisilite, or any of the halides; absorbers include sulphuric acid.

Dehumidification by Refrigeration

Air conditioning imposes requirements on refrigeration equipment not usually found in general cooling work, so that specially designed apparatus is often needed to replace that normally used for industrial cooling. Standard equipment can be adapted to meet air conditioning requirements but extreme care must be taken to determine the limits of its applicability.

In industrial or process cooling systems the load is fairly constant, noise in operation is not of paramount importance, space is available or relatively cheap, condenser water is not a source of worry, and the cooling system is to a great extent separate and independent of other mechanical equipment. By contrast, air conditioning, especially as used for space cooling and comfort work in office buildings, theaters, and places where people gather requires special consideration of all these factors. Space in public buildings is limited and condenser water is expensive. Noise interferes with the occupants, and the cooling equipment must dovetail with the other air-handling apparatus. Most important, the load fluctuates tremendously and is seasonal.

A complete discussion of the thermodynamic problems of refrigeration is given in Chapter 2.

---

[1]See Air Washer Performance in Chapter 12, also Theory of Atmospheric Cooling in same Chapter.

Silica Gel System

Silica gel is a chemical composition made from sodium silicate and acid, the chemical formula being SiO_2. It has an appearance greatly resembling that of clear quartz sand but it differs in structure in that the crystals are highly porous, with voids constituting 41 per cent by volume although the pores are microscopic in size. This material possesses the property of being able to adsorb a substantial portion (about 25 per cent of its own weight) of moisture from the air without any increase in its volume. After the silica gel has become "saturated" or has adsorbed moisture to the limit of its capacity, the moisture may be driven from it by the application of heat, again without change in the structure, volume, or chemical composition of the silica gel. This cycle may be repeated indefinitely. When applied to air conditioning the silica gel which is exposed to the air reduces the moisture content in the air and releases sensible heat which may be readily removed from the air A typical diagram of this system is shown in Fig. 10, of Chapter 2 on Refrigeration.

Practical Application of Silica Gel

Silica gel has two applications when used to replace refrigeration. In the one principally used, the air from which moisture is to be extracted is taken through silica gel beds by suction or pressure fans, and by means of this process the moisture becomes adsorbed by the silica gel and the air leaves at a lower dew point and a higher sensible temperature than those at which it entered. If this air is passed over surface coolers in which tap water or another cooling medium is flowing through tubes, a certain amount of sensible heat will be removed. The air leaves the surface cooler or interchanger with the same dew point with which it emerged from the silica gel beds, but with a lower dry-bulb temperature, although the dry-bulb temperature may be higher than the temperature of the air entering the silica gel beds.

In another method, the first two of the steps outlined are duplicated, and in addition the air is carried through a spray type washer. Because the air enters the washer with a low wet-bulb, and because adiabatic saturation will take place at a temperature close to the entering wet-bulb, considerable cooling of the air can be accomplished; but this can be done only with a consequent increase of the dew point.

It is necessary to reactivate the silica gel after it has adsorbed about 25 per cent of its own weight in the form of moisture. As reactivation requires a high temperature and since silica gel is only active at low temperatures, cooling of the beds must also be completed before they can be used again. This necessitates three stages in the silica gel containers and requires either three beds of silica gel or one bed divided and automatically put in position. The reactivation is usually done by means of gas or oil fires and the cooling of the beds by means of indirect water cooling or by means of small quantities of dehydrated air taken from the system beyond the interchanger.

Alumina System of Adsorption

Activated alumina contains a trifle over 91 per cent of aluminum oxide, Al_2O_2, which material will adsorb nearly 100 per cent of the vapor in the

air up to about 8 or 10 per cent of the weight of the adsorbing material, after which the adsorption falls off gradually as the saturation point is approached. The application is quite similar to that employed for silica gel; that is, the material is exposed to the air flow and after reaching about 75 per cent saturation is reactivated by removing the moisture adsorbed by means of applied heat. The actual scheme generally followed in the use of this material for continuous service varies somewhat from silica gel inasmuch as the material is placed in three units which are used consecutively for the different steps. These steps permit each unit to operate as follows:

a. In series with the preceding unit
b Alone
c. In series with the following unit

This plan allows for adsorption, reactivation, and cooling, in a manner similar to that used with silica gel.

Taking a single unit, when it is in the *a* step and operating with the preceding unit, the alumina adsorbs approximately 25 per cent of the moisture in the air and takes up about 1.3 per cent of its weight of water. During the second step when it is operating alone, it takes up 100 per cent of the moisture in the air until the weight of the water adsorbed is brought up to about 6.7 per cent. During the third step when the unit is operating with the succeeding unit, it extracts about 75 per cent of the moisture in the air until the water weight adsorbed comes up to about 10 per cent of the weight of the adsorber. The time allowable for reactivating is equal to the time occupied by the second unit adsorbing alone, plus the time when the second and third units are adsorbing in series, plus the time when the third unit is adsorbing alone, at the expiration of which time the first unit will be again required.

The temperature of air used for alumina reactivation is usually between 300 and 700 F and the air flow rate will have to be higher with the low temperature air than it will be with reactivating air of higher temperature. For example, air at 400 F for reactivating will, at 10 cu ft per hour per pound of alumina, require about 6 hours for reactivation. In the three unit system, after reactivation the cooling of the activated alumina may be carried out with considerable rapidity by using dry air from the adsorption unit for circulation through the unit which has just completed reactivation. The final temperature of the unit before it goes back into service should be not over 200 F. As a basis for computing the amount of cooling air required for reactivation, each cubic foot of cooling air has been found capable of removing 2.2 Btu when heated from 85 to 200 F and of providing a sufficient margin of safety in operation.

Lithium Chloride Adsorption System

Practically all salt solutions have the property of adsorbing or condensing moisture from a gas. The amount depends on the character of the solution, its gravity, temperature and viscosity. One of the best known dehydraters, calcium chloride, for instance, has been long known for this property. Its characteristics, however, present a limiting factor of about 30 per cent relative humidity, so that the resultant dew point of air brought in intimate contact with calcium chloride is usually too high for comfort work without further refrigeration.

The cycle of solid dehydraters or adsorbers and liquid adsorbers are fundamentally the same. A diagram of a lithium chloride adsorption system is shown in Fig. 1.

The liquid adsorber is brought in contact with air having a certain vapor pressure due to the moisture of water vapor that is in it, and the chloride, either sodium, lithium, calcium, or whatever is used, having a lower vapor pressure, adsorbs moisture in the form of water from the water vapor that is in the contacting air. Thermodynamically, it is definitely possible to measure this change and to know definitely that a change of state takes place because of the fact that there is a definite rise in temperature in the liquid adsorbing the moisture, which is definitely a function of the amount of water vapor condensed from the air stream.

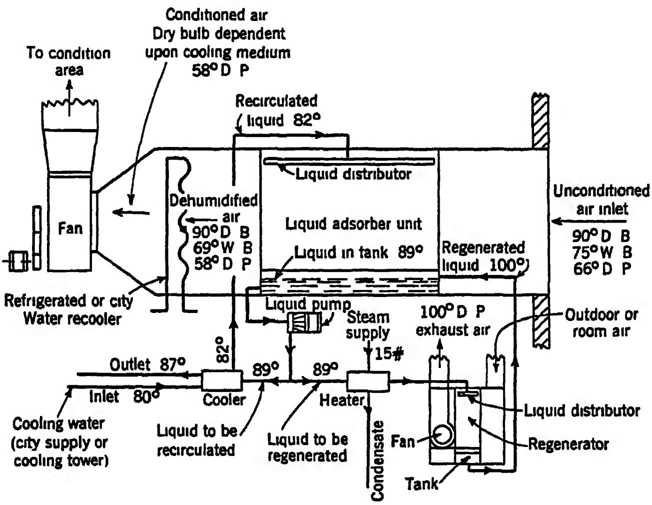


FIG. 1. DIAGRAM OF LITHIUM CHLORIDE ADSORPTION SYSTEM

Of course, the liquid adsorber, contrary to the solid adsorber such as silica gel, takes on two functions: one, as an adsorber, and that adsorption, as has been stated, tends to create a rise in temperature; on the other hand, the mass of liquid in the adsorber tends to keep down the rise in temperature, so that by cooling the adsorber below a certain temperature, a definite temperature can be maintained in both the liquid and the air, which is a function of the heat balance between the total heat of the air and the total heat of the liquid.

Adsorption of moisture by a liquid adsorber weakens the concentration of the liquid so that its adsorbing capacity is reduced and regeneration, or the driving off of the excess moisture in the liquid, must be performed, just as the driving off of the liquid adsorbed by a solid adsorber must be accomplished. However, with a liquid adsorber a very definite condition can be maintained; that is, a condition of density, by continuously withdrawing a certain small portion of the total liquid for intensive concentration and the concentrated liquid added to the mass of the liquid continuously, without varying to any marked degree the vapor pressure

of the total mass. This method of continuous regeneration and re-concentration tends to hold the relative humidity of the leaving air always at a very definite point.

There are two methods of regeneration. One is the boiling of the excess moisture by raising the temperature of the adsorbing liquid to above the boiling point of the particular concentration. As the salt in the solution does not vaporize, it is not carried off in the boiling process. The only care required is that in the small amounts of liquid diverted to the regenerator for concentration, too much moisture is not driven off, occasioning freezing or solidification of the salts, or solids.

The second method of regeneration which lends itself particularly well to ordinary low pressure heating systems, where steam pressure of about 12 lb per sq in., corresponding to a temperature of 242 F, is to raise the temperature of solution with ordinary steam coil interchangers to about 225 F, and then to pass the solution at this temperature over various types of scrubbers, over which ordinary air is passed. The increase in temperature of the liquid adsorber tends to increase its vapor pressure to such an extent that there is an exchange of vapor between the liquid adsorber and the air, as well as an equalization of temperature between the air and the liquid adsorber, so that the air is capable of taking up part of the moisture from the liquid adsorber to increase its density and to carry this excess moisture out into the atmosphere with the leaving air.

After this vaporation has taken place, the highly concentrated, hot liquid adsorber is taken through an interchanger through which the water used in cooling the main solution can be re-used to reduce the temperature of the concentrated solution to a point where it may be thrown back into the main solution tank at a slightly higher temperature than the main body of the solution. As approximately only 10 per cent of the solution is regenerated continuously, a difference of 5 to 10 F in the concentrating solution only creates a rise of 1 F or less in the main body of the adsorber, so that its effect on increasing the temperature of the air to be dried is negligible.

As can be seen, there are two conditions of continuous operation which have a tendency to raise the temperature of the liquid. One is the adsorption of vapor from the air, which, through a change of state, changes the latent heat of the vapor adsorbed to sensible heat, to raise the temperature of the liquid and consequently, the temperature of the air; secondly, the necessary heat added to the regenerator liquid in order to re-evaporate and carry off the excess moisture which has been condensed in the first stage.

In the regeneration process, the air currents in this method have a tendency to carry away the excess heat of evaporation. In the first stage, that is, the main stage of adsorption, the temperature of the main body of liquid is continuously cooled, and the excess heat carried away by some type of interchanger, through one part of which either cooling tower or city water is utilized. Of course, this water may be refrigerated, but under most conditions that is unessential. With the ordinary liquid interchanger, using cooling tower water, the maintained condition of the liquid and the air can be kept at a point very close to the prevailing wet-bulb, possibly within 5 F, and certainly within 10 F.

DESIGN OF SYSTEM

When designing air conditioning systems, the capacity of equipment is decided by selecting apparatus of sufficient size to maintain predetermined temperatures and humidities in treated spaces when arbitrarily established maximum atmospheric temperatures occur coincident with given conditions of population, lighting, and power consumption. These factors determine the maximum duty of the cooling system. The duty does not necessarily determine the size or capacity of the refrigeration apparatus. The refrigerating capacity is expressed in *tons*, each ton being equal to the absorption of the heat given up by one ton of ice at 32 F melting to water at 32 F in 24 hours. This is equivalent to heat absorption at a rate of approximately 200 Btu per minute, or 12,000 Btu per hour.

After the maximum duty is determined, the other factors concerning the installation must be investigated. The total heat to be removed by the cooling system has many sources, some substantially constant and others extremely variable. These sources can be roughly classified as follows, the first column indicating the order in amount and the second the order in variability:

1. Fresh air supplied.	1. Fresh air supplied.
2. Population.	2 Transmission through the structure
3. Transmission through the structure	3. Light and power consumed
4. Light and power consumed	4. Population

By combining these two columns, a third grouping is obtained as follows:

1 Fresh air supplied.
2 Transmission through the structure.
3 Population
4 Light and power consumed

In this last arrangement, the first two items are governed by atmospheric conditions and they are therefore subject to tremendous fluctuations in value. As they generally form 40 to 60 per cent of the entire maximum load, the duty of the cooling system will be much less than maximum most of the time.

The transmission through the structure is especially influenced by the sun. (See Chapter 8.) In many cases, because of the heat flow resistance of the structure, the heat from the sun is retarded until it is compensated for by a reduced general temperature out-of-doors.

A survey of Weather Bureau records indicates that maximum temperatures occur less than 5 per cent of the cooling period and also that the duration of peak conditions is never more than three or four hours.

Two factors control the size of the refrigeration system, the evaporator or suction temperature, and the condenser or head temperature. With the knowledge that the system will operate most of the time with a load of not over 60 per cent of maximum, and that maximum demands will occur infrequently and only for short periods, some provision must be made to insure economical operation under average conditions. This can be done by overloading the machine under extreme demands and basing the design on normal or average loads. Flexibility in arrangement can be provided in several ways.

Variations in load change the efficiency of any machine and a refrigerating system can be costly and inefficient if improperly designed or operated.

Fortunately, the trouble can be concentrated in the compressor and the problem relieved of many complications. It is comparatively easy to furnish condensers and evaporators to carry the maximum load so arranged that they will function properly at small demands. They affect the compressor performance to some extent but most of the compressor problems are in the machine itself.

Variations in load are usually effected by lowering the suction temperature and pumping a larger volume of gas per ton through a greater pressure range. This is possible because the latent heat of the refrigerant remains nearly constant throughout the small range used and the specific volume varies rapidly with change in pressure. As the compressor must remove the refrigerant evaporated, the evaporator temperature fixes the displacement required. The objection to such method is that the total power consumed remains nearly constant and the power per unit of cooling increases rapidly as the total output is reduced. Such operation is satisfactory as long as the load is kept within 10 per cent of the rating of the compressor but this condition does not commonly occur in air conditioning applications.

Operating Methods

It is possible to divide the entire refrigeration system into a number of small units, which will allow cutting in and out of compressors and condensers as the load fluctuates.

A second method of providing for economy of operation is to have storage capacity which can be utilized during the peak period. A further reference to the Weather Bureau records indicates that maximum conditions prevail during the day for not more than three hours, and consequently the refrigerating system can be run for a longer period at maximum efficiency with tanks to store cold water or brine for supplementing the actual output of the refrigerating equipment when the load is more than the machine will carry. This situation brings complications. Storage tanks require space and extra apparatus, which increase the cost of the entire system, and further, it is difficult to determine what the size of the compressor should be because of the other variables which enter the problem. Depending upon the availability of storage space, the compressor could be designed for any reasonable percentage of the maximum load, so the smaller the compressor, the larger the storage space, and vice versa.

A third method is to provide in the compressor itself some means of reducing the capacity. This can be done by varying the speed and consequently the displacement of the compressor, or by varying the displacement, either by a partial by-pass of the cylinder or by a clearance pocket in the head of the cylinder when reciprocating compressors are used. It might be assumed that the efficiency would remain practically constant. This is not correct, inasmuch as the machine friction remains constant with the by-pass or clearance pocket method and this raises the power required per ton of refrigeration developed. Also, the volumetric efficiency of the machine falls off rather rapidly when the clearance pocket or partial by-pass is used. By varying the speed of the compressor, the efficiency of the power unit falls off as the speed is reduced, while the compressor friction remains constant.

Another method of operation is the automatic starting and stopping of the refrigerating machine, with the automatic control designed to function as the load varies. All of the methods described are used from time to time.

The methods of varying the output of a refrigeration system which have been outlined apply to the reciprocating type of compressor, although variations in the speed of the compressor to change the refrigerating output are common to all types of mechanical refrigeration.

There is a further method of controlling the compressor output which is particularly adaptable to the centrifugal type of machine. This is accomplished by varying the amount of condensing water used with the fluctuation in demand load. Because of the characteristics of the centrifugal type of apparatus, as the condensing water quantity is reduced and the condensing temperature consequently raised, the discharge pressure of the centrifugal machine rises correspondingly and the horsepower input to the machine falls off. While this reduces the total power input to the machine, it does not necessarily reduce the power input per ton of refrigeration developed, as the power input does not drop with a rising discharge pressure as fast as the refrigerating effect produced drops. It is a method, however, which shows marked economies over the method generally used by the operating engineer, which is to lower the suction pressure in order to reduce the refrigerating output of the system.

STEAM JET SYSTEM

So far the discussion has been confined to reciprocating, centrifugal, and rotary compressors. The steam jet type of compressor, under certain circumstances, is desirable for use in air conditioning. A complete flow diagram of the system is shown in Fig. 4 of Chapter 2 on Refrigeration The power used for compressing the refrigerant is steam, taken directly from the boiler, thus eliminating the mechanical losses of manufacturing electric current. As the compression ratio between the evaporator and condenser under normal circumstances is large, the mechanical efficiencies of the equipment are somewhat lower than those of the positive mechanical type of compressor; also the condensing water requirements are considerably greater, as both the refrigerant and the impelling steam must be condensed.

The steam jet system functions on the principle that water under high vacuum will vaporize at low temperatures, and steam ejectors of the type commonly used in power plants for various processes will produce the necessary low absolute pressure to cause evaporation of the water.

In Chapter 2 on Refrigeration Fig 4 shows a typical water cooling application. The water to be cooled enters the evaporator and is cooled to a temperature corresponding to the vacuum maintained. Because of the high vacuum, a small amount of the water introduced in the evaporator is flashed into steam, and as this requires heat and the only source of heat is the rest of the water in the evaporator tank, this other water is almost instantly cooled to a temperature corresponding to the boiling point, determined by the vacuum maintained. The amount of water flashed into steam is a small percentage of the total water circulated through the evaporator, amounting to approximately 11 lb per hour per ton of refrigeration developed. The remainder of the water at the desired

low temperature is pumped out of the evaporator and used at the point where it is required.

The ejector compresses the vapor which has been flashed into the evaporator, plus any entrained air taken out of the water circulated, to a somewhat higher absolute pressure, and the vapor and air mix with the impelling steam on the discharge side of the jet. The total mixture of entrained air, evaporated water, and impelling steam is discharged into a surface condenser at a pressure which permits the available condensing medium to condense it. The resulting condensate is removed from the condenser by a small pump, from which it can be discharged to the sewer or returned to the system in the form of make-up water, or part of it may be returned to the boiler feed pump.

As the normal temperature of water required for air conditioning purposes is between 40 F and 50 F, with an average temperature of approximately 45 F, this type of water cooling is particularly desirable, as the efficiencies and operating costs compare very favorably with other types of refrigerating equipment, especially in view of the fact that the cooling apparatus is, as a general rule, less expensive to install.

Approximately three times as much condenser water is required for the steam jet cooling system as would be necessary with other types of mechanical refrigeration, but as the system can be designed with a large number of jets, each of which can be cut off as the load falls below maximum, constant refrigerating efficiency is maintained and frictional losses and volumetric inefficiencies are kept at a minimum.

The slight amount of air which may be entrained in the cooled water is removed by a small secondary ejector which raises the pressure sufficiently so that the air can be discharged to the atmosphere. A small secondary condenser, of course, is necessary to condense the steam used in the secondary jet.

Steam jet refrigeration has an advantage where cooling towers are used for supplying the condensing liquid, as there is a great saving in the amount of steam used per ton of refrigeration. As the outdoor weather conditions vary the load on the cooling system, the compression ratio between the condenser and evaporator can be reduced and less propelling steam need be used per ton of refrigeration developed. Roughly, in air conditioning work, mechanical compressors show a falling off of 30 to 40 per cent in the power input when using the most economical arrangement of compressors, as the load varies from 100 per cent to 25 per cent of the rated capacity; whereas with steam jet cooling equipment, the amount of steam required for producing the necessary refrigerating effect falls off in direct proportion to the load on the system. When steam refrigeration is employed with cooling towers, the efficiency increases as the output is reduced.

REFRIGERATING COMPRESSORS

There are many different types of compressors, a number of refrigerants, different types of evaporators, condensers and arrangements of the cycle, and each type has its particular place and usage.

The generally used compressors are of the following types:

1. Reciprocating compressors
2. Centrifugal compressors.
3. Rotary compressors
4. Steam jet compressors.

Over-all efficiency of the compressor in smaller commercial installations is not as important a requirement, as that the whole unit require little attention and make a minimum of noise. The noise level when the fan, sprays, and compressor are in full operation should not exceed 25 decibels. High compressor efficiency appears as an important factor only in the larger industrial air conditioning systems.

The refrigerants in most general use in commercial and industrial air conditioning are here listed in the order of their inoffensive odor characteristics:

1. Water vapor.
2. Carbon dioxide
3. Dichlorodifluoromethane and trichloromonofluoromethane
4. Dichloromethane, sometimes called methylene chloride
5. Methyl chloride
6. Ammonia
7. Sulphur dioxide

The various types of compressors bear varied relationships to the refrigerants used in both commercial and industrial air conditioning. *Reciprocating compressors* are generally used for any of the refrigerants listed except water vapor, dichloromethane, or other low pressure refrigerant, and they are used in both commercial and domestic air conditioning systems. They have been developed to a point where their efficiency is high and their operation very satisfactory. Relatively low speed operation makes them desirable for general use in large installations. They are of two types, vertical and horizontal, either single or double acting. The horizontal double-acting compressor is not generally used in air conditioning except when carbon dioxide is used as the refrigerant in the larger industrial systems. Vertical, single-acting, encased crank, reciprocating compressors of the uniflow type with valves in the pistons have proven reliable and are used in capacities from 1 hp to more than 100 hp. Reciprocating compressors can be used with more refrigerants than other types of compression units. For instance, when carbon dioxide is used as the refrigerant, a reciprocating compressor is required because of the extremely high pressures and the relatively high ratio of compression.

The production of refrigeration at temperature levels from 25 F to 55 F for general air conditioning involves special types of refrigerating compressors. Among these are:

1. Centrifugal compressors using a volatile refrigerant
2. Centrifugal compressors using water as a refrigerant
3. Steam jet or vacuum systems using water as a refrigerant
4. Reciprocating compressors using a volatile refrigerant.

The first two types, *centrifugal compressors*, using trichloromonofluoromethane, dichloromethane or water vapor, can theoretically be used with any of the other refrigerants, but the resulting loss in efficiency with the higher pressure gases limits the centrifugal compressor to the two refrigerants named. At the present time the centrifugal compressors are limited to air conditioning systems of a minimum of 50 tons and more. Centrifugal compressors are usually built in two or more stages where the

compression ratio is high, and their design follows closely that of any other centrifugal equipment, such as general service pumps and fans

Steam jet compressors which have recently entered the field are simple and compact and, having no moving parts, they produce practically no vibration but are not economical for water temperatures much below 40 F or where the cost of generating steam is higher than the cost of operation with other prime movers.

Reciprocating compressors are generally used for methyl chloride and dichlorodifluoromethane because of their relatively low pressure and compression ratios. These compressors find widest use for fractional tonnage duty.

Refrigerants

The source of condensing water to some extent governs the type of refrigerant used. If condensing water is available at temperatures of not more than 70 to 75 F any of the refrigerants mentioned can be used economically, but if the available condensing water temperature is above 80 F, carbon dioxide becomes uneconomical as its critical temperature is approximately 88 F. A condensing water temperature over 80 F makes the power required for compression high. All refrigerants, except carbon dioxide, have critical temperatures and pressures sufficiently high so that their efficiency is not materially affected by the condensing water temperatures, except insofar as this temperature affects the compression ratio. Steam jet cooling systems can use water up to 85 F, or even slightly warmer

The applicability of the various refrigerants is interesting. Carbon dioxide is limited by the condensing water temperature; the power consumption is slightly higher than that of other refrigerants; and the pressures are three to four times that of ammonia.

The condenser pressures of methyl chloride and dichlorodifluromethane are approximately one-half that of ammonia.

Ammonia, probably the best known refrigerant, has the disadvantage of being toxic, and under certain circumstances explosive, corrosive, and irritating, even in small quantities in the atmosphere. An indirect closed system vented method or double indirect open spray vented method of refrigeration is used for cooling water or brine in air conditioning systems employing ammonia as the refrigerant.

Sulphur dioxide is corrosive and irritating even in small quantities in the atmosphere and it is toxic under certain circumstances.

Dichloromethane operates at pressures below that of the atmosphere, and it is to some extent toxic.

Dichlorodifluromethane and trichloromonofluoromethane under normal circumstances are odorless, non-irritating, non-toxic, non-flammable, non-explosive, non-corrosive and will do no damage to any materials with which they may come in contact.

Methyl chloride, under certain conditions, is explosive and slightly toxic.

The steam ejector water vapor system has none of the disadvantages of toxicity, explosiveness and corrosiveness encountered in the other refri-

gerants, but the system operates at less than atmospheric pressure. This, however, is not an important factor as there are no moving parts in the compressor and the possibility of inleakage of air is remote as all of the equipment can be welded air and water tight. The supply of water is inexhaustible, and as a refrigerant, the make-up cost is negligible. The same boiler equipment can be used for heating in winter and for cooling in summer.

Electric Motors

The motors used for driving compressors can be roughly classified in three groups· synchronous, multispeed, or variable speed. Further information on motors and their controls may be found in Chapter 42.

CONDENSERS

Condensers are usually either the double pipe type or the shell and tube type. Shell and tube condensers are almost identical with coolers. Double pipe condensers are arranged so that water passes through the inner of two concentric pipes, and refrigeration passes through the annular space in the outer pipe. Where possible, there should be counter flow of the refrigerant and the condensing water to maintain maximum temperature differences.

The amount and temperature of the condensing water determine the condensing temperature and pressure, and indirectly the power required for compression. It is, therefore, necessary to strike a balance so that the quantity of water insures economical compressor operation.

As part of the condenser, or attached to it, there must be storage space for liquid refrigerant. The installation of all equipment should be made accessible for inspection, repair, and cleaning. Both the coolers and condensers should have space for pulling tubes.

Because there is a decided tendency to conserve the water in city mains and most large cities are restricting the use of water, in order to use air conditioning systems and refrigeration equipment it is often necessary to install cooling towers. The cooling towers, unfortunately, produce the warmest condensing water at the time when the load on the system is greatest, so that the refrigeration equipment must be designed to meet not only the maximum load at normal conditions, but also the maximum load at abnormal condensing water temperatures. If properly designed, this makes little difference in the efficiency of operation throughout the year except at those times when the condensing water temperature is highest. As this occurs only for 5 per cent of the entire cooling period it can be disregarded as a factor in establishing yearly operating costs.

The cooling tower has a certain advantage over the use of water from the city mains in that the temperature of the condensing water varies directly with the outdoor temperature and, as pointed out, the refrigeration load also varies with this temperature. Certain economies are possible when a cooling tower is used which cannot be achieved by the use of condensing water from city mains, even where the city water temperature is extremely low. Normally, the lowest city water temperature met during the summer months is from 65 to 70 F. This temperature range takes

place for the entire cooling period, regardless of what the outdoor temperatures are. With the cooling tower, the temperature of the condensing water may rise to 80 or 85 F under maximum conditions, but under less than maximum conditions the temperature of the water off the cooling tower drops considerably, and it has been established that 50 per cent of the time the outdoor wet-bulb temperature varies from 60 to 70 F and the cooling tower water, therefore, for the same periods, varies from 65 to 75 F. When the outdoor wet-bulb temperature drops below 60 F, which occurs approximately 30 per cent of the time, the condensing water temperature is still lower. The cost of water used for condensing is negligible, as the only water required is that used to make up the loss by evaporation in the cooling tower itself. See also Chapter 12.

Evaporative Condensers

Due to the high cost of city water for condenser purposes and due to ordinances in some localities prohibiting the discharge of large quantities of such water into the sewage systems, there has been developed a con-

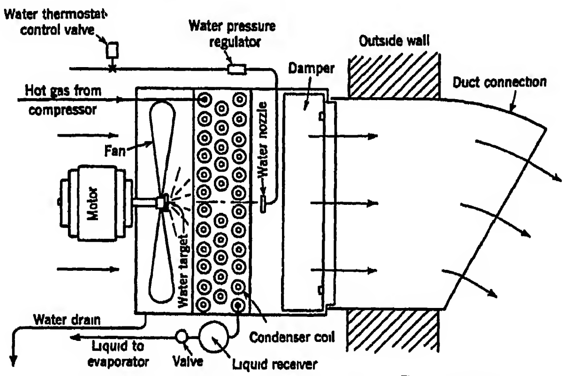


Fig 2 Diagram of Evaporative Condenser

denser which uses a minimum amount of water on a finned surface, cooling it to approximately the wet-bulb temperature of the surrounding atmosphere.

A diagram of a typical evaporative condenser is shown in Fig. 2, which includes a propeller type fan that forces the air over a finned tube condenser coil to the outside atmosphere through a connecting duct. A fine jet of water from a nozzle located at the center of the coil is directed to a target on the hub of the fan, where it is atomized and the spray is carried across the coil, wetting the surface. The hot refrigerant gases enter the top of the condenser coil and the liquid collects in the receiver below the condenser unit. A thermostat control valve and pressure regulator are arranged on the condenser water supply for regulating and adjusting the water flow to the nozzle. It is particularly desirable that a condenser of this type be located as near the compressor as possible.

Units of this design are available in sizes ranging from 2½ tons of refrigeration up to about 40 tons. For larger capacities several multiple units may be used in parallel. About 10 per cent as much water is required for this unit as is normally used in a shell and tube type condenser.

EVAPORATORS AND COOLERS

The types of coolers used in connection with air conditioning work fall into three general groups. The *first*, is the direct cooling of water; the *second*, direct cooling of air; and the *third*, cooling of brine for circulation in a closed system, which can cool either water or air. One method of the direct cooling of water is to install direct expansion coils in the spray chamber so that the water sprayed into the air comes in direct contact with the cooling coils. Another common and efficient method of cooling spray water is to use a Baudelot type of heat absorber where the water flows over direct expansion coils at a rate sufficiently high to give efficient heat transfer from water to refrigerant.

Another type of spray water cooler is the shell and tube heat exchanger in which the refrigerant is expanded into a shell enclosing the tubes through which the water flows. The velocity of the water in the tubes affects the rate of heat transfer, and as the refrigerant is in the shell completely surrounding the tubes at all times, good contact and a high rate of heat transfer are insured. The disadvantage of such a system is that with the falling off of load on the compressor the suction temperature or the temperature in the evaporator drops and there is a possibility of freezing the water in the tubes, which, of course, might split the tubes and allow the refrigerant to escape into the water passage. This danger can be eliminated by automatic safety devices.

Another system of cooling spray water is to submerge coils in the spray collecting tank, or in a separate tank used for storage. The heat transmission through the walls of the coils, however, is low and a great deal more surface is required than for any other type of cooler. However, with large storage tanks this type of cooling can be utilized to advantage.

When direct cooling of air is employed, the refrigerant is inside the coil and the air passes over it. Cooling depends upon convection and conduction for removing the heat from the air. The type of coil used can be either smooth or finned, the finned coil being more economical in space requirement than the smooth coil. The fins, however, must be far enough apart so as not to retain the moisture which condenses out of the air.

The indirect cooler, where brine is cooled by the refrigerant and the resulting cold brine is used to cool either air or water, introduces several other considerations. It is not the most economical from a power consumption standpoint, as it is necessary to cool the brine to a temperature sufficiently low so that there is an appreciable difference between the average brine temperature and that of the substance being cooled. This requires that the temperature of the refrigerant must be still lower, and consequently the amount of power required to produce a given amount of refrigeration increases due to the higher compression ratio, but there are other considerations which make such a system desirable. In the first place, where a toxic refrigerant is undesirable or cannot be used, due to fire or other risks especially in densely populated areas, the brine can be cooled in an isolated room or building and then be circulated through the air conditioning equipment in perfect safety because it is used to cool the water or air, without any possibility of direct contact between the air and refrigerant.

When an indirect system of cooling is used, it will be found that the heat

transfer rate of the water cooler is considerably higher as a general rule than that of a direct expansion cooler for the same requirements. With direct expansion interchangers, it is almost impossible to keep the entire system flooded with liquid, whereas with brine interchangers the cooling medium completely fills the space of the interchanger and perfect contact is insured

Ice may be used for chilling water or air for conditioning work. Its application is limited because of the cost of ice, and the difficulty of handling it. The word 'water cooling" is used advisedly in that the direct cooling of air by ice is, while not impossible, rather impractical. It might be said that ice coolers are economical for systems requiring a maximum of 20 tons per 24 hours where the load fluctuates considerably, and where ice is introduced only as it is required to cool water. The most general method of cooling water with ice is to spray the water over the surface of the ice, insuring as much contact as possible and approximating the same performance as the Baudelot type of cooler. Because of the large fluctuations in load in the air conditioning system, the higher cost of refrigerating effect when ice is used is offset by the fact that there are no motor and condenser inefficiencies under partial load Also, because the cost of the mechanical refrigeration equipment for the small system is so much higher per unit of effect, the fixed charges are small enough to overbalance the extra cost of the ice.

PROBLEMS IN PRACTICE

1 ● If, in a given air conditioning installation, fixed charges are estimated at 15 per cent per annum, what increased first cost would be justified for an auxiliary appliance on a refrigeration machine which could save $125.00 per year in operating cost?

\$125 00 capitalized at 15 per cent is, $\frac{\$125\ 00}{0.15}$ = \$834.00 increased first cost justified

2 ● Electrically driven Freon condensing units are to be used in an air conditioning system, requiring 20 tons refrigerating capacity for conditions of maximum load. An overall analysis of the seasons operating conditions shows an average load factor of 62.5 per cent, and allowing for variable time intervals of operation of refrigeration units installed, three-quarters of the operating season, or 750 hours, would require operation of the equipment at one-half load, and one-quarter of the operating season or 250 hours full load capacity of the refrigeration equipment would be required.

The increased first cost of 2–10 hp, 10 ton condensing units over 1–20 hp, 20 ton condensing unit is, $830.00 installed price, to the customer.

The increased first cost of a 2-speed compressor motor of 20 hp size over a constant speed 20 hp size motor including increased starter cost is $210.00. The efficiency of the 2-speed motor above is 83 per cent at full load full speed, and 79 per cent for full load at ½ speed. At ½ speed, full load is ½ total bhp of full load full speed.

Discuss the considerations involved in making a decision as to whether a single unit with a 20 hp motor of the 2 speed type would be used in preference to 2–10 hp constant speed units.

The cost of 2–10 hp 10 ton units in excess of 1–20 hp, 20 ton unit with 2-speed motor, is \$830 00 − \$210 00 or \$620.00, increased first cost. At 15 per cent fixed charges, this represents an increased annual cost of \$93 00 for 2 compressors over one compressor.

The advantage of 2 compressors instead of one compressor on an installation of this type, is in the breakdown service provided in the event one compressor is shut down for repairs the system could be operated at one-half capacity utilizing the duplicate machine The motor efficiency of the constant speed unit would be higher at full load than would be the efficiency of the 2-speed motor at low speed Offsetting this latter advantage however, is the fact that the condenser on the condensing unit would provide a lower refrigerant condensing temperature for ½ load operation with the same final condensing water temperature than would be the case with duplicate units each furnished with its own compressor and condenser Operation at a lower condensing temperature would provide for a power saving compensating for the lower efficiency of the 2-speed motor when operated at slow speeds It is, in a case of this kind, purely a question as to whether or not the purchaser would deem an investment of $620 00 more and an increased fixed charge of $93 00 a year, advisable to get breakdown service thru the installation of duplicate units In most cases, this increased first cost would not be warranted because of the fact that satisfactory indoor conditions could not be obtained at full load if only one-half the refrigeration capacity were available.

3 ● For an increased first cost of $210.00 for a 20 hp, 2-speed motor over a 20 hp constant speed motor to be used on a Freon automatic condensing unit for air conditioning duty, justify the increased investment based on a load factor of 62.5 per cent, on an operating basis of 1000 hours total of which ¾ of the operation of the refrigeration equipment would be at ½ load and ¼ of the operation season at full load. Motor efficiency 2 speed, 83 per cent, at full speed; and 79 per cent at half speed. Motor efficiency constant speed motor 83 per cent. City water is to be used for condensing purposes and is to be automatically controlled from the discharge pressure by means of automatic regulating valve. Compute increased first cost justifiable based on 15 per cent fixed charges and number of years required to pay off increased first cost.

At full load for 20 bhp load and 83 per cent efficiency, the kw input to the compressor motor would be 20 bhp times 0.746 kw per bhp divided by 83 per cent motor efficiency or 18.0 kw.

At ½ speed,

$$\frac{10 \text{ bhp} \times 0\ 746 \text{ kw per hp}}{0\ 79 \text{ efficiency}} = 9\ 45 \text{ kw input at } \tfrac{1}{2} \text{ speed}$$

For a constant speed motor installation, 1000 hours operation at 18 kw load, requires 18,000 kwhr per season for operation

With 2-speed motor 750 hours at ½ speed and 9 45 kw input = 7080 kwhr.

For full speed operation, 250 hours at 18 kw input = 4500 kwhr

The total kwhr required with 2-speed motor = 7080 + 4500 = 11,580 kwhr.

From the above it is seen that the power consumption is reduced by the use of a 2-speed motor from 18,000 kwhr per year to 11,580 kwhr or approximately a power saving of 35 per cent annually

For power at $ 02 per kwhr, 6400 kwhr saving per year would mean a saving of $128 00 per year which at 15 per cent would justify an increased first cost of $853 00

Further, the fixed charge added due to the increased first cost of a 2-speed motor would be 15 per cent of $210.00, or $31 50 per year, thus the net annual saving after correcting for fixed charged due to increased investment would be $128 00 less $31 50 fixed charge or a net annual saving of $96.50 with a 2-speed motor based on a power cost of $.02 per kwhr. Thus the increased first cost of $210.00 would be returned to the purchaser in $210 00 ÷ $96.50 or 2 18 years

Similarly a $ 03 power the gross saving exclusive of additional fixed charges to increased cost of 2-speed motor would be $192 00, net saving per year $160.50 and therefore, the 2-speed motor would pay for itself in 1 31 years

For $.04 power, the gross saving would be $256 00, net saving, $224 50 and the 2-speed motor would be paid for in 0 935 years

It is safe to assume that any appurtenance which will pay for itself in less than 5 years when computed on the above basis, will be accepted by practically all purchasers of air conditioning equipment, provided an analysis of the above type is properly presented

4 ● Discuss the difference in results obtained in cooling and dehumidifying in an air washer from those obtained in a surface cooling coil.

Air leaves a dehumidifying air washer in a saturated condition at a dew point temperature which can be easily maintained at a constant level by controlling the spray water temperature This saturated air may then be reheated to proper delivery temperature by reheating coils or by mixing with by-passed air

For a set air velocity and a set mean refrigerant temperature, a given cooling coil is capable of absorbing a definite amount of heat Whether the air leaving the coil is saturated or not depends then on the entering dry- and wet-bulb temperatures From, practical operating standpoint, the easiest way to control the output of the cooling coil is by means of the dry-bulb temperature of the conditioned space This means then that the final dew point will vary somewhat depending on entering air conditions.

Summarizing then, the air washers permit close control over both final dry-bulb and final dew point temperatures, while the surface coolers permit close control over the final dry-bulb only.

5 ● For condensing purposes, an air conditioning system uses city water which has an average 70 F supply temperature. The following table lists the number of hours per year during which definite wet-bulb temperatures and corresponding refrigeration rates pertain.

Wet-Bulb Temperature F	No. of Hours per Year	Refrigeration Required Tons
80	6	284
79 – 75	100	233
74 – 70	277	183
69 – 65	330	157
64 – 60	277	144
59 – 55	158	79
54 – 50	52	37
	Total 1200 hours	

If the power requirements of a dichlorodifluoromethane refrigeration system are in accordance with the following data on partial load operation, determine the seasonal power cost at 2 cents per kwhr:

Tons of Refrigeration	**284**	**233**	**183**	**157**	**144**	**79**	**37**
Kw per ton	**0.89**	**0.89**	**0.87**	**0.86**	**0.86**	**0.93**	**0.97**

Seasonal power cost:

Wet-Bulb Temperature F	Ton-Hours	Kwhr
80	6 × 284 = 1,704	1,704 × 0 89 = 1,517
79 – 75	100 × 233 = 23,300	23,300 × 0 89 = 20,750
74 – 70	277 × 183 = 50,700	50,700 × 0.87 = 44,100
69 – 65	330 × 157 = 51,800	51,800 × 0.86 = 44,500
64 – 60	277 × 144 = 39,900	39,900 × 0.86 = 34,300
59 – 55	158 × 79 = 12,500	12,500 × 0 93 = 11,600
54 – 50	52 × 37 = 1,920	1,920 × 0.97 = 1,860
Totals	181,824 ton-hours	158,627 kwhr

The 158,627 kwhr at 2 cents per kwhr will cost $3,173.

The average consumption will be $\frac{158{,}627 \text{ kwhr}}{181{,}824 \text{ ton-hours}} = 0.873$ kw per ton.

6 ● Using the data from Question 5, if city water costs 20 cents per thousand gallons, and if 1.25 gallons are used per minute per ton, estimate the annual water cost.

$$60 \times 1.25 = 75 \text{ gal per ton-hour}$$
$$181{,}824 \text{ ton-hours} \times 75 = 13{,}620{,}000 \text{ gal per year}$$
$$\frac{13{,}620{,}000 \times \$0.20}{1000} = \$2{,}724, \text{ the yearly cooling water cost}$$

7 ● Using the data of Question 5, if a cooling tower were installed for re-using the condensing water, estimate the annual operating cost of a dichlorodifluoromethane refrigeration system if the final temperatures of the water leaving the cooling tower and the kilowatt input per ton are the following:

Tons	**284**	**233**	**183**	**157**	**144**	**79**	**37**
Temperature of water leaving tower, F	**86.7**	**81.8**	**76.5**	**72.1**	**66.4**	**61.3**	**55.6**
Kw input per ton	**1.10**	**0.94**	**0.85**	**0.80**	**0.74**	**0.59**	**0.62**

Wet-Bulb Temperature F	Ton-Hours		Kw per Ton		Kwhr
80	1,704	×	1.10	=	1,875
79 – 75	23,300	×	0.94	=	21,900
74 – 70	50,700	×	0.85	=	43,300
69 – 65	51,800	×	0.80	=	41,400
64 – 60	39,900	×	0.74	=	29,500
59 – 55	12,500	×	0.59	=	7,370
54 – 50	1,920	×	0.62	=	1,200
Totals	181,824 ton-hours				146,545 kwhr

The 146,545 kwhr at 2 cents per kwhr will cost \$2,931

The average consumption will be $\frac{146{,}545 \text{ kwhr}}{181{,}824 \text{ ton-hours}} = 0.805$ kw per ton

8 ● If a steam ejector system were used to secure the refrigeration for the air conditioning system of Question 5, compute the annual steam cost if steam is sold for 53 cents per thousand pounds and if there is an average steam consumption of 20 lb of steam per hour per ton when used with a cooling tower system.

181,824 tons × 20 lb of steam per ton = 3,636,480 lb of steam.
The 3,636,480 lb at 53 cents per thousand pounds will cost \$1,929.

9 ● From the data given in the following table covering auxiliary equipment, make a comparison between the operating costs of the complete dichlorodifluoromethane system of Question 7 and the complete steam ejector cooling system of Question 8. A cooling tower is used for condenser water recovery.

Plant Operation	Dichlorodifluoromethane System	Steam Ejector System
Hours of operation	1200	1200
Cooling tower fan, bhp	17.8	35.6
Cooling tower pump, bhp	30.2	47.8
Chilled water, gpm	1200	1200
Discharge head on chilled water system, ft	75	75
Pump efficiency, per cent	75	75
Motor efficiency, per cent	80	80
Chilled water temperature, F	46	46

The flash tank or evaporator of the steam ejector system is of the open type, the flash water being pumped directly to the sprays of the washer used for cooling the air.

Dichlorodifluoromethane System.

Power requirements,

Cooling tower fan	17.8 bhp
Cooling tower pump	30.2
Total	48 0 bhp

$$\text{Power for cooling tower system} = \frac{48\ 0 \text{ bhp} \times 0.746 \times 1200 \text{ hr}}{0.80 \text{ motor efficiency}} = 53{,}700 \text{ kwhr.}$$

The water cooler in a dichlorodifluoromethane system of the surface type requires no additional pumping head other than the friction drop through the cooler, which in this problem is estimated to be 10 ft. The total pumping head is, therefore, 75 + 10 = 85 ft. Power required for the chilled water system will be,

$$\frac{1200 \text{ gpm} \times 8\ 34 \text{ lb per gallon} \times 85 \text{ ft head}}{33{,}000 \text{ ft lb} \times 0\ 75 \text{ pump efficiency}} = 34\ 3 \text{ bhp}$$

$$\frac{34\ 3 \text{ bhp} \times 0\ 746 \times 1200 \text{ hr}}{0\ 80 \text{ motor efficiency}} = 38{,}300 \text{ kwhr}$$

Thus, the total power required by the auxiliary equipment will be

$$53{,}700 + 38{,}300 = 92{,}000 \text{ kwhr}$$

The 92,000 kwhr at 2 cents per kwhr will cost	$1,840
The power cost of refrigeration, from Question 7, is	2,931
The total annual power cost, using a dichlorodifluoromethane system, is	$4,771

Steam Ejector System.

Power requirements,

Cooling tower fan	35.6 bhp
Cooling tower pump	47 8
Total	83 4 bhp

$$\text{Power for cooling tower systems} = \frac{83\ 4 \text{ bhp} \times 0\ 746 \times 1200 \text{ hr}}{0\ 80 \text{ motor efficiency}} = 93{,}300 \text{ kwhr}$$

In the flash tank or water cooler of the steam ejector system, the water is at a pressure corresponding to the chilled water temperature required In this case it is at 46 F, which corresponds to an absolute pressure of 0.1532 lb per sq in or 0 3118 in. Hg This increases the pumping head on the chilled water circulating pump by 14.7 − 0 15 = 14.55 lb per square inch, or 33.5 ft. The total pumping head is, therefore, 75 0 + 33 5 = 108.5 ft.

$$\frac{1200 \text{ gpm} \times 8.34 \text{ lb per gallon} \times 108\ 5 \text{ ft head}}{33{,}000 \text{ ft-lb} \times 0\ 75 \text{ pump efficiency}} = 43\ 7 \text{ bhp.}$$

$$\frac{43.7 \text{ bhp} \times 0.746 \times 1200 \text{ hr}}{0.80 \text{ motor efficiency}} = 48{,}800 \text{ kwhr.}$$

The total power required by the auxiliary equipment is

$$93{,}300 + 48{,}800 = 142{,}100 \text{ kwhr}$$

The 142,100 kwhr at 2 cents per kwhr will cost	$2,842
The cost of the steam, from Question 8, is	1,929
The total annual power cost, using a steam ejector system, is	$4,771

These calculations indicate that for the assumptions made, both the dichlorodifluoromethane system and the steam ejector system would cost 2 6 cents per ton-hour to operate In order to obtain a complete analysis it would be necessary to compare the fixed charges which include interest, depreciation, obsolescence, and maintenance These are customarily computed at 15 per cent of the initial cost per annum. To this cost must be added the cost of refrigerant make-up per year. In the steam system this cost is negligible, while in the dichlorodifluoromethane system it may be approximated at about 10 per cent of the refrigerant charge per year

Chapter 12

HUMIDIFICATION, DEHUMIDIFICATION AND WATER COOLING EQUIPMENT

Air Washers, Apparatus for Direct Industrial Humidification, Spray Generation and Distribution, Self-Contained Humidifiers, Atmospheric Water Cooling Equipment, Design Wet-Bulb Temperatures, Cooling Ponds, Spray Cooling Towers, Natural Draft Deck Type Towers, Mechanical Draft Towers, Winter Freezing

THE several available types of spray equipment are discussed in this chapter which are used for properly humidifying and dehumidifying the air circulated in a comfort or industrial air conditioning system or for cooling the condensing water of a refrigeration system.

AIR WASHERS

An air washer is essentially a chamber in which air is brought into intimate contact with water to (a) regulate the moisture content of the air, and (b) to wash dust and dirt particles out of the air. The air comes in contact with the water by passing it through a spray of water broken up into a fine mist or by passing it over surfaces wetted by a continuous flow of water; hence the classification: spray, scrubber, and combination spray[1] and scrubber type washers (See Figs. 1 and 2.)

A washer chamber may be constructed of wood, or stone, but it is most often constructed of sheet metal. The lower portion of it is specially designed as a tank to receive the water dropping through the chamber and to serve as a reservoir from which the water may be recirculated.

It is desirable that air leaving a washer contain no water in suspension, and for this reason eliminators are provided at the washer outlet. These may be in the form of plates or baffles upon which the free moisture is deposited as the air is deflected through several changes from its original direction of flow.

In some washers, units of either steel wool or special glass fiber sections serve as eliminators. However, specially designed sheet metal plates are more generally used because they offer the least resistance to the flow of air, while performing effectively the function of moisture elimination. They also have the advantage of acting as scrubber surfaces when flooded When the air has a high sulphur content tending to form a weak acid, the eliminators have been constructed of ribbed, wire glass plates set in a cypress frame so that frequent renewal will not be necessary.

It is essential to uniform performance in a washer, that air enter evenly

distributed over the washer inlet. To insure this, a perforated plate or eliminator plates are installed at the inlet. Eliminator plates are now more generally used. They serve a second purpose in preventing the escape of spray through the washer inlet

Scrubber type washers, used mainly to wash heavy reclaimable products from the air, are generally composed of one to three eliminator type baffle scrubber plates across the air stream. Water is supplied at the top of the scrubber plate with flooding nozzles spaced on about one foot centers across the top of the washer. The capacity of these nozzles vary with the manufacturer but a fair value of 5 gpm may be used. The pressure at the nozzle is about 5 lb per square inch.

Water is supplied to spray type units through atomizing nozzles generally arranged in banks across the washer. The nozzles spray either in the

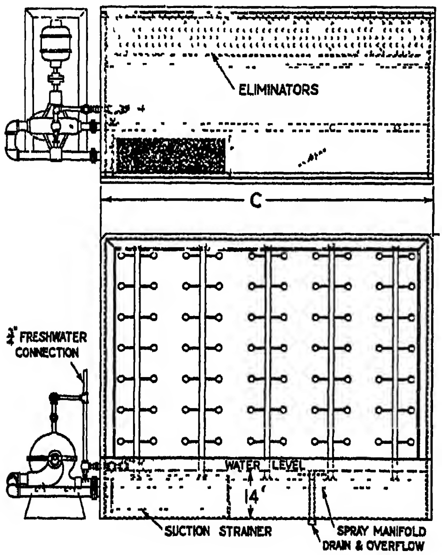


Fig. 1. Typical Single Bank Air Washer

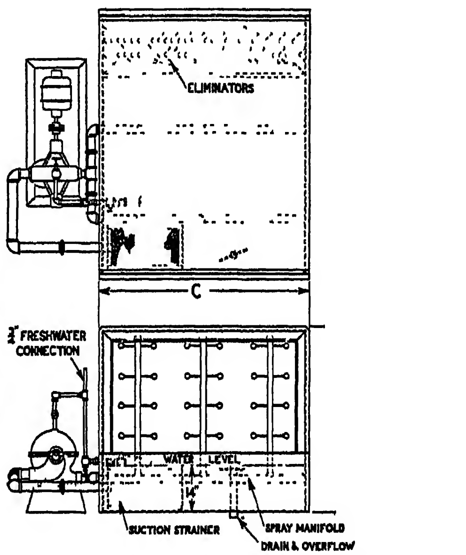


Fig. 2. Typical Two Bank Air Washer

direction of the air flow, that is, downstream, or against the air flow, or upstream. Nozzle capacities vary with the manufacturer, from 1½ to 2 gpm at a water pressure effective for atomization of about 25 lb per square inch. The spacing of spray nozzles is determined by the water requirements of the particular installation. A spray type washer may contain one, two or three banks of nozzles depending upon its application.

When an air washer is used for cleaning air it removes impurities and dusts. In general it does not function as efficiently in this service as a filter. For non-microscopic wettable dust its efficiency averages about 50 per cent, unless the concentration of dust is high. Its effectiveness in removing greasy microscopic dust is practically negligible as is also its deodorizing ability.

When a washer is used to regulate the moisture content of air it adds

moisture to (humidifies) or removes moisture from (dehumidifies) the air to achieve the desired moisture content.

When air passes through a washer wherein water is circulated without the addition or removal of heat, the air tends to become saturated at its entering wet-bulb temperature. What occurs here is partial or complete adiabatic saturation. The total heat content of the air is unchanged, inasmuch as the dry-bulb temperature of the air drops in proportion to the amount of additional water evaporated. This action is also known as evaporative cooling. A measure of the washer's effectiveness under these conditions is its saturating efficiency which is equal to the drop in dry-bulb temperature in per cent of the entering wet-bulb depression. Other things being equal, the saturating efficiency of a spray type washer is a function of the number of spray banks and the direction in which they spray. The following table gives a general comparison:

HUMIDIFYING

1 bank—downstream	60–70 per cent
1 bank—upstream	65–75 per cent
2 banks—downstream	85–90 per cent
2 banks—1 upstream and 1 downstream	90–95 per cent
2 banks—upstream	90–95 per cent

When air passes through a washer wherein the circulated water is either cooled or heated before being returned to the spray chamber, a heat interchange between the air and water occurs, and the air tends to become saturated at the temperature of the leaving water. The extent to which the leaving air and leaving water temperatures approach each other is an index to the effectiveness of the washer under the operating conditions. The total heat absorbed by the water in the process equals the total heat given up by the air, or the heat given up by the water equals the heat absorbed by the air. Depending on whether the moisture content of the air is increased or decreased during the operation, humidification or dehumidification occurs. Heat will be added to or removed from the air as the water supplied is of a higher or a lower temperature than the wet-bulb temperature of the entering air.

For dehumidifiers the ratio of the difference between the leaving wet-bulb and the leaving water to the difference between the entering wet-bulb and the entering water may be figured as follows:

DEHUMIDIFYING

3 banks—2 upstream and 1 downstream	0 per cent
2 banks—upstream	5 per cent
2 banks—1 upstream and 1 downstream	10 per cent
2 banks—downstream	15 per cent
1 bank—upstream	20 per cent
1 bank—downstream	35 per cent

Humidifiers may be figured on the same basis as dehumidifiers; the leaving water temperature, of course, will be higher than the wet-bulb temperature of the leaving air.

The problem of cooling or heating the circulated water before returning it to the washer chamber is generally external to the unit as illustrated in

Fig. 3. It will suffice here to note that heating is generally accomplished by passing the water through closed hot water heaters or by injecting steam into the water circuit, cooling, by passing the water through closed coolers or over refrigerating coils in a Baudelot chamber Often in a cooling and dehumidifying application, the refrigerating coils are located within the washer chamber.

Washers are sometimes arranged in two or more stages to cool through long ranges or to increase the over-all efficiency of heat transfer between air and the cooling or heating medium (water, brine, etc.). A multi-stage washer is equivalent to a number of washers in series arrangement. Each stage is in effect a separate washer.

Usually the catalog capacity of a washer is expressed in cubic feet of air per minute and is based upon an air velocity of 500 feet per minute through the gross cross-sectional area of the unit above the water level in its tank. At this rating spray type washers handle about 2½ gpm of

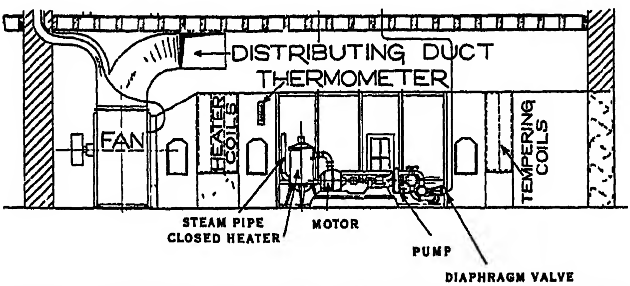


FIG 3. AIR WASHER WITH SPRAY WATER HEATING ARRANGEMENT

water per bank per square foot of area, that is, about 5 gpm per bank per 1000 cfm. These proportions of air, water, area, and velocity may be departed from to meet the needs of some particular job, but certain limiting relationships should be observed. Two of the more important items are:

a Choose a washer for air velocities above approximately 300 fpm and below approximately 600 fpm. Velocities outside this range are likely to result in faulty elimination of entrained moisture.

b When a high saturating efficiency is required, select a two or three bank spray type unit, having a total water capacity of not less than 15 gpm per 1000 cfm.

For a single stage air washer, a 15 F drop in wet-bulb temperature of the air passing through the washer is about the maximum that should be anticipated. For greater decrease in wet-bulb temperature, multi-stage washers should be utilized. A rise of 6 F should be the calculated maximum for the spray water.

Brine has been utilized as a cooling medium in place of water in air washers to obtain low outlet temperatures impossible with water. However, proper elimination of brine spray carry over into the duct work is much more difficult than with water. A greater number of air deflections should be used in the eliminators, and the air velocity maintained at a minimum.

The area of a washer may be dictated by space limitations outside the

washer, such as headroom, or by space requirements inside washer, such as face area needed by a bank of cooling coils. The length of a washer is determined by the number of spray banks, or scrubber plates, and if cooling coils are installed in the unit, by the number of banks of coils. Roughly, a spray space of about 2 ft 6 in. in length is required for each bank of sprays, (the *leaving* eliminators require about 1 ft 6 in., *entering* eliminators about 1 ft).

The resistance to air flow through an air washer varies with the type of eliminators, number of banks of sprays, direction of spray, air velocity, type of scrubber plates, size and type of cooling coils if located in the washer. Manufacturers should be consulted to obtain the resistance for a particular installation.

APPARATUS FOR DIRECT INDUSTRIAL HUMIDIFICATION

Apparatus for industrial air conditioning may be divided into two distinct groups, namely, (1) *humidifiers* for increasing the moisture content of the air and for producing cooling by evaporation and (2) *dehumidifiers* for removing moisture from the air and for producing cooling by contact with water or surfaces at a lower temperature than the air.

Strictly speaking, humidity control alone, whether it involves humidification or dehumidification, is not *air conditioning*. To be entitled to this classification according to the definition in Chapter 44, the process should include the *simultaneous control* of temperature, humidity and air motion.

Industrial humidifiers may be divided into the following general types, according to the method of operation: (1) *Direct*, which spray into the room; (2) *Indirect*, which introduce moistened air; and (3) *Combined* direct and indirect.

Spray Generation

Spray generation is obtained by (1) atomization, (2) impact, (3) hydraulic separation, and (4) mechanical separation.

Atomization involves the use of a compressed air jet to reduce the water particles to a fine spray. With the *impact* method, a jet of water under pressure impinges directly on the end of a small round wire. Where *hydraulic separation* is employed, a jet of water enters a cylindrical chamber and escapes through an axial port with a rapid rotation which causes it immediately to separate in a fine cone-shaped spray. In the *mechanical separation* process, water is thrown by centrifugal force from the surface of a rapidly revolving disc and separates into particles sufficiently small to be utilized in certain types of mechanical humidifiers.

Spray Distribution

Spray distribution is obtained by (1) air jet, (2) induction, and (3) fan propulsion.

The air jet which generates the spray in atomizers also carries the spray through a space sufficient for its distribution and evaporation, and this method of distribution is termed *air jet*. Where distribution is obtained by *induction*, the aspirating effect of an impact or centrifugal spray jet is utilized to induce a current of air to flow through a duct or casing, and

this air current distributes the spray. *Fan propulsion* obviously consists of the utilization of fans to entrain and distribute the spray.

Industrial type direct humidifiers are commonly classified as (1) atomizing, (2) high-duty, (3) spray and (4) self-contained or centrifugal.

Atomizing Humidifiers

There are several types of atomizing humidifiers, all of which rely upon compressed air as the atomizing and distributing agency, similar to the familiar method used in ordinary nasal atomizers. Compressed air (ordinarily about 30 lb per square inch) is supplied from a centrally-located air compressor through pipe lines to the atomizing units. The air lines are usually horizontal and parallel to water lines which supply water by gravity from a float tank. The water in the tank is maintained at a constant level slightly lower than the outlets of the atomizers themselves and is drawn constantly to the atomizer by aspiration when compressed air is supplied. This aspiration ceases and the flow of water stops when the air supply is cut off. The water should not be supplied under pressure to atomizers because of the possibility of leakage, drip, or coarse spray which cannot be permitted when water is supplied by aspiration.

High-Duty Humidifiers

Water is supplied to high-duty humidifiers under high pressure (usually about 150 lb per square inch) through pipe lines from a centrally-located pumping unit. The spray-generating nozzle which is of the impact type is located in a cylindrical casing. A drainage pan provides for the collection and return of unevaporated water which flows through a return pipe to a filter tank, from which it is recirculated. A powerful air current is forced through the humidifier by means of a fan mounted above the unit.

The air enters from above, is drawn through the head, charged with moisture, and cooled to the wet-bulb temperature. It then escapes from the opening below at a high velocity in a complete and nearly horizontal circle. The spray is quickly evaporated and the resulting vapor is rapidly and thoroughly diffused. This effective distribution of fine spray over the maximum possible area insures complete and extremely rapid vaporization even at the highest humidities.

Spray Humidifiers

This type of humidifier consists of an impact spray nozzle in a cylindrical casing with a drainage pan below it. The aspirating effect of the spray nozzle induces a moderate air current through the casing which distributes the entrained spray. The general method of circulating and returning the water is similar to that employed for high-duty humidifiers. A suitable pump and centrally-located filter tank are required.

The spray and high-duty types of humidifiers have many features in common but the latter, because of its finer spray and greater capacity, is often considered better adapted for producing high humidities.

Self-Contained Humidifiers

The self-contained or centrifugal humidifier has the ability to generate and distribute spray without the use of air compressors, pumps, or other

auxiliaries. These may be used either singly or in groups. In large installations, where suitable connections are provided to permit the cleaning and servicing of individual units without affecting the room as a whole, group control of the water and power may be employed.

Where large quantities of power are generated in a limited space and where a comparatively high relative humidity is required, it is often feasible and economical to use a combination of direct and indirect humidification. The indirect humidification provides the desired quantity of ventilation and cooling, and the additional direct humidification provides for increase in humidity without interfering with the ventilation or the cooling effected by the indirect system.

In general, it may be stated that direct humidification is most satisfactory where high humidities are desired but where little cooling, ventilation or air motion is required. Therefore, the indirect system is most applicable where either low or high relative humidities are desired with maximum cooling and ventilation effect. For conditions that require an unusually large amount of heat to be absorbed by ventilation, together with the maintenance of high humidities, it is often preferable to make use of the combination system of indirect and direct humidification. If the indirect system alone were used it would mean an unusually large volume of air to be handled, which might interfere, due to air motion, with production, even though it would result in greater cooling effect. If direct humidification alone were used, no ventilation would be obtained, with consequently higher room temperatures.

ATMOSPHERIC WATER COOLING EQUIPMENT

To successfully operate a refrigerating plant or a condensing turbine, the heat from the compressed refrigerant or the discharged steam must be removed and dissipated. This is accomplished ordinarily by first transferring the heat of the gas to water in a heat exchanger. If the plant is situated on the banks of a river or lake, an intake may be had upstream or at a considerable distance from the discharge, to prevent mixing of the heated discharged water with the inlet water. If the source of water is a city supply or well water, the discharge water may be run into the nearest sewer or open waterway. Lacking an unlimited water supply, or in cases where city water is too expensive or where the water available contains dissolved salts which would quickly form scales on the heat-exchanging apparatus, it is necessary to recirculate the water, and to cool it after each passage through the heat-exchanger by exposure to air in an atmospheric water cooling apparatus.

Air has a capacity for absorbing heat from water when the wet-bulb temperature of the air is lower than the temperature of the water with which it is in contact. The rapidity with which this transfer of heat occurs depends upon (1) the area of water in contact with the air, (2) the relative velocity of the air and water, and (3) the difference between the wet-bulb temperature of the air and the temperature of the water. Because the changes in rate do not occur in direct proportion to changes in the governing factors, data on the performance of atmospheric water cooling equipment are largely empirical.

As the heat content of the air increases, its wet-bulb temperature rises.

(See Chapter 1.) Because it is impractical to leave the air in contact with water for a long enough time to permit the wet-bulb temperature of the air and the temperature of the water to reach equilibrium, atmospheric water cooling equipment aims to circulate only enough air to cool the water to the desired temperature with the least possible expenditure of power.

In an air washer, humidifier or dehumidifier, the air is first conditioned by water to change its moisture and temperature, and it is then sent to the place where it is to be used. In water cooling equipment the temperature of the water is reduced by air, and the cooled water is carried to its point of usage. In the air washer, an excess of water is used to condition a fixed quantity of air, while in water cooling equipment, an excess quantity of air is used to cool a fixed quantity of water.

Both types of equipment have a common basis of design, however, in that the size of the equipment is determined by the quantity of air that must be handled. With the air washer, the size of the equipment is fixed by the quantity of air to be conditioned, and the amount of conditioning is controlled by the quantity and temperature of the water supplied and its method of application With water cooling apparatus, its size and the quantity of air required bear no direct relation to the quantity of water being cooled, but vary through a wide range for different services and conditions.

Sizes of Equipment

Assuming a definite quantity of water to be cooled, the size and design of atmospheric cooling equipment are affected by the following factors:

1. Temperature range through which the water must be cooled
2. Number of degrees above the wet-bulb temperature of the entering air to which the water temperature must be reduced
3. Temperature of the atmospheric wet-bulb at which the required cooling must be performed.
4. Time of contact of the air with the water. (This involves height or length of the apparatus and velocity of air.)
5. Surface of water exposed to each unit quantity of air.
6. Relative velocity of air and water

Items 1, 2, and 3 are established by the type of service and geographical location, while items 4, 5, and 6 depend upon the design of the equipment.

The establishment of a proper cooling range depends upon:

1. Type of service (refrigerating, internal combustion engine and steam condensing).
2. Wet-bulb temperature at which the equipment must operate satisfactorily.
3. Type of condenser or heat-exchanger used.

Because the design of an entire plant is usually affected by the quantity and temperature of the cooling water supply, plants should be designed for cooling water conditions which can be most efficiently attained. The first consideration is usually the limiting temperature of the plant. For example, if an ammonia compressor refrigerating plant is to be designed for 185 lb head pressure as a normal maximum, the limiting temperature of the ammonia in the condenser is 96 F. Should the ammonia temperature go above this figure the head pressure will exceed 185 lb and power consumption increases. To obtain this head pressure, the temperature of

the circulating water leaving the condenser must always be less than 96 F by an amount depending upon the size and design of the condenser, the quantity of water being circulated, and the refrigerating tonnage being produced. A condenser having a large surface per ton of refrigeration may be designed to operate satisfactorily with the leaving hot water temperature within 3 or 4 F of the ammonia temperature corresponding to the head pressure, while a small condenser might require a 10 F difference.

Table 1 lists several gases with data as to the temperatures and pressures for which commercial condensers are designed. Internal combustion engines have limiting hot water temperatures of 125 F to 140 F. The cooling of such fluids as milk or wort has variable requirements and is usually done in counter-flow heat-exchangers in which the leaving circulating water is at a much higher temperature than is the leaving fluid.

TABLE 1. CONDENSER DESIGN DATA

GAS	MAXIMUM PRESSURE DESIRED IN CONDENSER	GAS TEMPERATURE IN CONDENSER F	LEAVING HOT WATER TEMPERATURE F	
			Best Design	Average Design
Steam	28 in. vacuum	99 7	97	93
Steam	27 in. vacuum	114 3	110	105
Steam	26 in vacuum	126 0	120	114
Ammonia	185 lb gage head pressure	96 0	92	88
Carbon dioxide	1030 lb gage head pressure	86.0	83	81
Methyl chloride	102 lb gage head pressure	100.0	96	92
Dichlorodifluoromethane	117 lb gage head pressure	100.0	96	93

The temperature range, once the hot water temperature is approximately known, depends upon:

1 Maximum wet-bulb temperature at which the full quantity of heat must be dissipated.

2. Efficiency of the atmospheric cooling equipment considered

Design Wet-Bulb Temperatures

The maximum wet-bulb temperature at which the full quantity of water must be cooled through the entire range is never, in commercial design, the *maximum* wet-bulb temperature ever known to exist at the location nor the *average* wet-bulb temperature over any period. The former basis would require atmospheric cooling equipment several times greater than normal size, and the latter would result during a large part of the time, in higher condenser water temperatures than those for which the plant was designed. For instance, the maximum wet-bulb temperature recorded in New York City is 88 F, and the July noon average for 64 years is close to 68 F. Yet in the years 1925 to 1931, inclusive, there were but 6 hours per year when the wet-bulb temperature reached 80 F or more, and there were 975 hours in the average summer (June to September,

inclusive) when the wet-bulb temperature was 68 F or above. As these 975 hours represent a third of the summer period, cooling equipment based upon the noon average July wet-bulb of 68 F would be inadequate. Commercial practice is to choose a wet-bulb temperature for refrigeration design purposes which is not exceeded during more than 5 to 8 per cent of the summer hours (75 F for New York City), with somewhat lower requirements for steam turbines and internal combustion engines. This difference is made because the heaviest load on a refrigerating plant is coincident with high wet-bulb temperatures, whereas the heaviest electric power demand occurs either in the winter or after nightfall in summer, when the wet-bulb temperature is low. Table 1, Chapter 8, shows safe design wet-bulb temperatures which will not be exceeded more than 8 per cent of the time in an average summer.

Knowing the hot water temperature and the wet-bulb temperature for which the equipment must be designed, the cold water temperature must

Table 2 Efficiency of Atmospheric Water Cooling Equipment

Equipment	Cooling Efficiency—Per Cent		
	Minimum	Usual	Maximum
Spray Ponds	30	45 to 55	60
Spray Towers	40	45 to 55	60
Natural Draft Deck or Atmospheric Towers	35	50 to 70	90
Mechanical Draft	35	55 to 75	90

be chosen to place the requirement within the efficiency range of the type of atmospheric water cooling apparatus to be used. Efficiency of atmospheric water cooling apparatus is expressed as the percentage ratio of the actual cooling range to the possible cooling range. Since the wet-bulb temperature of the entering air is the lowest temperature to which the water could possibly be cooled this is:

Percentage cooling efficiency of atmospheric water cooling equipment =

$$\frac{(\text{hot water temperature} - \text{cold water temperature}) \times 100}{\text{hot water temperature} - \text{wet-bulb temperature of entering air}}.$$

Efficiencies of various types of atmospheric water cooling apparatus vary through wide limits, depending upon air velocity, concentration of water per square foot of area, and the type of equipment. The commercial range of efficiencies is given in Table 2 although unusual designs may operate outside these ranges.

From consideration of the factors which include the cooling range and design wet-bulb temperature, the quantity of water required can be calculated from the amount of heat to be dissipated. The normal amounts of heat to be removed from various parts of the cooling equipment are:

Compressor refrigeration	220 to 270 Btu per minute per ton
Condenser turbine	950 to 980 Btu per pound of steam
Steam jet refrigerating apparatus	1030 to 1150 Btu per pound of steam
Diesel engine	2800 to 4500 Btu per horsepower

Cooling Ponds

A natural pond is often used as a source of condensing water. The hot water should be discharged close to the surface at the shore line, as natural air movement over the surface of the water will cause evaporation and carry away heat. Because increased density due to the loss of heat causes the cooled water to sink to the bottom of the pond, the suction connection for intake water should be placed as far below the surface as possible, and at as great a distance from the discharge as practicable.

Spray Cooling Ponds

The spray pond consists of a basin, above which nozzles are located to spray water up into the air. Properly designed spray nozzles break up the water into small drops, but not into a mist because the individual drops must be heavy enough to fall back into the basin and not drift off. The water surface exposed to the air for cooling is the combined area of all the small drops. Since the rate of heat removal by atmospheric water cooling is a function of the area of water exposed to the air, the difference in temperature between the water and the wet-bulb temperature of the air, the relative velocity of air and water, and the duration of contact of the air with the water, a much larger quantity of heat may be dissipated in a given area with the spray pond than with the cooling pond, because of (1) the speed with which the drops travel as they are propelled into the air and fall back into the water basin, (2) the increased wind velocity at a point above the surrounding structures or terrain, (3) the increased volume of air used, and (4) the vastly increased area of contact between air and water.

Spray pond efficiencies are increased by (1) elevating the nozzles to a higher point above the surface of the water in the basin, (2) increasing the spacing between nozzles of any one capacity, (3) using smaller capacity nozzles, to decrease the concentration of water per unit area, and (4) using smaller nozzles and increasing the pressure to maintain the same concentration of water per unit area. Usual practice is to locate the nozzles from 3 ft to 6 ft above the edge of the basin, to supply from 5 lb to 12 lb pressure at the nozzles, using nozzles spraying from 20 gpm to 60 gpm each and spacing them so the average water delivered to the surface of the pond is from 0.1 gpm per square foot in a small pond to 0.8 gpm per square foot in a large pond.

Increasing the pressure, spacing the nozzles farther apart, or increasing the elevation of the nozzles will increase the cross-section of spray cloud exposed to the air, and therefore increase the quantity of air coming in contact with the water. Best results are obtained by placing the nozzles in a long relatively narrow area located broadside to the wind.

Spray ponds may be located on the ground if they have an earthen or a concrete basin, or they may be placed on roofs having special waterproof roofing. To prevent excessive drift loss, or the carrying of entrained water beyond the edge of the pond by the air on the leeward side, louver fences are required for roof locations and for those ground locations where space is so restricted that the outer nozzles cannot be located at least 20 ft to 25 ft from the edge of the basin. Such fences usually are constructed of horizontal louvers overlapping so the air is forced to turn a

corner in passing through the fence, and the heavier drops of water are thrown back, owing to their inertia. The louvers also restrict the flow of air, particularly at the higher wind velocities, and thus further reduce the possibility of water being carried off. The height of an effective fence should be equal to the height of the spray cloud. Louver boards are preferably of red gulf cypress or California redwood supported on cast-iron, steel or wood posts. Where building ordinances forbid the use of combustible materials, sheet metal is customarily used.

Algae formations may be a considerable nuisance in a spray pond. Such growths are killed by the periodic addition of potassium permanganate to the pond water. Addition of the dissolved chemical should be made until the water holds a faint pink color for at least 15 min.

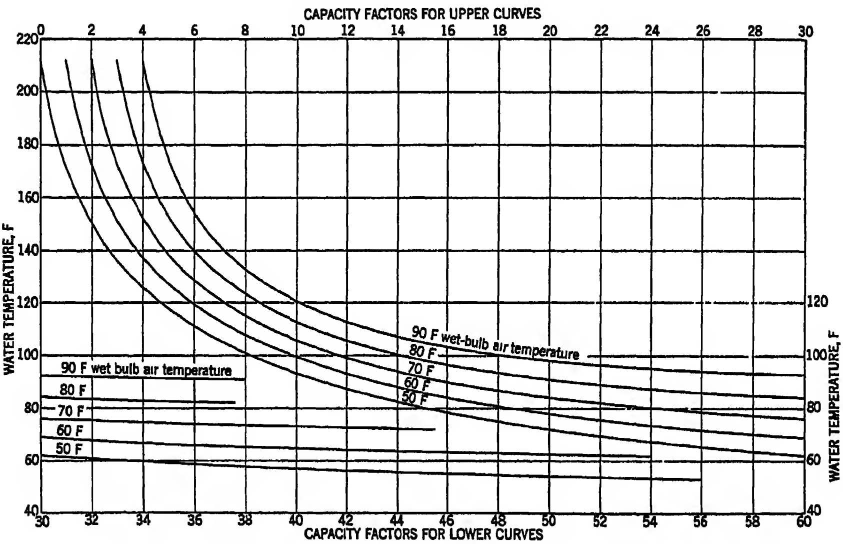


FIG. 4. NATURAL DRAFT COOLING TOWER DESIGN CURVES FOR DETERMINING CAPACITY FACTORS

Spray Cooling Towers

Where not more than 30,000 Btu per minute are to be dissipated, the spray cooling tower is a satisfactory apparatus. The word *tower* in this connection is somewhat of a misnomer as the apparatus is essentially a narrow spray pond with a high louver fence. As usually built, the nozzles spray down from the top of the structure and the distance from the center of the nozzle system to the fence on either side is not more than half the distance that the nozzles are elevated above the water basin. Heights range from 6 ft to 15 ft and the total width of a structure is not usually greater than its height. Spray cooling towers occupy less space on small jobs than spray ponds of equivalent capacities because the towers have a capacity of from 0.6 gpm to 1.5 gpm per square foot of tower area. The

louvers are continually wet, and so add to the surface of water exposed to the cooling air.

Natural Draft Deck Type Towers

In past years most of the atmospheric water cooling on refrigeration work has been done with natural draft deck type towers, which are also referred to as *wind* or *atmospheric* towers. These towers consist of heavy wooden or steel framework from 15 ft to 80 ft high and from 6 ft to 30 ft wide, having open horizontal lattice-work platforms or decks at regular intervals from top to bottom, and a catch basin at the foot. The hot water is distributed over the upper part of the structure by means of troughs, splash heads, or nozzles, and it drips from deck to deck down to the basin. The object of the decks is to arrest the fall of the water so as to present efficient cooling surfaces to the air, which passes through the tower parallel to the decks. The decks also add to the area of water surface exposed to the air, but since they furnish a resistance to air flow, too many decks are a detriment.

TABLE 3 NATURAL DRAFT SPRAY DISTRIBUTION COOLING TOWER DESIGN SIZES AND COEFFICIENTS

COEFFICIENT K (FORMULA 1)	NO OF DECKS	DECK SPACING FT CENTERS	UNIT LENGTH FT	DECK WIDTH FT
2860	9	3	11 75	12 0
3430	11	3	11.75	12 0
3620	17	1 5	11 75	12 0
4390	21	1 5	11 75	12 0

To prevent the loss of water on the leeward side of the tower, wide splash boards are attached at regular intervals from top to bottom. These boards or louvers extend outward and upward, and in most designs the top edge of each louver extends above the bottom edge of the one above it.

Efficiency of a deck tower is improved, within limits, by increased height, increased length, or increased width. The first two increase the area of water exposed to the wind, and the latter increases the time of contact of the air with the water.

A rating formula has been developed for determining the number of standard tower sections based upon the values given in Table 3 and the capacity factors obtained from Fig. 4 with estimated design water and air temperatures. The rating formula is,

$$N = \frac{Q\,(C_1 - C_2)}{K} \tag{1}$$

where

N = number standard tower sections required.

Q = quantity water circulated, gallons per minute

C_1 = capacity factor corresponding to temperature water leaving tower.

C_2 = capacity factor corresponding to temperature water entering tower

Example 1 It is desired to cool 1000 gpm of water from 100 to 80 F in a natural draft cooling tower utilizing spray distribution based upon an outside design wet-bulb air

temperature of 70 F. Determine the number of standard tower sections required for these conditions if each section has a unit length of 11 75 ft and a deck width of 12 ft with 11 decks arranged vertically on 3 ft centers

Solution Refer to Fig 4 and determine capacity factors for insertion in Formula 1 With an entering water temperature of 100 F follow horizontally left to the 70 F air temperature curve and read on the upper scale a capacity factor C_2 = 11 65 Similarly for leaving water temperature 80 F, C_1 = 24 00 Select from Table 3, for the design conditions stated a coefficient K = 3430 Substituting in Formula 1

$$N = \frac{1000\ (24\ 00 - 11\ 65)}{3430} = 3\ 6 \text{ sections}$$

Select 4 sections which would result in space requirements of 4 (11 75 × 12) = 563 sq ft of area on decks

Wind Velocities on Natural Draft Equipment

Since natural air movement is the prime requirement for a deck type tower, spray cooling tower, or spray pond, the apparatus must be designed to produce the desired cooling on days when the wind velocity is below average when the wet-bulb temperature is at the maximum chosen for design, and when the plant is operating at full load. The apparatus must also, for best results, be located with its longest axis at right angles to the direction of the prevailing hot weather breeze. Table 1 Chapter 8, gives the average summer wind velocities and directions in representative cities. Natural draft cooling equipment should be designed to operate properly with *not more than one-half* of the *average* wind velocity, and in no case should it need a wind velocity of more than 5 mph. It is obvious that natural draft towers and other natural draft equipment must be so located that they are not obstructed by trees, buildings, or other wind deflectors.

Mechanical Draft Towers

Mechanical draft towers usually consist of vertical shells, constructed of wood, metal, or masonry, in which water is distributed uniformly at the top and falls to a collecting basin at the bottom. The inside of the tower may be filled with wood checker-work over which the water drips, or the water surface may be presented to the air by filling the entire inside of the structure with spray from nozzles. Air is circulated through the tower from bottom to top by forced or induced draft fans. Since the air flows counter to the water, the air is in contact with the hottest of the water just before leaving the top of the tower, and each unit of air picks up more heat than a similar unit would on natural draft equipment, so the mechanical draft tower cools water by using less air than the other types of equipment need. As movement of the air through the towers is obtained by power-consuming fans, it is essential that the air used be reduced to a minimum so as to secure the lowest possible operating cost.

The efficiency of a mechanical draft tower is increased by increasing height, area, or air quantity. Increasing the height increases the length of time the air is in contact with the water without affecting seriously the fan power required, but it increases the pumping power needed. Increasing the area while maintaining constant fan power increases the air quantity somewhat and because of louvered velocities it increases the time this air is in contact with the water. The surface area of water in

TABLE 4 MECHANICAL DRAFT COOLING TOWER DESIGN FACTORS[a]

Cooling Temperature Range F	Tower Area Required per Gallon of Water, in Square Feet													
	Approach to Wet-Bulb Temperature, F													
	3	4	5	6	7	8	9	10	12	14	15	16	18	20
5	0.51	0.45	0.37	0.31	0.27	0.23								
6	0.57	0.48	0.41	0.35	0.31	0.27	0.23	0.21						
7	0.62	0.52	0.45	0.40	0.34	0.30	0.27	0.24						
8	0.67	0.57	0.50	0.43	0.38	0.33	0.30	0.27	0.22					
9	0.70	0.61	0.53	0.47	0.42	0.37	0.33	0.30	0.24	0.20	0.18	0.17		
10	0.72	0.64	0.56	0.50	0.44	0.40	0.35	0.32	0.27	0.22	0.20	0.18	0.16	
12	0.78	0.69	0.61	0.55	0.50	0.44	0.40	0.36	0.30	0.25	0.23	0.22	0.18	0.16
14	0.81	0.72	0.65	0.58	0.53	0.48	0.44	0.40	0.33	0.28	0.26	0.24	0.21	0.18
16	0.85	0.75	0.68	0.61	0.56	0.50	0.46	0.42	0.36	0.31	0.29	0.26	0.23	0.20
18	0.87	0.78	0.70	0.64	0.59	0.53	0.49	0.45	0.38	0.33	0.31	0.29	0.25	0.22
20	0.89	0.80	0.72	0.67	0.61	0.56	0.51	0.47	0.40	0.35	0.33	0.31	0.27	0.23
22	0.91	0.82	0.74	0.69	0.63	0.58	0.52	0.49	0.42	0.37	0.34	0.32	0.29	0.25
24	0.92	0.83	0.76	0.70	0.65	0.59	0.55	0.50	0.44	0.38	0.36	0.34	0.30	0.27
26	0.94	0.85	0.78	0.71	0.67	0.61	0.57	0.52	0.45	0.40	0.37	0.35	0.31	0.28
28	0.96	0.86	0.80	0.73	0.67	0.62	0.58	0.53	0.46	0.41	0.38	0.36	0.32	0.29
30	0.98	0.88	0.81	0.74	0.69	0.64	0.59	0.55	0.47	0.42	0.40	0.37	0.33	0.30
32	0.99	0.89	0.82	0.75	0.70	0.65	0.60	0.56	0.49	0.43	0.41	0.38	0.34	0.31
34	1.01	0.91	0.83	0.77	0.71	0.66	0.61	0.57	0.50	0.44	0.42	0.40	0.35	0.32
36		0.92	0.85	0.79	0.72	0.68	0.62	0.58	0.51	0.45	0.43	0.40	0.36	0.33
38		0.93	0.85	0.80	0.73	0.69	0.64	0.59	0.52	0.46	0.44	0.41	0.37	0.34
40		0.94	0.86	0.81	0.74	0.70	0.65	0.60	0.53	0.47	0.45	0.42	0.38	0.35
45		0.97	0.89	0.83	0.77	0.72	0.67	0.63	0.55	0.49	0.47	0.44	0.40	0.36
50		0.99	0.91	0.85	0.81	0.74	0.69	0.65	0.57	0.51	0.49	0.46	0.42	0.38
55			0.93	0.87	0.83	0.77	0.71	0.68	0.59	0.53	0.50	0.49	0.44	0.40
60				0.89	0.84	0.79	0.73	0.69	0.61	0.55	0.52	0.50	0.45	0.42
65					0.85	0.81	0.75	0.71	0.63	0.57	0.54	0.50	0.47	0.43
70						0.83	0.77	0.73	0.65	0.58	0.55	0.52	0.48	0.45
75						0.85	0.79	0.75	0.67	0.60	0.57	0.54	0.50	0.47
80							0.81	0.77	0.68	0.62	0.59	0.56	0.51	0.48

[a]Based upon 70 F wet-bulb air temperature, with air velocity through tower of 300 fpm and a spray eliminator located at top of tower

Correction Factors Multiply above loading by correction factor corresponding to the design wet-bulb temperatures as follows

40 — 2.25	62 — 1.26	68 — 1.08	74 — 0.90	80 — 0.75
45 — 1.95	63 — 1.23	69 — 1.04	75 — 0.88	81 — 0.72
50 — 1.70	64 — 1.20	70 — 1.00	76 — 0.85	82 — 0.70
55 — 1.50	65 — 1.17	71 — 0.97	77 — 0.82	
60 — 1.32	66 — 1.14	72 — 0.95	78 — 0.80	
61 — 1.29	67 — 1.11	73 — 0.92	79 — 0.77	

contact with the air is increased in both cases. Increasing the air quantity decreases the time the air is in contact with the water, but, since a greater quantity is passing through, the average differential between the water temperature and the wet-bulb temperature of the air is increased, and this speeds up the heat transfer rate. Increased air quantities are obtained only at the expense of increased fan power, which increases approximately as the cube of the air quantity. Air velocities through

mechanical draft towers vary from 250 fpm to 600 fpm over the gross area of the structure.

Mechanical draft water cooling equipment may be set up inside buildings, where it usually draws its air supply from the general space in which it is installed, and discharges its exhaust air through a duct to the outside. Indoor cooling towers may be either of the wood-filled or the spray-filled type. In many cases where little height but considerable area is available, water is cooled in a spray-filled structure similar to an air washer, with the air passing horizontally through the apparatus and being discharged through a duct to the outside. Such apparatus does not have the counter flow advantage of the vertical mechanical draft water cooling equipment, and therefore requires a much larger excess of air for proper operation. Air velocities and operating powers are considerably above those required by vertical mechanical draft water cooling equipment.

Data are given in Table 4 for determining the tower area based upon the water cooling range, and the difference between the air wet-bulb temperature and the cooling water temperature leaving the tower, frequently referred to as the *approach to the wet-bulb*. The values given in this table are based on an air velocity of 300 fpm through the tower.

For a specific installation, manufacturers should be consulted, since the data on water cooling equipment included in this chapter are for general application only.

Example 2 Based on the design data given in Example 1, determine the size of mechanical draft cooling tower to cool the same amount of water if an air velocity of 300 fpm is used through the tower and a spray eliminator is located at the top of the unit.

Solution With an air wet-bulb temperature of 70 F and a leaving water temperature of 80 F, the approach to the wet-bulb is 10 F The cooling range is equal to the water temperature difference of 100 and 80 F or 20 F From Table 4 a value of 0.47 sq ft of tower area per gallon of water is selected for an approach of 10 F and a 20 F cooling range With a 70 F wet-bulb air temperature the correction factor is 1 00

$$1000 \times 0.47 \times 1.0 = 470 \text{ sq ft of tower area}$$

Make-Up Water

Since the atmospheric water cooling equipment performs its functions chiefly by evaporating a portion of the water in order to cool the remainder, there is a continual drain on the quantity of water in the system, and this loss must be replaced. Approximately 1 gal of water is lost for every 1000 gal of water cooled per degree of cooling range; so if 1000 gpm of water are cooled through a 10 F range, 10 gpm of water will be required to replace evaporated water. Replacement supply is usually regulated by a float control valve. Because the evaporation of the water leaves behind the salts which the water contained, high concentration of salts may make chemical treatment of the make-up water necessary to avoid excessive deposits in the condensers.

Winter Freezing

If atmospheric water cooling equipment is operated in freezing weather, the water may be cooled below freezing temperature so ice forms and collects until its weight causes damage. To obviate freezing during continued operation, the efficiency of the apparatus may be lowered. This

Table 5 Comparison of Various Types of Atmospheric Water Cooling Equipment

Figures indicate order of desirability

	Cooling Pond	Spray Pond	Spray Tower	Deck Tower	Mechanical Draft	Indoor Tower
Cost	x	2	1	3	4	5
Area	5	4	3	2	1	x
Height	1	2	3	4–5	4–5	x
Weight per square foot	x	x	1	3	4	2
Independence of wind velocity	6	3	4	5	1–2	1–2
Drift nuisance	1	6	5	4	2–3	2–3
Make-up water required	1	6	5	4	2–3	2–3
Pumping head	1	2	3	4–5	4–5	6
Maintenance	2	1	3	4	5	6
Suitability for congested districts	x	5	4	3	1	2
Water quantity required for definite result	6	5	4	1–2	1–2	3

xNot comparable

is done on the spray pond and the spray cooling tower by reducing the quantity of water fed to the apparatus, thereby lowering the pressure at the nozzles and increasing the size of the drops produced. On the deck tower the upper system may be shut off and a secondary distribution system put in service midway down the height of the tower. The water will be kept above freezing because it will have shorter contact with the air. The mechanical draft tower can be protected by reducing the air flow through the tower, by stopping or reducing the speed of the fans, or by partially closing dampers.

If the system is operated intermittently in freezing weather, water in the basin may freeze and the expansion of the ice may do harm. Freezing during intermittent operation can be prevented only by draining the water basin when it is out of service. On small roof installations, a tank large enough to hold all the water in the system is often installed inside the building and the basin is drained into this by gravity, the pump suction being taken from this inside tank.

A comparison of various types of water cooling equipment is given in Table 5.

PROBLEMS IN PRACTICE

1 ● What performance tests should be given air washers?

a. Capacity, *b* Resistance, *c.* Visible entrainment of free moisture, and *d.* Humidifying or dehumidifying efficiency

2 ● What are different types of air washers?

a. Spray, *b.* Wet scrubber, and *c.* Combination spray and scrubber

3 ● Upon what air velocity are air washers usually rated?

500 fpm through the area above the tank.

4 ● What is the difference between direct and indirect humidification?

Direct humidification signifies that the humidifiers are within the space to be humidified

with distribution produced by the number of humidifiers With direct humidification there is relatively little air movement

Indirect humidification signifies that the air is drawn from the enclosure and passed through the humidifier (air washer) and distributed by means of a duct system

5 ● Where is direct humidification desirable?

Direct humidification is desirable where high humidity is required accompanied with cooling, ventilation, or air motion

6 ● Where is indirect humidification desirable?

Indirect humidification is desirable when high humidity is required with simultaneous removal of heat by ventilation

7 ● Why do cooling towers give best results when the humidity of the air is low?

The cooling of the water by dropping it through the air depends mostly upon the evaporation of the water. If the relative humidity of the air is low, the water vapor will be readily absorbed and carried away, while if the relative humidity is high, its capacity to pick up water vapor is less and the water is cooled less with the same exposure to the air

From Table 5, the final temperature would be approximately 78 2 F

8 ● What are some of the advantages and disadvantageous of a forced draft cooling tower compared with a natural draft wind tower?

Advantages: *a* Does not depend on wind, *b* More adjustable as to space requirements, and *c* Higher efficiency.

Disadvantages *a* Higher first cost, *b* Higher pumping head, and *c*. Higher maintenance cost

9 ● What wet-bulb temperature for outside air is usually selected in air conditioning design when cooling is to be accomplished?

One which is not exceeded more than 5 to 8 per cent of the time in the locality where the plant is situated

10 ● Where should the suction connection be placed in a cooling pond?

As far below the surface as possible and as far away from the discharge as practicable.

11 ● What chemical is used to kill algae formation in spray ponds?

Potassium permanganate.

12 ● What is the usual amount of spray water delivered to a cooling pond per square foot of area?

From 0 1 gpm on small sizes to 0 8 gpm on large sizes.

13 ● About how much water is lost by evaporation in atmospheric cooling?

About 1 gal per 1000 gal for each degree of cooling range.

14 ● How is freezing obviated in cooling pond sprays?

The pressure and quantity of water is lowered so that the drops become larger in size and do not freeze so readily.

Chapter 13

UNIT HEATERS, VENTILATORS, AIR CONDITIONING AND COOLING UNITS

Classification of Unitary Equipment and Related Systems, Unit Heaters, Heating Medium, Estimating Heat Losses, Air Temperatures, Output of Heaters, Direction of Discharge, Boiler Capacity, Quietness, Piping Connections, Unit Ventilators, Split and Combined Systems, Vents, Cooling Units, Air Conditioning Units, Heating, Humidifying and Dehumidification, Filtering, Location of Units, Air Distribution, Residential Central System Units, Capacities, Costs, Accessories to Unitary Equipment

IN other chapters, complete descriptions have been given of heating, cooling, ventilating, humidifying, and dehumidifying systems. These descriptions have covered the detailed principles of each and have, in general, described the assembled equipment included in the complete systems. The success of such completely engineered, heating, cooling, and air conditioning systems has inevitably led to the production of smaller factory-assembled equipment employing a majority of the principles of these complete systems. As a result, present day practice involves the use of this unitary equipment in the majority of smaller installations where capacity demands are within the limits of such units. Thus, unit heaters, unit ventilators, cooling units and air conditioning units have come to occupy a place of their own in the industry.

With the growth of this unitary industry, it becomes increasingly evident that there is no sharp line of demarcation, on the basis of capacity, between a *unit* and a *central station* system. Definitions contained in a code, Standard Method of Rating and Testing Air Conditioning Equipment[1], have helped to clarify and identify the various types available.

A *unit* is a factory-made encased assembly of the functional elements indicated by its name, such as air conditioning unit, room cooling unit, humidifying unit, etc. Such units are shipped substantially complete or built and shipped in sections so that the only field work necessary is the assembling together of the sections, without resorting to any field fabrication. A unit of this type may be complete in itself, employing its own direct means of air distribution and sources of refrigeration or heating, in which case, it thus represents a complete self-contained unit. Or it may be coupled with separate means of air distribution such as duct work and outlets, in which case, it will still be considered as a unit system, in dis-

---

[1]Proposed code prepared by Joint Committee of *American Society of Refrigerating Engineers*, AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, *Refrigerating Machinery Association*, *National Electric Manufacturers Association* and *Air Conditioning Manufacturers' Association*.

tinction to the generally accepted term of a field fabricated central station system. The manufacturer of the unit is responsible for the output and performance of the unit under rated conditions, whereas the contractor installing the complete unitary system is normally held responsible for the complete performance of the system

Unit equipment justifies its existence due to the following features:

1 Lower cost per unit capacity. Standardized design and volume production makes possible low cost factory assembly thereby eliminating individual design and handling of every part for each installation

2 Flexibility and mobility of equipment. Unitary equipment can be readily located in existing buildings without the necessity of running large ducts through floors and many partitions Such equipment can be shifted to meet changing requirements Tenants may obtain the advantages of conditioning when the entire building is not equipped with a conditioning system. In industrial process work, the flexibility of unitary equipment is also advantageous.

3. Lower installation costs. The fact that the equipment arrives on the job in an assembled condition, coupled with the lesser problems of duct work and connecting piping, materially reduces installation costs.

4. Small capacities The small capacities available in unitary equipment have brought the advantages of controlled air conditions to a number of small offices, stores, shops, and individual rooms where specially designed and built central system equipment would have been uneconomic.

This chapter will limit itself in general to a description and discussion of the funtions and construction of the units themselves indicating the accessory features necessary to complete unitary systems.

SUB-DIVISION OF UNITARY EQUIPMENT

For descriptive purposes unitary equipment is sub-divided on a purely functional basis. The following definitions are included in the previously referred to code[2].

1. A *Heating Unit* is a specific air treating combination consisting of means for air circulation and heating within prescribed temperature limits

2. A *Cooling Unit* is a specific air treating combination consisting of means for air circulation and cooling within prescribed temperature limits.

3. A *Humidifying Unit* adds water vapor to and circulates air in a space to be humidified.

4 A *Dehumidifying Unit* removes water from and circulates air in a space to be dehumidified.

5. An *Air Conditioning Unit* is a specific air treating combination consisting of means for ventilation, air circulation, air cleaning and heat transfer with control means for maintaining temperature and humidity within prescribed limits.

6. A *Cooling Air Conditioning Unit* is a specific air treating combination consisting of means for ventilation, air circulation, air cleaning and heat transfer with control means for cooling and maintaining humidity within prescribed limits.

7. A *Heating Air Conditioning Unit* is a specific air treating combination consisting of means for ventilation, air circulation, air cleaning and heat transfer with control means for heating and maintaining humidity within prescribed limits.

8 A *Self-Contained Air Conditioning or Cooling Unit* is one in which a condensing unit is combined in the same cabinet with the other functional elements.

[2]Loc Cit Note 1.

9 A *Free Delivery Type Unit* takes in air and discharges it directly to the space to be treated without external elements which impose air resistance

10 A *Pressure Type Unit* is for use with one or more external elements which impose air resistance

There has grown up in the industry definite branches, in which the engineering and application of the equipment vary quite widely. Thus common acceptance recognizes the following groups of the unitary equipment defined above.

1. *Unit Heaters* consisting of an encased heating surface through which air is forced by means of a fan or blower, located either in or closely adjacent to the heated space, and normally employed only for industrial and commercial applications.

2 *Unit Ventilators* which are similar in principle to unit heaters but are designed to use outside air with or without provision for recirculation of the air While unit heaters are largely used for commercial and industrial applications, unit ventilators are intended primarily for school, offices and semi-commercial applications.

3 *Cooling Units* which are similar to unit heaters except that a cooling medium is used in place of a heating medium and provision is made to collect and remove the condensate Cooling units are normally applied to the cooling of products for their preservation or processing (commercial air conditioning) and air conditioning units are used for cooling for comfort

4 *Air Conditioning Units* which consist of equipment to provide control of heating with humidifying or cooling with dehumidifying, coupled with air circulation, all compactly housed in a single casing.

5 *Miscellaneous Unit Equipment and Accessories* such as filtering equipment, attic fans, humidifying units and special controls

UNIT HEATERS

A *unit heater* consists of the combination of a heating element and fan or blower having a common enclosure and placed within or adjacent to the space to be heated. Generally no ducts are attached to inlets or outlets, although it is common practice with many unit heaters to equip them with directional outlets or adjustable louvers. While unit heaters were first applied to industrial heating, the constant development and improvement of this equipment both as to appearance and quietness has broadened its field of application until today unit heaters are available for fields extending from large factory spaces to recreation rooms, churches, and offices.

While unit heaters are designed primarily to handle all recirculated air, they may be installed to handle either partial or total outdoor air. Compared with the older method of heating by means of radiation, properly designed and applied unit heaters should:

1. Circulate air in the building at a rapid rate but without objectionable draft
2. Reduce the temperature differential between the floor and ceiling.
3. Direct the heated air so that uniform temperature distribution be obtained throughout the heated space.
4. Prevent or remove the cold stratum of air commonly found at the floor level
5. Reduce the number of heating elements required and thereby decrease the cost and extent of the piping necessary.
6. Maintain a closer control of room temperature either manually or by means of simple thermostats
7. Produce an economy in heating costs resulting from the sum total of the above advantages

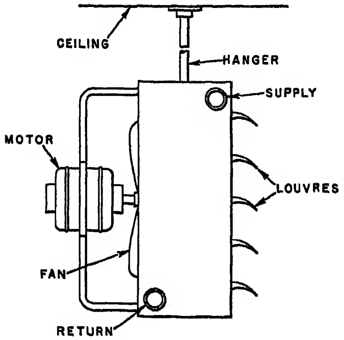


Fig 1 Suspended Unit Heater, Propeller Type Fan

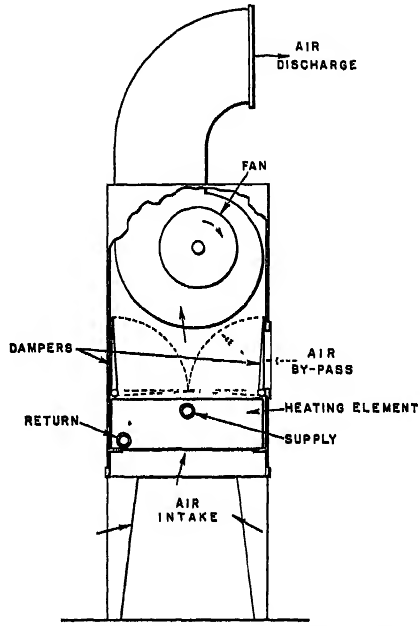


Fig. 2 Floor Mounted Unit Heater, Housed Type Fan

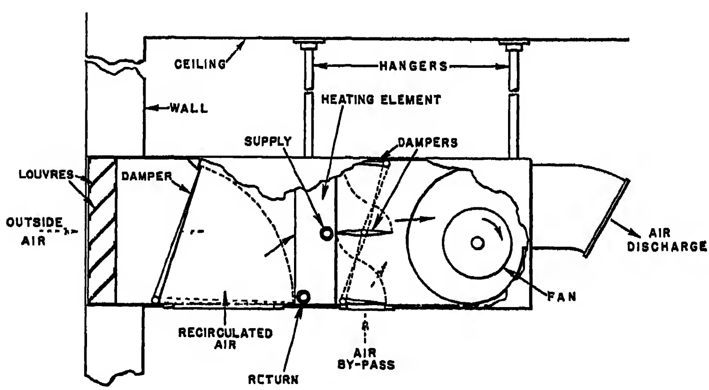


Fig. 3. Suspended Type Unit Heater, Housed Type Fan

TYPES OF UNITS

There are two major types of unit heaters, propeller fan type and centrifugal housed fan type. The housed fan, high velocity (1500 to 2500 fpm) discharge units with outlets adjustable to deliver air in several directions, are able to project their heating effect over distances of from 30 ft to as much as 200 ft from the unit This makes possible the location of these units at considerable distances from each other, thus reducing greatly the piping and loss of floor space due to the heating equipment. Propeller-type units, illustrated in Fig. 1, with outlet velocities of from 300 to 1000 fpm are usually placed from 30 up to 100 ft apart.

Two methods of application of unit heaters are commonly used. Floor mounted units, shown in Fig 2 are available either with or without the air by-pass, and withdraw the cold air from the floor and discharge the heated air above the working zone. Suspended type units are located in an elevated position withdrawing air from this higher level and discharging the heated air down into the working zone. In closely occupied spaces where direct air drafts into the working zone are not permitted, the floor mounted unit will give more uniform temperature distribution. On the other hand, if opportunity is provided to deliver the heated air from suspended units down into the working zone, excellent temperature distribution is possible. A suspended high-velocity type unit heater connected to an outside air intake with damper to control the volume of ventilation is shown in Fig. 3.

A wide variety of structural designs is available. All employ some form of convector, supplied with either steam or hot water, although occasionally equipped for gas or electric heat. Air is always forced over these convectors by a fan of either the propeller or centrifugal type. Heating surfaces may be in the form of steel pipe coils, non-ferrous tubes or pipes with extended surfaces, cast iron, and pressed or built up sections of the cartridge or automotive type.

HEATING MEDIUM

The convectors of unit heaters or ventilators may be supplied with either hot water or steam. When water is used, it should be circulated mechanically and the pump rate and friction loss should be based upon test data for the particular unit to be employed. Normally the friction within the convector is too great for gravity circulation. The heat output of a given heater will normally be less when using water than with steam at the same temperature, unless high rates of circulation are maintained.

Either high or low pressure steam may be used but it is essential that proper venting of the air from the coil be obtained at all times. Where unit heaters are used with outside air in freezing climates, a steam pressure of not less than 5 lb at the coil must be maintained due to the possible danger of freezing of condensate in the tubes. It is essential that properly constructed traps and some form of thermostatic air by-pass be used with high pressure steam as well as adequate condensing legs to prevent steam in the returns. Increasing the return temperature tends to increase return line corrosion, especially at points where overheated condensate or steam enters the return line.

When low pressure steam is used with unit heaters and ventilating units, it is important that proper means be provided for the removal of the heavy condensation. Such units should not be applied to low pressure gravity return systems except where the difference in heater level and boiler water line is large enough to compensate for the pressure loss through the convector at its highest condensation rate. The use of vacuum or return pumps and receivers is advisable with jobs of any considerable size, as the best method of taking care of the condensate and at the same time providing for proper venting of the units directly into a vacuum return line system, or into an open vented return system, the latter having some advantage in preventing the formation of any vacuum in the unit itself, which sometimes tends to hold up condensate and causes freezing.

ESTIMATING HEAT LOSSES

The heat losses of a building to be equipped with unit heaters are determined in the same manner as for any other heating system, excepting so far as the unit heaters may prevent air stratification and thus reduce the temperature difference between the ceiling and floor. (See Chapter 7.)

Unit heaters may be arranged to recirculate the air or to supply warmed air from the outside for ventilation or to make up air exhausted.

If all or a part of the air is to be taken in from out-of-doors, the heat necessary to warm this air from the outside temperature to the inside temperature must be added to the transmission or other losses. Units of the number and size needed to furnish the total heat required are then selected from the manufacturers' rating tables, using these ratings at the steam pressure to be used and at the temperature at which the air will enter the convector.

AIR TEMPERATURES [3]

For recirculating heaters with intakes at the floor level, the temperature to be maintained in the room should be considered as the temperature of the air entering the heater. Where outside air is introduced, the temperature of the mixture must be calculated and used as the entering air temperature to the heater. Where suspended heaters are used without any intake boxes extending down to the floor level, a higher entering air temperature should be used than that at which the room is to be maintained.

With suspended unit heaters taking air at some distance above the floor, the temperature variation from floor to ceiling may reach as much as 1 deg for each foot of elevation during the periods when the maximum capacity of the heaters is required. Thus this allowance should be made in calculating the capacity of suspended heaters. High velocity discharge units (blower type) will maintain slightly lower temperature differences than will low velocity units (propeller type). Unit heaters taking in recirculated air at the floor level should maintain temperature differentials

[3]Temperature Gradient Observations in a Large Heated Space, by G L Larson, D W Nelson, and O. C. Cromer (A S H V E Transactions, Vol 39, 1933)

Tests of Three Heating Systems in an Industrial Type of Building, by G L Larson, D W Nelson, and John James (A S H V E. Journal Section, *Heating, Piping and Air Conditioning*, November, 1934)

of less than 0.5 deg per foot of elevation when the maximum capacity of the heaters is required. This temperature difference per foot of elevation is less than the corresponding variations for spaces heated by direct radiation.

Unit heaters save fuel because of their ability to circulate air at a lower average temperature than the air circulated by direct radiators; however, the unit heaters must circulate more air in any given time than is needed with direct radiators. This requires the selection of heaters having a liberal air capacity for the required heat output, which in turn means a relatively low final temperature. Extremely low final temperatures can be had only at the expense of larger heaters and increased power, so that an economic limit is imposed. In general, for heating purposes it is advisable to use a delivery temperature not more than 70 F above the average room temperature desired, and considerably less where possible. Since the delivery temperature increases with increase in steam pressure with high pressure steam, units should be selected that have a minimum heating surface in order to provide outlet air not in excess of 70 F above average room temperature.

OUTPUT OF HEATERS

It is standard practice to rate unit heaters in Btu per hour at a given temperature of air entering the heater and at a given steam pressure maintained in the coil. Steam at 2 lb pressure and air entering at 60 F are used as the standard basis of rating[4]. The capacity of a heater increases as the steam pressure increases, and decreases as the entering air temperature increases. The heat capacity for any condition of steam pressure and entering air temperature may be calculated approximately from any given rating by the use of factors in Tables 1 and 2. Table 1 is for blow-through and Table 2 is for draw-through unit heaters. These tables are accurate within 5 per cent.

Unit heaters are customarily rated as free delivery type units. If outside air intakes, filters, or ducts on the discharge side are used with the heater, proper consideration should be given to the reduction in air and heat capacity that will result because of this added resistance.

The percentage of this reduction in capacity will depend upon the characteristics of the heater and on the type, design, and speed of the fans employed, so that no specific percentage of reduction can be assigned for all heaters for a given added resistance. In general, however, disc or propeller fan units will have a larger reduction in capacity than housed fan units for a given added resistance, and a given heater will have a larger reduction in capacity as the fan speed is lowered. When confronted with this problem the ratings under the conditions expected should be secured from the manufacturer.

When steam supplied to the heaters contains superheat, the capacity of the heater will be but slightly less than with saturated steam at the same pressure. Recent tests indicate that the reduction of capacity

[4]A.S.H.V.E. Standard Code for Testing and Rating Steam Unit Heaters (A.S.H.V.E. TRANSACTIONS, Vol 36, 1930)

Table 1. Constants for Determining the Capacity of Blow-Through Type Unit Heaters for Various Steam Pressures and Temperatures of Entering Air

(Based on Steam Pressure of 2-lb Gage and Entering Air Temperature of 60 F)

Steam Pressure Lb per Sq In	Temperature of Entering Air											
	−10°	0°	10°	20°	30°	40°	50°	60°	70°	80°	90°	100°
0	1.538	1 446	1.369	1.273	1.191	1.110	1.034	0 956	0.881	0 809	0.739	0.671
2	1 585	1 495	1.405	1.320	1 237	1.155	1.078	**1.000**	0 926	0.853	0 782	0.713
5	1 640	1 550	1.456	1.370	1 289	1 206	1 127	1.050	0 974	0 901	0 829	0.760
10	1 730	1 639	1 545	1.460	1 375	1.290	1.211	1.131	1.056	0 982	0.908	0.838
15	1 799	1 708	1.614	1 525	1 441	1.335	1 275	1.194	1.117	1 043	0 970	0 897
20	1 861	1 769	1.675	1 584	1 498	1 416	1.333	1 251	1.174	1 097	1 024	0 952
30	1.966	1 871	1 775	1.684	1 597	1 509	1.429	1.346	1.266	1 190	1 115	1 042
40	2.058	1.959	1 862	1 771	1.683	1.596	1 511	1 430	1 349	1 270	1 194	1 119
50	2 134	2.035	1 936	1.845	1.755	1 666	1.582	1.498	1 416	1 338	1 262	1 187
60	2 196	2 094	1 997	1.902	1.811	1.725	1 640	1 555	1 472	1 393	1 314	1 239
70	2 256	2.157	2 057	1 961	1 872	1 782	1 696	1 610	1.527	1 447	1.368	1 293
75	2 283	2 183	2 085	1 990	1 896	1 808	1 721	1.635	1 552	1 472	1 392	1 316
80	2 312	2 211	2 112	2 015	1.925	1.836	1.748	1 660	1 577	1 497	1 418	1 342
90	2 361	2 258	2 159	2.063	1.968	1 880	1 792	1.705	1 621	1 541	1.461	1 383
100	2 409	2.307	2 204	2 108	2 015	1 927	1 836	1 749	1 663	1 581	1 502	1 424

Note—To determine capacity at any steam pressure and entering temperature, multiply constant from table by rated capacity at 60 F entering and 2 lb pressure

TABLE 2. CONSTANTS FOR DETERMINING THE CAPACITY OF DRAW-THROUGH TYPE UNIT HEATERS FOR VARIOUS STEAM PRESSURES AND TEMPERATURES OF ENTERING AIR

(Based on Steam Pressure of 2-lb Gage and Entering Air Temperature of 60 F)

STEAM PRESSURE LB PER SQ IN	TEMPERATURE OF AIR ENTERING HEATER											
	−10°	0°	10°	20°	30°	40°	50°	60°	70°	80°	90°	100°
0	1.483	1.405	1.329	1.253	1.178	1.105	1.032	0.962	0.892	0.822	0.754	0.688
2	1.520	1.442	1.363	1.290	1.215	1.141	1.069	**1.000**	0.930	0.861	0.792	0.728
5	1.565	1.485	1.410	1.334	1.260	1.187	1.114	1.045	0.975	0.906	0.838	0.771
10	1.637	1.558	1.480	1.403	1.328	1.253	1.182	1.112	1.042	0.973	0.903	0.838
15	1.688	1.610	1.533	1.458	1.382	1.310	1.239	1.168	1.099	1.028	0.960	0.895
20	1.728	1.649	1.572	1.498	1.421	1.350	1.278	1.208	1.138	1.070	1.002	0.936
30	1.803	1.725	1.648	1.572	1.497	1.423	1.352	1.281	1.212	1.145	1.078	1.010
40	1.864	1.787	1.710	1.637	1.563	1.491	1.420	1.350	1.282	1.215	1.148	1.081
50	1.927	1.850	1.773	1.700	1.628	1.554	1.483	1.416	1.347	1.278	1.211	1.145
60	1.973	1.897	1.820	1.748	1.673	1.601	1.531	1.463	1.394	1.325	1.260	1.194
70	2.018	1.943	1.869	1.795	1.722	1.651	1.582	1.512	1.443	1.377	1.310	1.243
75	2.043	1.970	1.895	1.822	1.750	1.680	1.609	1.540	1.471	1.402	1.333	1.268
80	2.064	1.988	1.914	1.841	1.770	1.698	1.629	1.560	1.491	1.422	1.354	1.288
90	2.102	2.028	1.951	1.878	1.804	1.732	1.661	1.590	1.523	1.457	1.387	1.321
100	2.150	2.071	1.994	1.919	1.845	1.770	1.700	1.630	1.560	1.492	1.425	1.359

Note.—To determine capacity at any steam pressure and entering temperature, multiply constant from table by rated capacity at 60 F entering and 2 lb pressure

from this cause is negligible for superheat up to 50 deg and will not exceed 3½ per cent for any degree of superheat.

DIRECTION OF DISCHARGE

Heaters may be distributed through the central portions of a room discharging toward exposed surfaces, or may be spaced around the walls, discharging along the walls and inward as well, especially when there are considerable roof losses.

In general, it is better to direct the discharge from the unit heaters in such fashion that rotational circulation of the entire room content is set up by the system rather than to have the heaters discharge at random and in counter-directions.

Various types and makes of unit heaters are illustrated in the *Catalog Section* of this edition. Usually hot blasts of air in working zones are objectionable, so heaters mounted on the floor should have their discharge outlets above the head line and suspended heaters should be placed in such manner and turned in such direction that the heated air stream will not be objectionable in the working zone. In the interest of economy, however, the elevation of the heater outlet and the direction of discharge should be so arranged that the heated air shall be brought as close to the head line as possible, yet not into the working zone. In general, the higher the elevation of the unit, the greater the volume and velocity required to bring the warm air down to the working zone, and consequently, the lower the required temperature of the air leaving the unit.

BOILER CAPACITY

The capacity of the boiler should be based on the rated capacity of the heaters at the lowest entering air temperature that will occur, plus an allowance for line losses. Ordinarily for recirculating heaters the lowest entering temperature will occur at the beginning of the heating period and is usually taken as 40 F, while for ventilators taking air from outdoors the lowest entering temperature will be the extreme outdoor temperature expected in the district. No greater allowance in boiler capacity beyond the calculated heat demand need be added in order to supply unit heaters than for any other type of system.

It is unwise to install a single unit heater as the sole load on any boiler, particularly if the unit heater motor is started and stopped by thermostatic control. The wide and sudden fluctuations of load that occur under such conditions would require closer attendance to the boiler than is usually possible in a small installation. Where oil or gas is used to fire the boiler, it is possible by means of a pressurestat to control the boiler, in response to this rapid fluctuation. In most cases, however, and particularly where the boiler is coal-fired, it is advisable to use two or more smaller heating units instead of one large unit.

Steam pressures below 5 lb can be used with safety for recirculating unit heaters when their coils are designed for the purpose and when proper provision is made for returning the condensate. If ventilators are to take in air that may be at a temperature below freezing, however, a steam pressure of not less than 5 lb should be maintained on the convector

or a corresponding differential in pressure between the supply and returns be maintained by means of a vacuum.

QUIETNESS

Fan speed alone is not a measure of relative quietness of fans having different designs and proportions. Quietness is a function of type, diameter, blade form and other variables besides speed, and all these must be considered. In general for a given design, the higher the fan speed, the greater the noise, and centrifugal fans are more quiet than disc or propeller fans.

PIPING CONNECTIONS

Piping connections for unit heaters are similar to those for other types of fan-blast heaters. Typical connections are shown in Figs. 4 and 5. One-pipe gravity and vapor systems are not recommended for unit heater

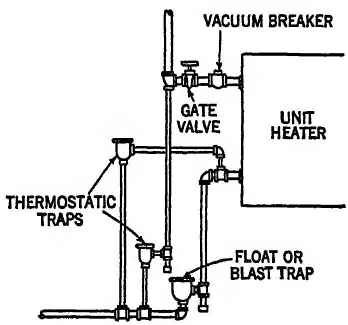


Fig 4. Unit Heater Connections Where Condensation Is Returned to Vacuum Pump or to an Open Vented Receiver

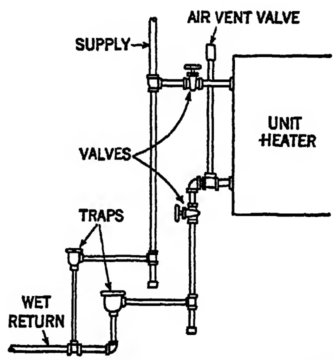


Fig 5 Unit Heater Connections Where Condensation Is Returned to Boiler Through Wet Return

work. For two-pipe closed gravity return systems the return from each unit should be fitted with a heavy-duty or blast trap, and an automatic air valve should be connected into the return header of each unit. Pressure-drop must be compensated for by elevation of the heater above the water line of the boiler or of the receiver.

In pump and receiver systems the air may be eliminated by individual air valves on the heaters, or it may be carried into the returns the same as for vacuum systems and the entire return system be free-vented to the atmosphere, provided all units, drip points, and radiation are properly trapped to prevent steam entering the returns.

On vacuum or open vented systems the return from each unit should be fitted with a large capacity trap to discharge the water of condensation and with a thermostatic air valve for eliminating the air, or with a heavy-duty trap for handling both the condensation and the air, provided the air finally can be eliminated at some other point in the return system.

For high pressure systems the same kind of traps may be used as with

vacuum systems, except that they must be constructed for the pressure used. If the air is to be eliminated at the return header of the unit, a high pressure air valve can be used; otherwise the air may be passed with the condensate through the high-pressure return trap, with some danger of return pipe corrosion and the problem of its elimination at some other point in the system.

The connections for steam and return piping to unit heaters must always be calculated on the basis of the high heat emission or condensation rate of such devices. The pipe-size tables given in Chapter 32 may be used for unit heater work by multiplying EDR values by 240 to get Btu values.

OTHER TYPES OF UNITS

All Electric

The foregoing discussion relates generally to units in which steam or hot water is used as the heating medium. On rare occasions electrical resistances are used as the heating element. These are applied only where electric power is abundant and cheap and where other forms of fuel are scarce and expensive. (See Chapter 39.)

Direct Fired

A recent development in gas burning equipment is the direct-fired industrial unit heater. These heaters are of the warm-air type and are equipped with fans which cause the air to pass over the heating surfaces at a fairly high velocity and then direct the warm air in to the space to be heated. As is the case with the steam-fed unit heaters, the gas-fired appliances may be used for heating stores, shops, and warehouses. They usually are suspended in the space to be heated and in most instances leave the entire floor and wall area free for commercial use. Partial or complete automatic control also may be secured on appliances of this type. This type of heater is often used for temporary heat during building construction or where the installation of a steam or hot water plant is for some reason not justified. For permanent installations, it is usually advisable to provide an exhaust duct from the gas-fired unit heaters to remove products of combustion from the occupied space. While this is not necessary in large open industrial plants, in smaller closed rooms, it becomes essential.

Turbine Driven

Where high pressure steam is available it is sometimes used to drive a steam turbine direct-connected to the unit heater. The exhaust from this turbine, reduced in pressure, is then passed into the heating coil where it is condensed and returned to the boiler.

INDUSTRIAL USES

In addition to their prime function of heating buildings, unit heaters may be adapted to a number of industrial processes, such as drying and curing, with which the use of heated air in rapid circulation with uniform distribution is of particular advantage. They may be used for moisture absorption, such as fog removal in dye-houses, or for the pre-

vention of condensation on ceilings or other cold surfaces of buildings in which process moisture is given off. When such conditions are severe, it is necessary that the heaters draw air from outside in enough volume to provide a rapid air change and that they operate in conjunction with ventilators or fans for exhausting the moisture-laden air. (See discussion of condensation in Chapter 7.)

UNIT VENTILATORS [5]

Unit ventilators while designed primarily for ventilation must incorporate controlled heating. Since these units are generally located in the occupied space, they must be pleasing in design and must harmonize

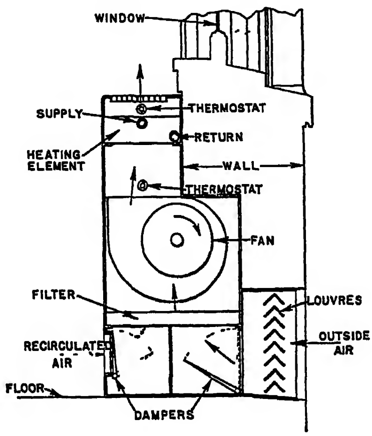


FIG 6. TYPICAL UNIT VENTILATOR SHOWING ONE OF MANY ARRANGEMENTS OF DAMPERS AND HEATING COILS

with the furniture or with the decorative scheme. A typical unit ventilator is illustrated in Fig. 6. They usually consist of a semi-decorative cabinet containing the following necessary or optional parts:

1. Outside air inlet.
2. Inlet damper for closing the opening to the outside air inlet when the unit is not in use.
3 Adhesive or dry type filters for cleaning the air (optional)
4 A heating element usually of special design and intended for low pressure steam
5. Motor and fan assembly.
6. Mixing chamber where warm and cold air streams are brought together (No mixing chamber is normally provided where sectional type convectors are used)
7 Outdoor air inlet and recirculating air mixing damper (optional).
8. Device for ozonizing air (optional).
9. Discharge grille or diffuser.
10. Temperature control arrangement

[5]A roof ventilator is sometimes termed a *unit ventilator*. For information on roof ventilators, see Chapter 4

The primary functions of a unit ventilator are:

1. To supply a given quantity of outdoor air for ventilation or to mix indoor and outdoor air.

2. To warm the air to approximately the room temperature if the unit is intended for ventilation only, or to a higher temperature if it is intended to take care of all or a part of the heat transmission losses from the room.

3. To control the temperature of the air delivered so as to prevent both cold drafts and overheating.

4. To deliver air to the room in such a manner that proper distribution is obtained without drafts.

5 To recirculate room air for the purpose of heating or promoting comfort when ventilation is unnecessary

6. To perform all its functions without objectionable noise.

In addition to these functions, unit ventilators frequently are arranged so that the air supplied may be cleaned by means of filters of either the dry or viscous type. If filters are used, the proper allowance must be made for the increased resistance offered to the air flow. Humidifiers in unit ventilators are rather difficult to control and are only furnished upon special order.

1. *Air Supply for Ventilation.* The outdoor air supply for ventilation is delivered by motor-driven fans operated at comparatively low speeds, the back of the cabinet being connected to the outside through rust-proof louvers and screens. Air quantities may be estimated on the basis of data given in Chapter 3. (See A.S.H.V.E. Ventilation Standards.)

2. *Warming Incoming Air.* The air is heated by passing it through specially designed convectors. The amount of heating surface to be provided in the unit is determined by the volume of air to be heated and the temperature range. If the unit is to be used for supplying air for ventilation only, the convector must be sufficient in capacity to maintain a final air temperature of about 70 F. If the unit is to be used for heating as well as for ventilation, the convector must be sufficient to maintain the necessary final air temperature for the conditions involved.

3. *Control of Temperature.* This is accomplished by varying the temperature of the air discharged from the unit (1) by the automatic operation of a mixing damper which controls the relative quantities of air being blown through the heating unit or by-passed around it, (2) by operation of valves on different layers of convector surfaces, or (3) by variation in the temperature of the circulating heating medium.

The outside air inlet damper and recirculating damper (where one is provided) should be so connected that there will be an uninterrupted supply of air to the fans at all times the unit is in operation. These dampers may be operated by hand or by pneumatic or electric motors manually controlled from some central point.

These dampers may also be linked together, in the form of mixing dampers and be controlled by a thermostat in the cold air intake, by a differential thermostat acted upon by both the cold air and the recirculated air, or by a thermostat in the two streams of air after they are mixed, so as to keep the relative proportion of air taken in from out-of-doors commensurate with outside temperatures and to prevent drafts of cold air being blown through the unit into the room.

Provision should be made for the inlet damper to close automatically whenever the fans are shut down, and not to open until the room is properly heated when the fans are again started. The minimum temperature of the air delivered by the machine should be regulated automatically by a thermostat in the outlet air which controls the temperature of the heated convector, or this minimum temperature may be maintained by properly mixing the inside and outside air by means of the mixing dampers under thermostatic control referred to above. Another thermostat in the recirculated air intake to the unit or elsewhere in the room controls by-pass dampers or the supply of heating medium, or both, so as to control the temperature of the air leaving the unit according to the heat requirements of the room. In addition to these thermostats, a room thermostat is needed to control any other heat sources for the room.

Thermostats for controlling by-pass dampers must be of the intermediate type to hold the dampers in intermediate positions to prevent objectionable drafts. When direct radiators are used in conjunction with unit ventilators, the control is usually arranged so as automatically to open the valves to the direct radiators when the room temperature falls about 2 deg below the setting of the thermostat for the unit ventilator. Another arrangement opens the radiator valve whenever the unit ventilator control reaches the full heating position. Further information on this subject is contained in Chapter 14.

4. *Distribution.* This function is governed by the proper selection and location of the unit. Diffusion and distribution are dependent upon a relatively high velocity air stream discharged in a generally vertical direction, and in order to insure satisfactory diffusion in the room the less the difference between the temperature of the air discharged from the unit and that of the room air, the better. With a final temperature above 110 F, excessive stratification of the air may be experienced. Troublesome drafts may be eliminated to a large extent if a static pressure is built up in the room.

5. *Recirculation of air* requires less fuel than does the use of all outside air and aids in heating up quickly. Certain units are designed to recirculate all air at all times, except when the admission of outside air is needed to regulate room temperatures. Under this arrangement, the outside air for ventilating purposes is obtained solely from infiltration, but the amount thus obtained may or may not be sufficient to meet legal ventilating requirements for public buildings. Recirculation of the air in schools is therefore prohibited by ordinance in many communities. Ventilating systems in schools should be arranged for taking in a sufficient quantity of air to constitute, with infiltration, not less than 10 cfm per occupant of a room.

6. *Quiet Operation.* Since the unit ventilator is generally set in close proximity to the room occupants, it must operate with exceeding quietness.

SPLIT AND COMBINED SYSTEMS

In a *split* system the unit is used primarily for ventilation. Air is delivered to the room at very near the room temperature, and enough separate direct heaters are placed in the room to warm it to the desired

temperature, independently of the unit. Their principal advantage lies in offsetting the cooling effect of window and wall surfaces long before these can be heated to room temperature and in retaining heat for this purpose after the ventilation is shut down.

Where the unit ventilator selected has a capacity more than sufficient to warm the air needed to meet the ventilating requirements, a corresponding reduction may be made in the amount of direct heating surface installed. The greater the amount of excess capacity of the unit, the more efficient will be the temperature regulation of the room. The split system permits the heating of the room during failure of electric current, since the direct radiators will furnish heat, but it permits a careless operator to avoid operating the ventilating equipment.

A combined system employs the unit ventilator alone, its capacity being sufficient both for ventilation and for supplying the heat loss. Direct heating surface is omitted altogether. It becomes necessary then that the fan be running whenever the room is to be heated but this also gives assurance of ventilation, especially if automatic dampers are used in the air intake from out-of-doors and in the recirculating intake arranged so as to give a certain quantity of air from the outside (commensurate with weather conditions) whenever the unit is operating and after the room is heated. The cost of installation of a combined system is usually less than that of a split system and there is less danger of overheating, but if the electric energy fails there will be practically no heating.

LOCATION OF UNIT

The location of the unit ventilator in a room is important. Wherever possible it should be placed against an outside wall. It is difficult to obtain proper air distribution if the unit is erected either on an inside wall or in a corner of the room. Standard units discharge the air stream upward, but for special cases units may be installed to discharge air horizontally. Units may be set away from the wall or partially recessed into the wall to save space without materially affecting the results. The air inlet may enter the cabinet at the back at any point from top to bottom.

VENTS [6]

The size and location of the vent outlet is important. In many cases the sizes for public buildings are regulated by law, but the location of the vents generally is left to the discretion of the engineer.

Best results have been obtained with a velocity through the vent openings nearly equal to that at which the air is introduced into the room, thus maintaining a slight pressure in the room. Calculated velocities at the vent openings of from 600 to 800 fpm produce the best diffusion results from this system.

The cross-sectional area of the vent flue itself may be figured on the basis of 15 sq in. of flue for each 100 cfm. Thus the vent flue area of a flue for a room equipped with one 1200 cfm unit ventilating machine

[6]Investigation of Air Outlets in Class Room Ventilation, by G L Larson, D. W Nelson, and R W Kubasta (A S H V.E Transactions, Vol 38, 1932)

Air Supply to Classrooms in Relation to Vent Flue Openings, by F C Houghten, Carl Gutberlet, and M F Lichtenfels (A S H V E Journal Section, *Heating, Piping and Air Conditioning*, June, 1935)

would be 180 sq in. The area of vent flue opening from the room may be figured on the basis of 25 sq in. per 100 cfm.

In school buildings provided with wardrobes or cloakrooms the vents may be so located that the air shall pass through these spaces, heating and ventilating them with air which otherwise would be passed to the outside without being used to the best advantage. Many state codes for ventilation of public buildings make this arrangement mandatory.

There has been much controversy over the use of corridor ventilation in school building practice, one group holding the view that when each classroom has a separate vent flue there is a minimum fire risk and less likelihood of cross-contamination, while others emphasize the economy features of the corridor discharge and minimize the fire, contamination, and other hazards.

CAPACITIES

Unit ventilators are available in air capacities ranging from 450 cfm to 5000 cfm and with corresponding heat capacities (above that required for ventilation purposes based upon an outside temperature of zero and an inside temperature of 70 F) ranging from 30 Mbh to 144 Mbh (1 Mbh = 1000 Btu per hour). Some manufacturers furnish a unit with several heating capacities for each air capacity, thus enabling the engineer to select the unit best adapted to the heating and ventilating load. Typical capacities are given in Table 3[7].

The amount of heat to be supplied by the unit ventilator will depend on the amount of air passed through the unit and the temperature range through which the air is heated. The weight of air (W) to be circulated per hour is fixed by the ventilating requirements.

If no direct heating surface (radiation) is installed, the combined heating and ventilating requirements must be taken care of by the unit ventilators, and the total heat to be supplied is obtained by means of the following formulæ:

When all of the air handled by the unit is taken from the outside,

$$H_t = 0.24\, W\, (t_y - t_o) \tag{1}$$

$$W = d\, 60\, Q \tag{2}$$

$$t_y = \frac{H}{0.24W} + t \tag{3}$$

where

d = density of air, pounds per cubic foot.
H = heat loss of room, Btu per hour.
H_v = heat requred to warm air for ventilation, Btu per hour.
H_t = total heat requirements for both heating and ventilation, Btu per hour $= H + H_v$.
Q = volume of air handled by the ventilating equipment, cubic feet per minute.
t = temperature to be maintained in the room.
t_o = outside temperature.
t_y = temperature of the air leaving the unit.

[7]A S H V E Standard Code for Testing and Rating Steam Unit Ventilators (A S H V E TRANSACTIONS, Vol 38, 1932).

W = weight of air circulated, pounds per hour.
0.24 = specific heat of air at constant pressure.

From Equations 1, 2 and 3:

$$H_t = H + 0.24\, d\, 60\, Q\, (t - t_o) \qquad (4)$$

Example 1. The heat loss of a certain room is 24,000 Btu per hour, and the ventilating requirements are 1000 cfm If the room temperature is to be 70 F and all air is taken from the outside at zero, what will be the total heat demand on the unit if it is required to provide for both the heating and ventilating requirements (combined system)?

Solution $H = 24{,}000$, $d = 0.075$ $Q = 1000$ cfm, $t = 70$ F, $t_o = 0$ F

Substituting in Equation 4.

$$H_t = 24{,}000 + 0.24 \times 0.075 \times 60 \times 1000\,(70-0) = 99{,}600 \text{ Btu}$$

$$t_y = \frac{24{,}000}{0.24 \times 0.075 \times 60 \times 1000} + 70 = 92.2 \text{ F}$$

When part of the air handled by the unit is taken from the room and the remainder from the outside,

$$H_t = 0.24 W_o\,(t_y - t_o) + 0.24\, W_1\,(t_y - t) \qquad (5)$$

where

W_o = weight of air, pounds per hour taken from out-of-doors
W_1 = weight of air, pounds per hour taken from the room

$$W_o = d_o\, 60\, Q_o \qquad (6)$$

$$W_1 = d_1\, 60\, Q_1 \qquad (7)$$

where

d_o = density of air, pounds per cubic foot at temperature t_o
d_1 = density of air, pounds per cubic foot at temperature t
Q_o = volume of air taken in from the outside, cubic feet per minute
Q_1 = volume of air taken in from the room, cubic feet per minute

$$t_y = \frac{H}{0.24\,(W_o + W_1)} + t \qquad (8)$$

$$H_t = H + 0.24\, d_o\, 60\, Q_o\,(t - t_o) \qquad (9)$$

Equations 5, 6, 7, 8, and 9 may be used in the same manner as is illustrated above for Equations 1, 2, 3, and 4. It may be noted in Equation 9, representing the total heat requirements, that as the quantity Q_o is diminished the heat requirements for the unit diminish very materially.

In Example 1, if the quantity of air taken in from the outside is reduced to zero, or all of the air handled by the unit is recirculated, the total heat requirements H_t reduce from 99,600 Btu to 24,000 Btu, or to about one fourth. Such a unit handling one third of its air volume from the outside and two thirds from the room would show a total heat requirement of $24{,}000 + \frac{99{,}600 - 24{,}000}{3} = 59{,}200$ Btu. Units designed and operated

Table 3 Typical Capacities of Unit Ventilators for an Entering Air Temperature of Zero

Cubic Feet of Air per Minute	Total Capacity in Square Feet of Equivalent Direct Heating Surface (Radiation)		Capacity Available for Heating the Room in Square Feet of Equivalent Direct Heating Surface (Radiation)		Final Air Temperature (Deg Fahr)
	EDR	Mbh	EDR	Mbh	
600	285	68	95	23	105
750	350	84	115	28	105
1000	455	110	150	36	105
1200	565	136	190	46	105
1500	705	169	235	56	105

on this principle show an average heat requirement and, therefore, a boiler capacity requirement of less than 50 per cent of that required for units taking all their air from the outside.

If all of the air is recirculated, the total heat required is the same as the heat loss of the room, or

$$H_t = H = 0.24\ W\ (t_y - t) \tag{10}$$

If the heat loss of the room is to be taken care of by the direct heating surface, the unit ventilators will be required to warm the air introduced for the ventilating requirements. Therefore:

$$H_v = 0.24\ W\ (t_y - t_o) \tag{11}$$

In this case t_y should be equal to or slightly higher than t. If the unit ventilator were of such capacity as to exactly provide for the ventilating requirements, the direct radiation would be selected on the usual basis. However, it is necessary to employ a unit which may not exactly meet the ventilating requirements, since standard units are usually rated in terms of the volume of air that will be delivered at a certain temperature t_y for an initial temperature of t_o. Therefore a certain amount of heat (H_h) may be available from the unit ventilator for heating purposes, as previously stated, and the amount of equivalent direct heating surface may, if desired, be deducted from the amount required for heating the room.

COOLING UNITS

Cooling units as applied to industrial product conditioning and processing are similar in construction to unit heaters except that the heat transfer surface is supplied with refrigeration instead of with steam or hot water. They are normally installed within the space to be served, or at least closely adjacent thereto. Occasionally they are provided to receive outside air in which case this air is invariably filtered or washed to prevent any possible contamination of the product.

Cooling units are provided in two major types similar to unit heaters, either floor mounted with housed fan, or suspended with propeller type fans. Normally, air outlet velocities are lower than for heating, due largely to the effect of high velocities on the product. Cooling units are

normally of the free delivery type although they occasionally are supplemented with ductwork to provide more careful air distribution.

Product cooling originally was accomplished by means of stationary pipe coils. This was later supplemented with the forced fan bunker systems in which air was passed over banks of coils. The present trend in this field is toward a more accurate control of both temperature and humidity, thus placing these units in the classification of complete air conditioning units as discussed in the next section However, in the majority of these cases dry-bulb temperature is controlled separately from the control of humidity, thus classifying these units as cooling units.

The principal field for cooling units is in cold storage plants, fur storage, fruit packing houses, provision stores, brewery fermentation and stock rooms, candy plants, and other industrial process work In replacing bunker and wall coils in meat storage plants, cooling units give distinct

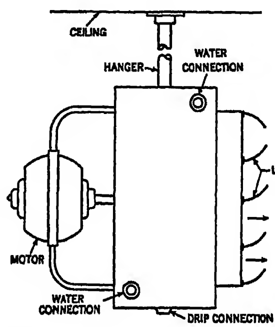


Fig. 7. Ceiling Type Cooling Unit

advantages in compactness, lower first cost and maintenance expense, ease of defrosting, freedom from drip and the maintenance of sanitary conditions, as well as uniform temperature and humidity under variable load conditions. Cooling units by means of their positive air circulation prevent dead-air spots, frequently objectionable in this industry.

Typical cooling units are shown in Figs 7 and 8. The former indicates a suspended type cooling unit which may be designed with or without a moisture eliminator. If high air velocities are maintained, an eliminator will be necessary to prevent the drops of moisture from being carried through with the air. The condensation that occurs is collected in a drip pan and removed from the system through a drain pipe. Fig. 8 indicates a typical floor-mounted unit of the housed fan type. The illustration shows a common form of distributing outlet designed to give low outlet velocities together with a controlled distribution. In process work, it is often important that direct air distribution does not impinge on the product.

Cooling units are normally constructed of galvanized steel or nonferrous material in order to reduce the corrosive effect of their constant wetted condition.

Cooling units are often called upon to operate in rooms where a temperature below freezing is maintained and low refrigerant temperatures are

required. This results in the collection of frost on the heat transfer surface which in turn leads to a rapid loss in capacity and requires eventual defrosting. Such defrosting is accomplished by the following methods:

1. When the room is above freezing the source of refrigeration is cut off and the fan allowed to operate until the unit has defrosted

2 A reversal of the refrigeration system may be provided and the so-called hot gas defrosting method used This is accomplished by reversing the flow of the hot gas so that it is delivered directly from the compressor to the evaporator of the cooling unit As soon as the ice and frost has been melted, the system is again returned to its normal cycle

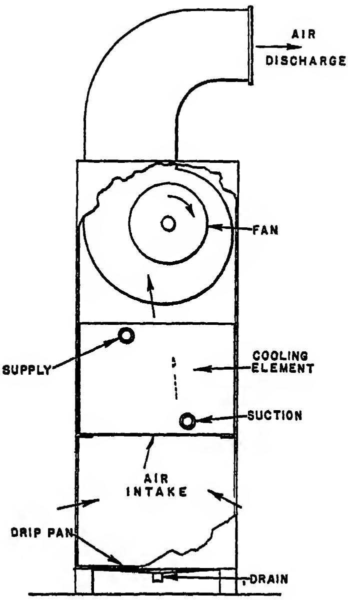


FIG. 8. SURFACE TYPE COOLING UNIT

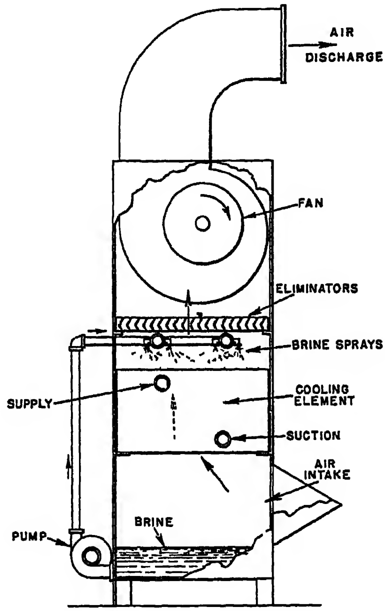


FIG. 9 BRINE SPRAY TYPE COOLING UNIT

3 Where brine is used as a refrigerant, heated brine may be sent through the cooler to remove the ice

4 When the room is at very low temperatures, warm air defrosting is sometimes used by providing for the admission and removal of warm air from outside the cooled space

5. The surface may be sprayed with a strong brine solution

In order to prevent the collection of frost in low temperature rooms where high latent heat loads are present, unit coolers equipped with a constant brine spray are frequently used. These are normally of the housed fan type similar to Fig. 9, but equipped with a pump for recirculating brine over the coil. It is, of course, necessary to strengthen the brine at intervals to maintain a non-freezing mixture.

Ratings of cooling units may be expressed in Btu per hour, or in tons of refrigeration and should specify the quantity, temperature and humidity

of the air entering the unit with a stipulated refrigerant temperature within the coil. When chilled water or brine is used, the rate of circulation of the cooling media as well as its entering temperature must be given.

AIR CONDITIONING UNITS

Air conditioning with unit equipment has gained in popularity during the last few years and this type of apparatus now represents the bulk of the production of the industry. It is to be noted that some equipment does not fulfill all of the basic requirements of a true air conditioning unit. True air conditioning equipment involves not only the ability to alter temperature and humidity conditions within the conditioned space, but it must also be able to *control* these conditions.

The means for accomplishing these functions are outlined herewith:

Heating

The normal air conditioning unit derives its heating function from a heating coil, usually of the non-ferrous finned tube type supplied with either steam or hot water. Steam may be supplied directly from a self-contained and built-in oil or gas fired unit, from a separate domestic steam boiler, or even from an outside source of a central heating plant. Hot water is supplied either from a separate hot water boiler or in rare instances from a domestic water heater

In domestic or household conditioning units, adapted from a warm-air heater to which humidification is added, or possibly all-year-round conditioning, a direct-fired air interchanger is frequently used. The source of heat in this case may be from the combustion of coal, oil, or gas. A wide variety of designs and structures are used. In such direct-fired systems, the bulk and volume of the heat transfer surface is necessarily large in proportion to the rest of the equipment.

Where electric power is low in cost, electric heat has been furnished for air conditioning units either in the form of encased heaters, or open wire heaters. (See Chapter 39.) Radiant electric heaters are seldom used except as their radiant heat is absorbed by some receiving wall and there transmitted to the air in the form of convected heat.

Another method of applying electric heat is by means of the reversed refrigeration cycle, whereby electric energy is used to compress a refrigerant and to deliver the heat of compression, withdrawn from a lower temperature source, to the conditioned space by locating the condensing coils in the air circulation circuit of the air conditioning unit. While this method of heating has gained wide interest, it is practical only in a limited number of applications. Its economic practicability depends largely upon the availability of a source of heat during the winter season at as high a temperature as is possible. Thus, while it is frequently applicable in mild climates, it is not of great value in the northern climates where subfreezing temperatures are common. It has the other unfortunate property that when the outdoor temperature is the lowest and the heat requirement the greatest, the heating capacity of the system is a minimum. This means that in many installations, an auxiliary and separate source of heat must be supplied to carry through these peak winter conditions. Such

factors as these have limited a more general application of this principle. (See Chapter 2 on Refrigeration.)

Humidifying

A variety of methods have been used to furnish humidification in winter to air conditioning units. The oldest and best known is by means of a direct spray which is used in many different ways. The simplest system is where the spray water is furnished from a constant water source, such as city water, and is permitted to run to waste Under such conditions, the spray may be either of the direct atomizing type where, by means of the nozzles, the water is broken into fine particles, or of the so-called target spray type, where a fine stream of water under pressure is caused to impinge upon a flat surface or target. Such methods are normally rather inefficient in the use of water

Any spray system where water is run to waste must of necessity atomize as completely as possible to provide for evaporation of the maximum amount of water. This normally calls for moderately high pressures and for very fine orifices or passages in order to produce fine atomization. This in turn involves the danger of occasional clogging of the nozzles with resulting service problems. Self cleaning spray nozzles developed during the past two or three years materially reduce this hazard. Such methods even at best are wasteful of water inasmuch as only a comparatively small proportion of the water used is actually evaporated, the remainder running to waste. In spite of the water wastage this method is normally more economical in the case of small conditioning units than the installation of a pump to recirculate and reuse the waste water.

In the larger conditioning units where much larger quantities of water are used and the amount of water wastage becomes an appreciable item, it frequently is profitable to install a recirculating pump. It is important to note that when spray water is recirculated in such equipment in industrial areas and in large cities, it becomes necessary that water treatment be supplied, in order to reduce the corrosive action resulting from the absorption of acids and impurities from the air.

In some units, in order to increase the humidifying capacity and to utilize a greater portion of the spray water, the atomized spray is permitted to impinge against a heated surface thereby forcing its evaporation. While this is practical in some instances, there is danger of scale formation where hard water is employed. It should be pointed out that in any atomizing system if extremely hard water is used, the air will carry with it a content of infinitely fine solids which should be removed by a filter before discharging into the conditioned space. If it is not so removed, this may result in the formation of an apparent *dust* upon furniture and exposed surfaces.

One of the simplest methods of humidification in winter is by means of a direct steam spray. This is seldom used in air conditioning units for comfort applications due to the resulting odors. In industrial applications, however, it finds frequent use. The steam is usually introduced to the air through a perforated tube or through some type of porous material.

If a small atomizing spray is not used in comfort conditioning units, the

evaporative pan type of humidifier is usually employed. This consists of a container offering as much water surface as possible and equipped with means of heating the water. This heat may be applied either electrically, by steam, hot water, or by circulation of the water from a heated space through the evaporating pan. Since the humidification is accomplished by surface evaporation only, it is essential that the air stream be directed across the surface and that the evaporating surface be large. While this system eliminates the dusting hazard when hard water is used with a spray system, hard water tends to scale the heating surface and results in loss of capacity and the need for frequent cleaning.

The rate of evaporation per unit surface exposed is low, thus it frequently becomes difficult to provide sufficient surface for adequate capacity. The higher the temperature of the water, the lower the relative humidity of the air; the greater the velocity over the surface, the greater is the rate of humidification. The evaporative pan type of humidification limits the water wastage and is usually supplied with water through a float valve. Due to the collection of salts in this evaporating pan such humidifying systems require occasional drainage and cleaning.

Other methods of humidification attempted in air conditioning units are through the use of wetted fabrics, porous earthenware plates, or other capillary surfaces. These methods rely upon the capillary absorption of the moisture up from the liquid level into the portion exposed to the air. They have a tendency to lose their effectiveness due to the resulting deposit of mineral salts at the evaporating surfaces thereby clogging the pores and reducing the contact of the air with the water. Also they frequently become foul and often support bacterial growth.

Cooling and Dehumidification

Those units that employ recirculated water sprays will undoubtedly use such sprays as their means of cooling and dehumidification by furnishing refrigeration to the water in circulation. Occasionally where an adequate source of cold well water is available, this may be used as a direct spray and run to waste.

Other methods of dehumidification accomplished by direct contact with the transfer medium are by means of the so-called *adsorption* and *absorption* systems. (See Chapter 2 on Refrigeration.) It must be recognized that these methods of dehumidification do not in themselves provide cooling. The substance removes the water vapor from the air thereby heating it. This highly dehumidified air may then be cooled either by partial rehumidification or by direct contact with a cooling medium of cold water or direct expansion refrigerant. There are now on the market solid adsorbents such as silica gel, and activated alumina, and liquid adsorbents such as calcium chloride in solution and lithium chloride which is one of the most promising examples.

Finally a common direct means of cooling and dehumidification is through the use of ice. In such units the ice is brought into as intimate contact as possible with the air handled. Provision is made for the removal of the moisture as rapidly as it is formed from the melting of the ice. Ice is also used to cool water which is circulated through the sprays.

In conditioning units, the use of surface cooling is probably more

common than direct spray or other direct transfer means. The type of surface employed may, of course, be cast or fabricated from tubes. In present day practice finned tubes or plate fins through which tubes are passed form the most generally used cooling surface. The detailed fabrication of this surface and the arrangement of the tubes will depend largely upon the type of refrigerant for which it is intended.

The simplest construction is where chilled water or brine is used as the refrigerating medium. With direct expansion refrigerant it is usually necessary to provide a special arrangement of headers so that proper distribution of refrigerant through all the surface is obtained. In some cases, ordinary brine coils can be used when operated as a flooded refrigerant system. In some units a combination of a direct spray and a refrigerant surface is used, the spray being directed against the surface. Such systems claim the advantage of air washing together with the maintenance of a clean and effective cooling coil.

It should be noted that when surface coolers are used, adequate protection in the form of filters or at least lint screens are necessary to prevent fouling of the surface from the air borne dirt. Surface not so protected frequently becomes completely matted with lint, grease, and similar dirt.

The sources of refrigeration used with these surface type conditioning units are discussed in Chapter 2 on Refrigeration. However, they may be divided into the following groups:

1 Direct expansion refrigerant in which the liquid refrigerant is evaporated within the coils of the unit The vapor from these coils may be recompressed in centrifugal, rotary, or reciprocating type compressors, and the refrigerant again returned to the evaporator coil

2 Indirect refrigeration by means of:
   a Cold well water
   b. Cold city water.
   c. Artificial refrigerated water provided by direct expansion of refrigerant in a water cooler, direct steam jet refrigeration, or by the melting of ice.

It should be recognized that direct expansion of refrigerant within the coils of an air conditioning unit, represents a certain hazard unless the refrigerant is of the non-irritating, non-inflammable type. Therefore, in many cities rather stringent codes have been set up limiting the use of such direct expansion units. These codes are becoming more and more prevalent and it is advisable to keep informed of latest restrictions in each locality. In the case of hospitals, or places where a large number of people are present, direct expansion is prohibited and the so-called indirect systems are required. In these sytems the refrigerant is used to cool a brine which is then circulated through the cooling coil.

Filtering—Air Cleaning

A variety of methods are employed as a means of controlling air purity. In unit systems where filtering alone is considered satisfactory, the degree of filtering varies widely and in proportion to the actual needs. If the air is chiefly recirculated with but little outside air used for ventilation, filtering requirements are largely limited to keeping the coils in a clean and operable condition. Thus such units are frequently furnished with simple lint screens of low resistance and formed of moderately close

meshed wire. Where outside air is used for ventilation, more complete filtering of dust particles is necessary and for this purpose, there are a large number of filters available on the market. Some of these filters are of the so-called *throw-away* type, constructed of inexpensive material so that when they become dirty or clogged they may be thrown away and replaced with new ones. All of these filtering methods are described in detail in Chapter 16.

A number of air conditioning units are being offered for the relief of hay fever sufferers and render relief in this disease both by maintaining proper air conditions and by removing most of the air borne pollens from the ventilated air. Such filters, however, must be designed for this purpose and must have an exceptionally high efficiency.

The degree of filtration and its efficiency depends entirely upon the design of the unit or the unitary system. For high efficiency of air cleaning, large and expensive filtering area is required. Small filters mean either low efficiency of cleaning or else high resistance with increased fan power or decreased air capacity.

Ventilation

Inasmuch as air purity is one of the factors that constitute true air conditioning, ventilation or the introduction of outside air is an essential part of any air conditioning unit or system. While a unit that recirculates all its air capacity is still considered an air conditioning unit, the better type system provides for the introduction of a certain proportion of outdoor air. In some instances one of several units may operate entirely on outside air, while in other cases only a portion of the air handled by the unit is drawn from out of doors. In such cases a damper is provided either in the unit or in the duct connections for controlling the proportion of outdoor air.

Location of Air Conditioning Units

The characteristics of the conditioned space, the building construction, the type of system employed, the duct connections as well as the source of power, piping and refrigeration influence directly the location of air conditioning units.

Primarily a unit is either of the portable or fixed type. The portable unit is usually a simple self-contained air conditioning or cooling unit either with or without ventilation, but includes the condensing unit. If the condensing unit is of the air-cooled type, the unit must be located adjacent to a window or other source of outside air. On the other hand, if it is of the water-cooled type, its location should be convenient to sources of water and drainage. Portable units are invariably located within the conditioned space.

Non-portable units, or units of fixed location may or may not be located within the occupied space. Naturally units that are located in such spaces must be built with decorative cases in order to harmonize with the surroundings. Such units are normally of comparatively small capacity varying from a fraction of a ton up to as much as five or six tons. Many conditioning units and particularly those of the larger sizes are located

externally to the occupied and conditioned space and are connected thereto by means of delivery and return ducts. Such an arrangement permits the location of the conditioning unit convenient to either the sources of refrigeration or outside air. It frequently permits the use of the basement or of space less valuable than that on the level or floor of the occupied zone. Oftentimes the same type of unit may find application in an exposed position for one job and in a concealed location for another. Thus it can be seen that it is not possible to define a unit merely on the basis of its location. Frequently conditioning units are built into the structure or into the architectural design of a room so that they are entirely concealed except for the discharge and return grilles or openings which are designed so as to correspond to the decorative scheme of the room.

Air Distribution

With portable units or units exposed within the conditioned space, the distribution is usually through grilles or louvres built entirely into the equipment. The discharge of the air from this unit, in general, should be upward immediately at the unit, with sufficient horizontal component to carry it to the most remote point in the room. Such a distribution permits the cool air to drop slowly over the entire zone and return to the inlet of the unit below the breathing line and along the floor. The location of doorways, air vents, and heat exposed walls should be carefully observed, as they have a marked effect on the direction of the air flow and on its uniformity of temperature. Velocities below the breathing line should be kept low, normally not over 40 to 70 fpm. Such velocities are usually best checked by a Kata thermometer rather than by means of an anemometer.

With the suspended type of unit, located within the conditioned space, sufficient outlet air velocity should be provided to give adequate induction and mixing with the room air thereby preventing the immediate dropping of the air stream and resulting objectionable cold drafts. This is one of the limitations to the size of such suspended units inasmuch as it is difficult to distribute from one point a large refrigeration tonnage without cold drafts.

Where the units are located outside the conditioned space, air distribution is more frequently provided through multiple outlets located in ducts from the conditioning unit. The location of these outlets is quite critical and is influenced both by the building construction, economies of connections and by the distribution of load. Normally with a cooling system, it is desirable to blow the conditioned air toward the exposed walls or windows. This delivers the cool air to the point where it is most needed. These problems are discussed in considerable detail in Chapter 19.

There are a wide variety of outlet types used, and most of these have fixed delivery characteristics, thus requiring careful consideration in their location. Some types of outlets are now available with adjustable vanes thereby permitting some alteration in the delivery of the air stream after installation. This frequently eliminates objectionable down drafts resulting from the impingement of the air stream against posts, pillars, lighting fixtures, and beams.

TYPES OF UNITS

Several types and designs of air conditioning units in production and proposed are available for selection. New designs are constantly appearing, with new improvements, greater capacities, wider range of application and superior construction. It will be impossible to cover in this chapter the many types of construction on the market. Illustrations of current makes and models will be found in the *Catalog Data Section*. A few typical designs of conditioning units will be described in detail.

An all-year floor type heating and cooling unit for an exposed location and with direct expansion coil supplied with refrigerant from a remotely located compressor is shown in Fig. 10. A cooling coil for use with chilled water may be substituted for the direct expansion coil indicated. The fans below the separate cooling and heating elements deliver the air

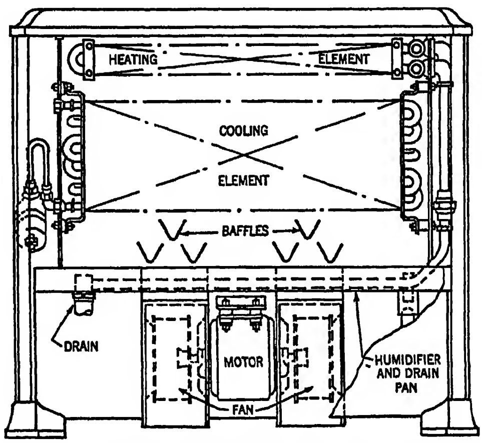


FIG 10. FLOOR TYPE HEATING AND COOLING UNIT

against deflectors thereby obtaining distribution across the face of the element and preventing condensate from dripping down into the fans. The plate upon which the fans are mounted serves as the drip pan from which the water is conducted to the drain. Separate elements are used for heating and for cooling. Thus this unit may be used automatically for heating and cooling without manual control. When the unit is used for summer conditioning only, the heating coil may be omitted for the installation. The illustration indicates an evaporative type humidifier and drain pan.

Other units are available in which a target spray humidifier is substituted for the evaporative type thereby providing a unit for summer cooling and dehumidification, at the same time supplying humidification in winter for application in rooms with other existing heat sources.

Still another unit is available in which the fans are mounted at the top of the unit delivering directly through a grille and drawing their air supply through the cooling and heating coils. Other variations in proportion and details of construction of this general arrangement are

common. With this type of unit, ventilation is usually provided by means of a separate duct connected to the inlet of the unit.

An entirely different arrangement shown in Fig. 11 places both the air inlet and the discharge at the top of the unit. The fan at one side discharges the air downward to the bottom where it turns and passes horizontally through an atomizing spray air washer. The path then continues upward through eliminators, a cooling surface and a heating surface before it leaves the unit. With steam or hot water connected to the heating element, tempered water to the sprays and refrigerated water to the cooling element, this unit gives controlled temperature, humidity, air cleaning, and air movement in both summer and winter. Air washing may be connected in summer, or in intermediate season to remove room

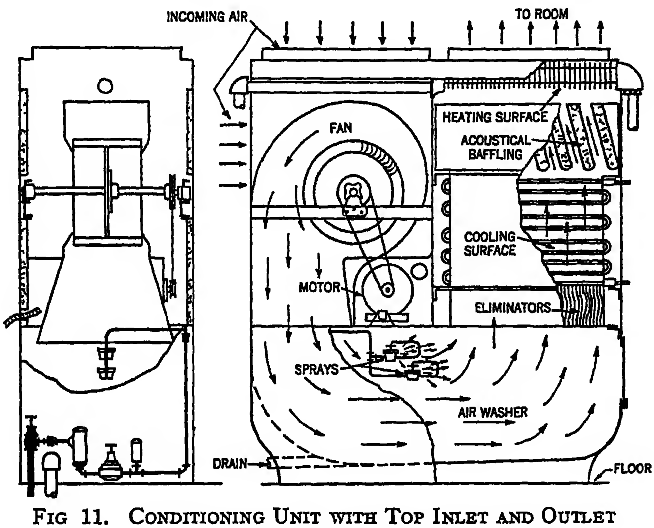


Fig 11. Conditioning Unit with Top Inlet and Outlet

odors. Excess water is run to waste. Acoustical treatment of the housing and outlet baffles permits installation where noise requirements are exacting.

One of the recently developed units, designed particularly for low cost installation is shown in Fig. 12. Twin fans with wheels mounted on extensions of the motor shaft take the air from the floor, send it downward through a passage containing a water spray and then upward through a double set of coils for cooling and heating, the air leaving the cabinet through a top grille. The spray nozzles are supplied with city water, the excess collecting in the air reversal chamber and running to the drain. The cooling coil uses water either from city mains or from other low temperature sources, or a special direct expansion coil may be provided. This unit like others may be equipped with automatic controls.

A common type of suspended type unit for exposed location utilizing a propeller type fan and suitable for summer conditioning only is illustrated in Fig. 13. Such units are equipped with either a direct expansion coil or

one for chilled water or brine circulation. The outer cabinet is commonly of wood-grained steel or baked enamel and is insulated from the cool air chamber to prevent external condensation. The drip from the coil is collected in an insulated drip pan and carried to a drain. The inlet to the unit is provided with a lint screen to protect the cooling surface. Such units are normally used for recirculation only but may be connected for ventilation through short full-size ducts. Similar units are available with twin housed fans of the same general construction, although usually such fans draw the air instead of blow it through the coils.

A self-contained completely portable cooling air conditioning unit is illustrated in Fig. 14. For the operation of this unit, it is only necessary

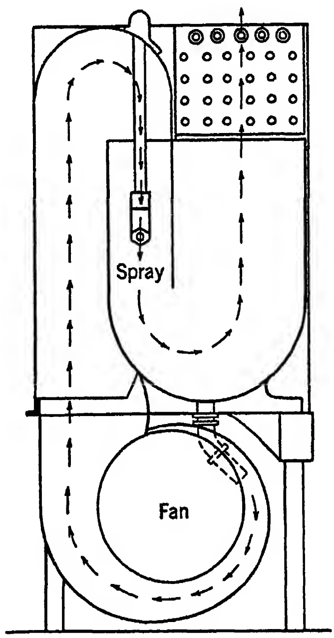


Fig. 12. Simple Spray and Coil Unit

that it be located adjacent to a window or shaft to which air connections can be made and to plug in the motors to a convenient light socket. In this unit, the conditioned air enters on the side, passing through a grille, filter, and cooling coil and is delivered vertically to the room through a special motor and fan assembly. Refrigeration is furnished by a reciprocating compressor driven from a motor located in the base. This compressor utilizes an air cooled condenser. Air is drawn into the base by a fan mounted on the compressor motor, so arranged that the air passes through the refrigeration condenser and is again discharged out through the window connection. A novel feature of this design is that the condensate from the cooling coil is sprayed over the condenser surface and there vaporized, thus eliminating the need for drain connections. One

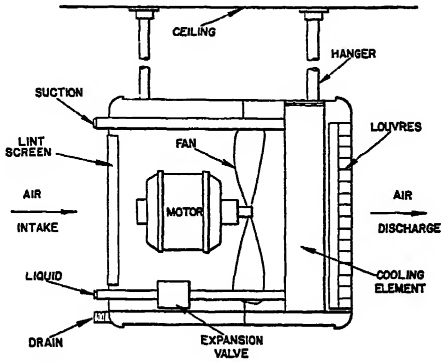


FIG. 13. SUSPENDED PROPELLER FAN TYPE COOLING AIR CONDITIONING UNIT

advantage of this type of conditioning unit is that it may be removed from the occupied space during the winter season when cooling is not needed.

Another type of portable unit for occupied space locations differs from the former in that the compressor is water cooled and a connection to water and drain must be provided in addition to the electric connection. Water and drain lines are carried in a composite hose, especially built for this purpose and connections are usually made to a nearby washbowl. In order to reduce the starting load, one model has two separate motors brought on to the line at delayed intervals thereby decreasing the initial line surge and reducing light flicker. These water cooled units either eliminate or reduce the need for outdoor air connections. Due to the necessity of water and drain connections they are not as portable as the air cooled type.

Remotely located conditioning units vary widely in details of construction. Figs. 15, 16 and 17 indicate one type built in sections thereby permitting interchangeability of application with a minimum change in

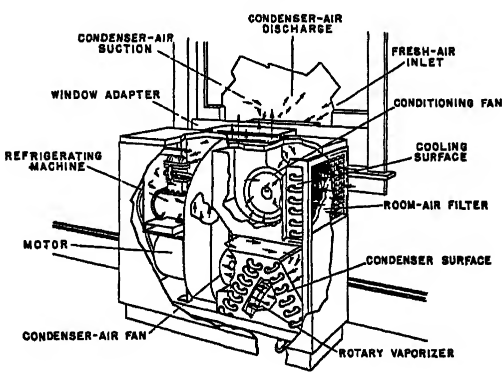


FIG. 14. PORTABLE SELF-CONTAINED CONDITIONING UNIT FOR COOLING

parts. The vertical unit shown in Fig. 15 consists of a fan section, housing one or more fans, mounted on a coil section in which are located a heating coil and a cooling coil, which may be built for either direct expansion refrigerant, chilled water, or brine. These two sections are supported on a third or drip-pan section. The distributing duct system is attached to the fan outlets and returned fresh air connections are made to the drip-pan. A filter box is illustrated attached to the drip-pan section. By eliminating the vertical type drip-pan and substituting a horizontal drip-pan, this unit is converted into a horizontal suspended type conditioning unit for connection to duct work, both with and without filters, as shown in Fig. 17.

A spray type conditioning unit is illustrated in Fig. 16. This spray type unit, which is similar to the arrangement given in Fig. 15, provides

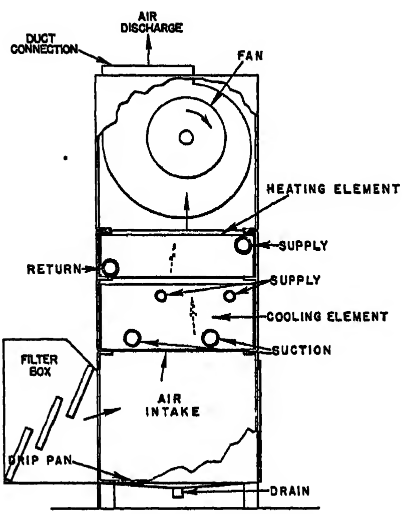


FIG 15. VERTICAL REMOTE TYPE ALL-YEAR-ROUND CONDITIONING UNIT

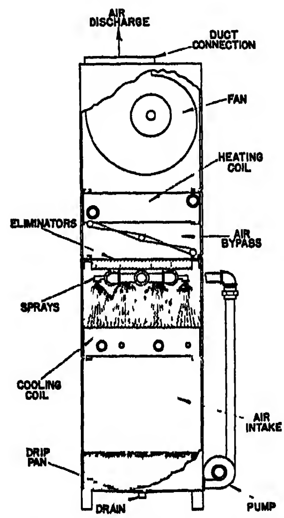


FIG. 16. SPRAY TYPE AIR CONDITIONING UNIT

for the complete washing of the air and the cooling coil. For winter operation the spray provides means for humidification. The units may also be obtained with by-pass dampers as shown in Fig. 16, to provide control of cooling in summer and humidification in winter. The spray type unit without the cooling coil may be used for humidification and heat control. This type of air conditioning unit is used in industrial process air conditioning as well as for comfort air conditioning.

Residential Central System Units

The previous figures have largely confined themselves to the illustration of all-year-round or summer conditioning units for office or commercial application. The smaller units have, of course, been applicable to residences. There remains a field of air conditioning units primarily adaptable to residence work. They are in general the outgrowth or

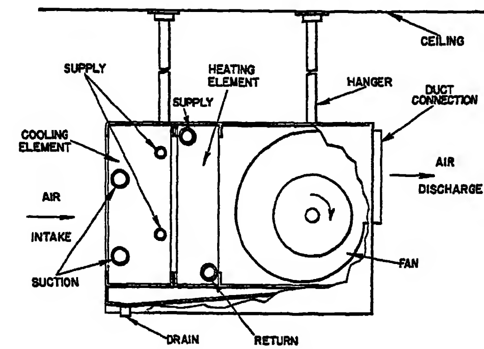


FIG 17 HORIZONTAL REMOTE TYPE ALL-YEAR-ROUND CONDITIONING UNIT

adaptation of mechanical warm air systems to conditioning, which are covered in Chapter 24. However, the following illustrations will cover details not included in that Chapter.

In Fig. 18 is shown a conditioning unit which may be operated in conjunction with a hot water or steam boiler. Heat generated in the boiler is supplied to an exchanger which raises the air temperature as it is circulated through the unit. The connection of a cooling coil to a

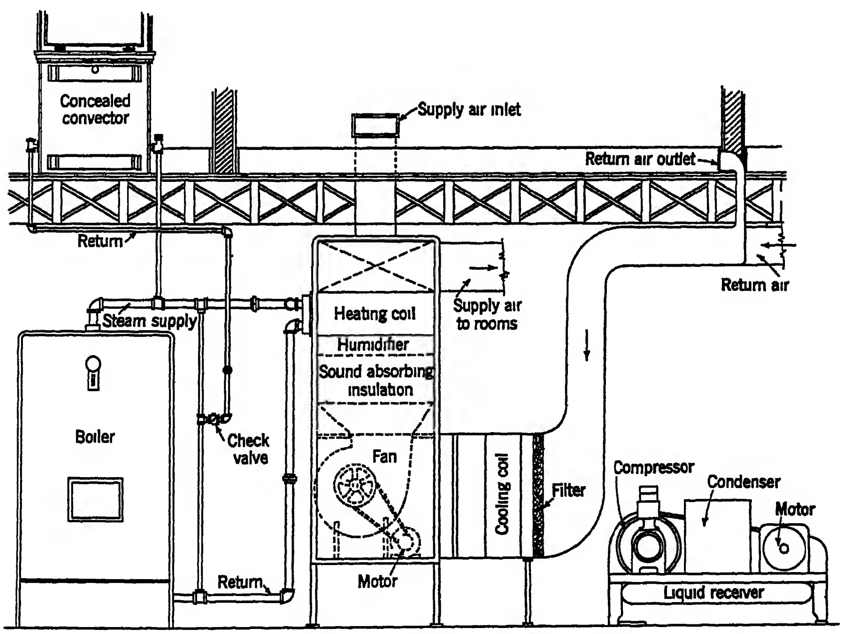


FIG. 18. RESIDENTIAL CONDITIONING UNIT WITH STEAM BOILER

source of refrigeration will provide year-round air conditioning. This type of system is particularly adaptable to a *split* system in which a portion of the residence may be conditioned in the winter and summer while the garage, servants' quarters and less freqently used rooms may be provided with radiator or convector heating directly from the boiler in the winter.

A gas-fired winter conditioning unit is illustrated in Fig. 19 which is equipped with apparatus to filter, heat, humidify and circulate the air in a residence. If a cooling coil is added to the arrangement this unit may also become a year-round conditioner.

A diagram of a direct fired fuel oil conditioning unit is given in Fig. 20. A circulating fan forces the filtered air over a heat exchanger through which the combustion gases from the oil burner are being directed. A

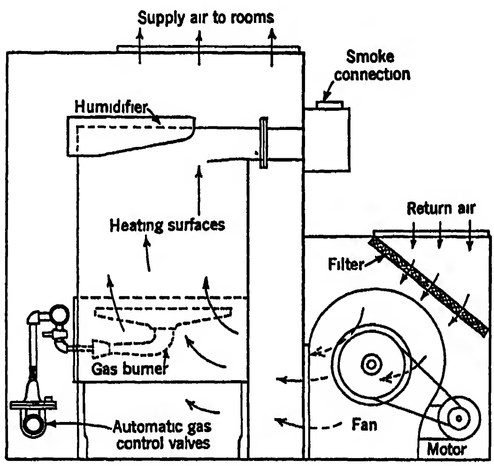


FIG. 19. GAS-FIRED FURNACE CONDITIONING UNIT

cooling section may be placed in the air inlet, with cold water, or refrigerant being circulated through the cooling element. For thermostatic control a room thermostat may be provided to start the oil burner when the temperature falls, and as soon as the temperature in the heat exchanger rises a control mechanism starts the motor operating the circulating fan. As soon as the temperature in the house rises to normal the room thermostat stops the oil burner and the fan.

BASIS OF RATING

In the past, the unit air conditioning industry has been handicapped by the lack of any standard method of rating. A proposed code giving a Standard Method of Rating and Testing Air Conditioning Equipment[8] has recently been prepared.

[8]Loc Cit Note 1

On the basis of this new code, air conditioners are to be classified primarily as *free delivery type* and *pressure type*, where delivery will be measured in cfm standard air at specified fan speed. Pressure type units will specify the delivery against various total fan static pressures. Cooling capacity will be expressed in total Btu per hour and this total will be subdivided into sensible heat cooling effect and dehumidification or latent heat effect. Cooling capacities will be given on entering air temperatures of 85 F, 50 per cent relative humidity with 40 F refrigerant temperature for comfort conditions and 45 F, 85 per cent relative humidity with 30 F refrigerant temperature for commercial applications.

The duty for heating surfaces will be specified in Btu per hour for 70 F entering air temperature based on 2 lb gage steam pressure or 180 F

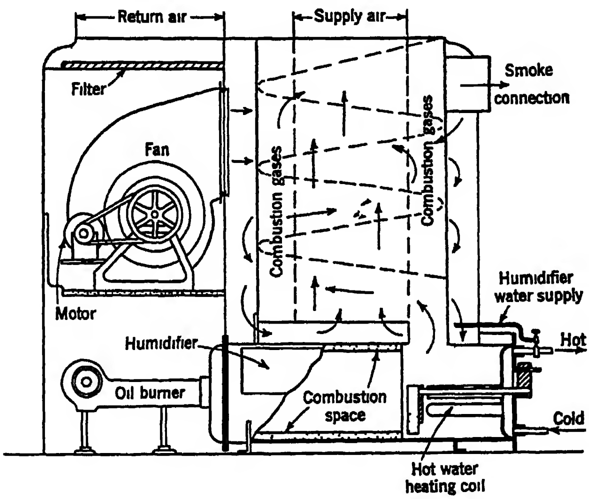


FIG. 20. OIL-FIRED CONDITIONING UNIT

entering water temperature with 20 F drop. Humidification will be specified in pounds of water evaporated per hour at 70 F and 30 per cent relative humidity entering air conditions. The *Catalog Data Section* gives ratings of current models offered by leading manufacturers.

METHODS OF CALCULATING CAPACITIES

The methods of calculating heating and cooling loads for conditioning units are similar to those described under Chapters 7 and 8. Certain manufacturers have adopted simplified and approximate methods which through experience they have found applicable to unitary equipment. Such methods involve certain averaging approximations which are suitable for estimating purposes but many of which require rechecking on a more accurate basis before actual installations are made.

The greatest error in calculating cooling loads is apt to be introduced in the failure to appreciate the magnitude of the latent heat loads and the

relationship between the latent heat removal and the refrigerant temperature. It is extremely important that the proper balance be obtained between the refrigerant temperature, the conditioning unit surface and the conditions to be maintained within the occupied space. Unsatisfactory conditions often result through the attempt to apply units with a source of refrigeration which gives too high a surface temperature. Such conditions may be obtained when well water of too high a temperature is used or when a direct expansion evaporator is connected to a refrigeration compressor of inadequate size. The high refrigerant temperature even though it may give adequate dry-bulb temperatures, due to over-size conditioning units, will not give proper humidity control due to its inability to furnish sufficient dehumidification.

The use of surface coolers with widely extended fins may lead to similar results since the large ratio of extended surfaces gives an average surface temperature considerably above the refrigerant temperature within the tubes. All these factors must be kept in mind when making the selection of equipment. It is furthermore vital that the calculation of a cooling load be based on an accurate survey before the selection of the equipment is made.

COSTS

Due to the rapid development of the air conditioning industry and the great progress that is being made each year, it is impossible to give any cost figures that will be of value. There are, however, certain factors that influence the cost of unit air conditioning installations.

1 Since the cost of the total job involves material cost plus installation labor and since through the use of unitary equipment, material costs can be kept to a minimum, every effort should be made to simplify installation.

2 Self-contained units in the small sizes now available, probably represent the lowest cost individual installations. They have, however, their limitations

3 The floor type all-year-round air conditioning units for the occupied space with a remotely controlled compressor, heating sources being either the existing heat system or steam connections to the unit, probably afford the lowest cost all-year-round service for most individual rooms. This is particularly true in the case of residences With offices, this will probably be true if the compressor can be located immediately adjacent to the conditioned space, as for example in a closet or nearby storeroom. The expenses increase rapidly as the distance from the unit increases.

4. For multiple rooms or offices, the remotely located unit with connecting ducts probably represents the most economical installation. Such installations are also particularly adaptable to stores, residences and small commercial installations.

Costs of operation vary widely depending entirely upon the cost of power and water. Water costs in the larger installations are being materially reduced through the use of cooling towers and special types of condensers. The normal expense of operating the cooling system is considerably in excess of that of winter heating both as to the first cost and as to operation. Consequently, the more rapid growth of air conditioning has been along commercial lines where it has represented an actual profitable investment resulting in increased business returns rather than along the lines of residential comfort cooling where it still represents a luxury in comfort.

MISCELLANEOUS UNITARY EQUIPMENT

There are a number of units available which were not covered in the previous discussion that accomplish only one or two of the functions of air conditioning.

Attic Fans

Attic fans, used during the warm months of the year to draw large volumes of outside air through a house, offer a means of using the comparative coolness of outside evening and night air to bring down the inside temperature of a house.

Because the low static pressures involved are usually less than ⅛ in. of water, disc or propeller fans are generally used instead of the blower or housed types. The fans should have quiet operating characteristics, and they should be capable of giving about thirty air changes per hour. The two general types of attic fan installations in common use are:

Open attic fans, in which the fan is installed in a gable or dormer and one or more grilles are provided in the ceilings of the rooms below. Fresh air, which enters the house through open windows, is drawn into the attic through the grilles, and is discharged out-of-doors by the fan. An attic stairway may be used in place of the central grille. It is essential that the roof and the attic walls be free from air leaks.

Boxed in fans, in which the fan is installed within the attic in a box or housing directly over a central ceiling grille, or in a bulkhead enclosing an attic stair. The fan may be connected by a duct system to the grilles in individual rooms. Fresh air entering through the windows of the rooms below is discharged into the attic space and escapes to the outside through louvers, dormer windows, or screened openings under the eaves.

The locations of the fan, the outlet openings, and the grilles should be selected after consideration of the room and attic arrangement in order to give uniform air distribution in the individual rooms served. If the outlet for the air is not on the side away from the direction of the prevailing wind, openings should be provided on all sides. Kitchens should be separately ventilated because of the fire hazard, and to prevent the spread of cooking odors.

The operating routine which will secure best results with an attic fan is an important consideration. A typical routine might require that in the late afternoon when the outdoor temperature begins to fall, the windows on the first floor and the grilles in the ceiling or the attic floor should be opened, and the second story windows should be kept closed. This will place the principal cooling effect in the living rooms. Shortly before bedtime, the first floor windows may be closed and those on the second floor opened, to transfer the cooling effect to the sleeping rooms. A time clock may shut off the fan before waking time, or the fan may be stopped manually at a later hour.

A disadvantage arising from the passing of a great amount of outside air through a house is the dust nuisance, which varies considerably in different locations. Persons suffering from allergic diseases caused by air-

borne pollens will have their troubles increased with attic type coolers.

Some typical data on an attic fan installation in an average six-room house of frame construction containing 14,000 cu ft and located in the southern part of this country are:

Installation cost	$75 to $400, average $250
Fan data	9000 cfm average, 280 rpm if belt driven, 570 rpm if direct connected, 500 watts input
Operating period	April 15 to October 15, intermittently as weather conditions demand
Power consumption	500 kwh per year for 8 months' operation

Humidifiers

Humidifying units may be installed as part of an air conditioning unit system, or may be installed individually to furnish additional humidity. Fig. 21 illustrates a humidifying unit for installation in connection with a warm air heating system, and as such it is located at the intake of the furnace. The air passes through a lint filter, then through the fans and finally through an air washer or spray system. Surplus spray is eliminated and the air delivered to the air distribution system In other cases, similar spray type apparatus is used to deliver humidified air through ducts to openings beneath existing radiators in a steam heated residence.

For other steam heated homes, there is a humidifying unit as illustrated in Fig. 22. This unit is normally placed at some central location on the first floor, and receives the air from the floor into fans, delivering it through a heating coil and up through a target spray atomizer. Surplus moisture is removed by means of an eliminator filter and the humidified

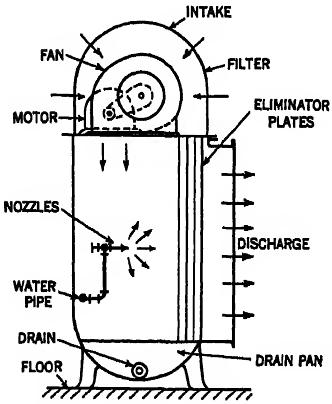


Fig. 21. Humidifying Unit for Warm Air Furnace

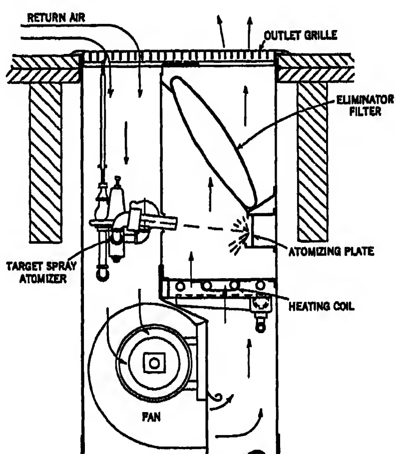


Fig. 22. Humidifying Unit for Radiator Heated Homes

air is delivered upward through the other half of the floor grille. Since a large percentage of the heater capacity is transformed into the latent heat of humidification, this unit does not eliminate any existing steam radiation. It may also be used with hot water systems but its capacity is considerably reduced.

AUTOMATIC CONTROLS

The controls of all unitary equipment represent a vital part of their successful operation. This is particularly true in the case of conditioning equipment where a close inter-relationship exists between the thermostatic room controls and the refrigerating unit controls.

The proper selection of controls and the proper adjustment is extremely important to prevent short cycling of compressors. Furthermore, the proper adjustment of direct expansion valve controls is likewise extremely important. A detailed discussion of control problems is contained in Chapter 14.

PROBLEMS IN PRACTICE

1 ● Distinguish between a unit and a central type of air conditioning system.

In a unit system, the air treating apparatus consists of factory assembled equipment which is shipped substantially complete or in sections and is installed without field fabrication except for the duct connections between the equipment and the point of delivery of the air. Usually the air treating equipment is located closely adjacent to the conditioned space and serves a limited area A central type of air conditioning system localizes the air treating equipment for the entire area at one point and involves the field assembly of a large number of individual elements Manufacturer of the unit is responsible for the output and performance of the unit under-rated conditions, whereas, the contractor installing the completely unitary or central type equipment is held responsible for the complete performance of the system.

2 ● Is it satisfactory to use superheated steam in unit heaters?

Superheated steam can be satisfactorily used in unit heaters provided the capacity is based on the saturated steam temperature and not on the total temperature. If unusually high superheat is used, trouble may be experienced from the excessive expansion and contraction of the heating elements.

3 ● Is it satisfactory to install one unit heater as the total load on a coal fired boiler?

Such an arrangement is impractical if the unit heater is started and stopped in keeping with the room temperature. However, if the room temperature controls the steam pressure and the unit heater is arranged to start when there is steam in the mains and to stop when there is no steam in the mains, such an installation will be satisfactory.

4 ● Will a unit heater with a slow speed fan be more quiet than one with a high speed fan?

Quietness is a function of the type, diameter, blade form, and location of the fan, as well as the speed For a given fan, slower speeds mean less noise.

5 ● Is it satisfactory to use steam at pressures less than atmospheric for unit heaters or unit ventilators?

If the air inlet temperature is above freezing, steam at any pressure may be used in the heating element of the unit heater or unit ventilator. If the inlet temperature is below freezing the heating element should be filled with steam of at least 5 lb pressure (or with

a positive 5 lb pressure differential between supply and return) and the steam supply should never be throttled or the heating element may be frozen.

6 ● In general, what is the primary function of a unit ventilator?

To maintain the desired room air conditions as to temperature, air change, and air cleanliness, without drafts regardless of variations in outdoor temperature, occupancy, sun, heat, and wind

7 ● What are the usual working parts of a unit ventilator?

The fan and motor assembly, a set of heating elements, outdoor and indoor air dampers, filters (optional), outlet grille, some method of varying the outlet temperature in keeping with the room requirements, and, in the case of some unit ventilators, a method of limiting the outlet temperature to a minimum of 60 F.

8 ● Do all unit ventilators introduce a constant amount of outdoor air?

Certain types employ full recirculation except when outdoor air is obtained by throttling the steam valve on the heating element so the proportion of outdoor air to room air is varied This is a very economical type of unit ventilator but in some communities it cannot be used because of existing laws which require that some fixed amount of outdoor air be introduced whenever the room is occupied. Certain types of units are designed to always take in a minimum quantity of air from the outside and to automatically vary this with the weather.

9 ● Why are metal surface cooling elements instead of liquid spray chambers used in the design of most air conditioning units and cooling units?

The first cost of the surface cooling type of unit is considerably less than the cost of spray type equipment. Further, the requirements of many industrial air conditioning jobs and of all comfort cooling jobs where unit equipment is applicable can often be effectively met with the use of surface type units, with a reduction in the space required for making the installation. Where space conditions are especially limited, the cross-sectional area of the surface cooler can be reduced because the resulting increase in velocity over the coil surface increases the effectiveness of the surface, whereas an increase in velocity through a liquid spray would reduce its effectiveness

10 ● Why are air conditioning units with metal cooling surfaces not desirable for all industrial jobs?

Wherever unusually close control of relative humidity is required, a spray type unit will prove to be more satisfactory. Relative humidity control and accurate temperature control, however, can be maintained without difficulty with the use of metal surface units.

11 ● Why is accurate control of relative humidity with surface coolers more or less complicated?

A surface cooler cannot add moisture to the air, and moisture is removed only when the surface temperature is below the entering dew-point temperature. Any change in condition of the entering air will result in a change in the dry-bulb depression of the leaving air. This change in entering condition requires not only a readjustment of the air volume but also a change in the coil temperature, if accurate control over the relative humidity is to be maintained.

12 ● What in general are the characteristics of operation of a unit using surface coils?

For a constant entering dry-bulb temperature and a constant refrigerant temperature any increase in the entering wet-bulb temperature will produce a rise in the leaving dry-bulb temperature with an accompanying reduction in the wet-bulb depression of the leaving air. The sensible heat removed by the unit decreases and the latent heat increases, while the total heat removed also increases When the dry-bulb temperature of entering air is increased, with constant refrigerant temperature and constant wet-bulb temperature of entering air, the wet-bulb depression of the leaving air increases, and since it is this depression which determines the maintained relative humidity it must be carefully considered when selecting the unit.

Chapter 14

AUTOMATIC CONTROL

Purpose of Automatic Control, Definitions of Control Units and Terms, Types of Control, Central Fan Systems, Unit Systems, Control of Automatic Fuel Appliances, Residential Control Systems, Control of Refrigeration Equipment, Industrial Processes

THIS chapter is prepared with the purpose of acquainting the engineer with the principles underlying the use of automatic control, the general types and varieties of control equipment available and their application.

Automatic control, properly applied to heating, ventilating and air conditioning systems, makes possible the maintenance of desired conditions with maximum operating economy. A properly designed and complete control system has the ability to interlock and coordinate the various functions of heating, ventilating and air conditioning in a manner impossible to accomplish with manual regulation.

Automatic control is an integral and essential part of a heating, ventilating or air conditioning installation and cannot be regarded as an accessory. In order to insure satisfactory results, the control should be designed with and incorporated in the heating, ventilating or air conditioning system. The control equipment should be given careful consideration in the planning of any installation in order that the entire system may operate together with satisfactory results.

In order that proper selection and application of controlling devices may be made it is important that a broad understanding exist as to the types of control available and their principles of operation. Improper selection and application of control equipment will result in unsatisfactory and inefficient operation. Specific control devices and systems are described in the Catalog Data Section.

PURPOSE OF AUTOMATIC CONTROL

Automatic control is normally applied to heating, ventilating or air conditioning systems:

1. To insure the maintenance of certain desired or required conditions of temperature, pressure, humidity, air motion or air distribution.

2. To serve a safety function, limiting pressures or temperatures within predetermined points, or preventing the operation of mechanical equipment unless it may function without hazard.

3. To produce economical results and thereby insure operation of the system at a minimum of expense.

DEFINITIONS OF CONTROL UNITS AND TERMS

Controlling devices and terms commonly used in the automatic control of heating, ventilating and air conditioning systems are:

Thermostats: Thermostats are defined as temperature sensitive devices reacting to temperature changes. There are four major types of thermostats.

A Room Thermostat is normally installed on the wall of the room whose temperature it is to control, and in reacting to rising or falling temperatures, the thermostat causes the operation of heating or cooling equipment so that desired temperatures will be maintained.

The temperature sensitive element will usually consist of a bi-metal strip or coil, or a vapor filled bellows as illustrated in Fig. 1.

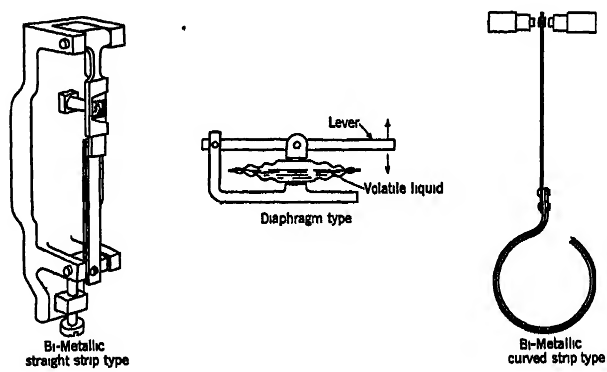


FIG. 1. TYPICAL THERMOSTATIC ELEMENTS

Immersion Thermostats are used for controlling liquid temperatures. The sensitive element will normally be encased in a protective well which is inserted in the liquid, the temperature of which is being controlled.

The temperature sensitive element will usually consist of a bi-metal coil, thermal expansion rod, or a vapor-filled system. If the latter is used the temperature sensitive bulb may be connected to the case of the instrument by either a flexible or rigid tube.

Insertion Thermostats are similar to immersion thermostats except that they are for use in controlling the temperature of a gas such as air. The sensitive element will often be encased in a protective well which prevents mechanical damage but which permits the gas to come in direct contact with the element.

Surface Thermostats include those devices which measure surface temperatures. These surface temperatures will often be an indirect measure of the temperature of a gas or fluid as in the case of a pipe within which water is flowing. The sensitive element will usually be placed in direct contact with the surface of the object whose temperature it is to measure and may consist of a bi-metal spiral or vapor-filled bellows.

Humidity Controls: Humidity controls are defined as automatic devices reacting to changes in relative humidity. Within this group, the devices which operate in controlling humidity supplying equipment are regulating devices and when operating only to prevent relative humidity from exceeding a predetermined maximum are a form of limit control.

The humidity sensitive element may consist of hair, paper, wood, skin or any other material which changes its dimensions with changes in humidity.

Controls are available provided with both temperature and humidity sensitive elements, which operate to maintain definite relations between dry-bulb temperature and relative humidity.

Pressure Controllers: Pressure controllers are defined as devices reacting to pressure and pressure changes. Examples of such devices are the pressure controls governing the operation of refrigeration equipment from either head or suction pressure, devices reacting to steam or water pressure or the pressure of air in the distribution systems.

Damper Motors: Damper motors are defined as specialized power units, the purpose of which is to position outdoor air, face, by-pass or distribution dampers, regulating the flow of air through the system. Connected by suitable linkages, these damper motors react at the command of thermostats, humidity controllers and pressure controllers to adjust the air flow to the needs of the system.

Control Valves: Control valves are defined as steam valves, water valves or air valves which may be adjusted at the command of controllers to regulate the flow of the medium passing through them to the needs of the system. Such control valves are usually constructed with an electric or pneumatic power unit connected to the valve stem so that the movement of the power unit will react to position the valve as conditions demand.

Self-contained valves are also included under this classification. Their application is principally limited to the regulation of the steam supply to individual radiators in two-pipe low pressure steam heating systems, and the temperature of hot water supply tanks.

Solenoid Valves: Solenoid valves are, as their name implies, valves actuated by the magnetic effect of an electric solenoid built within them. While normally these valves are opened when the solenoid is energized, they are sometimes built in a reverse acting manner and closed when energized. In heating, ventilating and air conditioning systems, they are normally adapted to the control of oil or gas burners as fuel valves, as water valves on humidifiers, or as refrigerant valves in refrigeration systems.

Relays: A relay is defined as a unit installed between a controller and the device under control, for purposes of amplifying the capacity of the controller or performing an auxiliary control function. For example, a thermostat, in order to preserve its sensitivity may be constructed so that it is not capable of handling the power required of a motor. A relay is, therefore, installed between the two. The thermostat actuates the relay and the relay, in turn, actuates the motor. Motor driven switching devices are also often used as relays.

TYPES OF AUTOMATIC CONTROL

Operating Media or Source of Power Supply

Automatic control systems may be classified in three broad groups based upon their primary operating media or source of power, as follows:

1. *Electric Control Systems.* In such control systems the primary medium utilized to provide for the operation is electricity, and the basic function of these controls consists of switching or otherwise adjusting electric circuits to govern electric motors, relays or solenoids. The individual units of this type of system are interconnected by line voltage or low voltage wiring, and this wiring serves to complete the circuits carrying the commands of the controllers to the controlled valves or damper motors.

2. *Pneumatic Control Systems.* In the pneumatic control systems, the primary source of operation is obtained through a medium of compressed air, the pressure of which is varied by the controlling devices. In these systems one or more centrally located air compressors furnish a supply of compressed air which is distributed in special piping to the various controlling and controlled devices. By means of leak ports or orifices, the pressure of the air is varied in the branch lines and the changing pressures are utilized in air operated damper motors or valves to obtain the movement necessary to the operation of valves and dampers.

3. *Self-Contained Control Systems.* Self-contained control systems have, in general, been restricted to such operations as could be effectively handled by a power unit with integrally mounted or direct-connected controller. Such applications consist of valves utilized to admit steam or other media into coils to regulate the temperature of tanks or to regulate the admission of steam into heating coils or radiators as determined by the controller element.

Motion of Controlled Equipment

Automatic control equipment can also be classified into two general types with respect to the characteristics of the motion imparted by the controls to the controlled equipment, such as two position or positive acting control and modulating or graduated action control.

In any control system it is necessary to choose the type of equipment whose characteristics permit the type of control operation desired and in many cases both types of control are used in the same system to best meet various requirements.

1. *Two Position or Positive-acting Control.* This type of control operates positively between two positions such as *on and off* or *open and closed* with no intermediate positions or degrees of motion between the two extremes of operation. A simple thermostat which starts and stops an oil burner or a unit heater motor is an example of this type. As applied to a valve or a damper, the action of the controlling device would serve to fully open or fully close the valve or damper.

In some applications of this type of control, artificial heat is applied to the sensitive element of the room thermostat at the same time that heat is being added to the space under the control of the thermostat in order to

increase its sensitivity. This usually results in more accurate control and more frequent operation of the heat source.

2. *Modulating or Graduated-acting Control.* This type of control causes motion in the controlled device in proportion to motion caused in the controller by fractional degree variations in the medium to which the controller is responsive. After a fractional change has been measured at the controller and has effected a new position of the valve or damper in proportion to the amount of such change, the system stands by awaiting further change at the controller before any additional motion occurs. The extent of the motion is limited only by the limits of the controller and by the intensity of the change of conditions as measured. With this type of control, the damper or control valve may be operated in intermediate positions between its extreme limits in order to properly modulate or proportion the flow of air, steam or water, reacting with changes of conditions at the controller. Various modifications of this type of control are available, designed to meet special requirements and conditions, all based on operation of the controlled equipment in intermediate positions.

This type of control motion cannot be used on valves of one-pipe steam systems as the partial opening of the valves will not permit the condensate to escape against the flow of incoming steam. This type of control should not be used to control the flow of steam to a heater coil of a fan system which is in the direct path of untempered outdoor air at temperatures below freezing, because of the possibilities of freezing condensate in the bottom of the coil.

Division of Space under Control

Control systems vary considerably with the type and size of the building, occupancy of the building, and with the heating or cooling system, humidity supplying equipment and ventilating means available for control. In the following paragraphs the general requirements of various phases of these different buildings will be discussed.

1. *Individual Room Control.* The most accurate and flexible form of control for any structure is that calling for the regulation of each individual room by control equipment reacting to conditions in that room only. Such control necessitates a thermostat in each room, located to properly measure the conditions of the room, controlling the radiator, unit heater, unit ventilator or other heating source supplying heat to that room only in which the thermostat is located. This arrangement permits the maintenance of any desired conditions in any room, entirely independent of any other room. In the case of large rooms, where one thermostat location will not serve to properly measure the conditions throughout the room, and where two or more sources of heat supply are provided in the room, additional thermostats may be used, each controlling its respective section of the heating source. This form of control, due primarily to the number of control devices required over the entire building, normally is the most expensive type of control system. However, where maximum flexibility and the most accurate control is desired, individual room control can be depended on to furnish the desired results.

2. *Single Thermostat Control.* Probably more widely used than any other form of control is the type of automatic system regulated entirely

from a single room thermostat. The wide use of this particular means of control is primarily due to the fact that it is the form of regulation best adapted to residences and small buildings, which far out-number the larger structures. In larger buildings, this form of control has definite shortcomings. In the small buildings and average size residences it is possible to select a location and install a thermostat of suitable characteristics which, in controlling from the surrounding air temperature, will hold the temperature of the entire building within entirely satisfactory limits. It must be recognized that the thermostat reacts to and controls from the temperatures to which it is subjected and that, therefore, the position selected for the thermostat must be representative of general conditions throughout the structure. It must further be recognized that if certain areas or rooms of a structure are not properly balanced as regards heating or cooling capacity and distribution, the control as dictated by the thermostat will not produce satisfactory results in these unbalanced areas.

3. *Zone Control.* As the size of buildings increases, it becomes increasingly difficult to provide proper regulation for the entire structure from a single thermostat control. In such instances, where the advantages of individual room control are not obtainable by reason of its cost, an intermediate form of control system is available, commonly described as *zone control.* In this form of control system a building is divided into areas or zones such that the general requirements and the general conditions through the areas are relatively constant as to exposure and occupancy, and then each zone is provided with control equipment which functions to regulate the conditions in that particular zone. As in the case of individual room control, each zone may be regulated to its own needs which may vary from the needs of other zones within the same structure.

Variations of the usual zone control methods by the use of recently developed special devices have been quite successful in obtaining greater economy from heating systems. Frequently these use an outside thermostat or group of thermostats which adjust the operation of the controls to conform to variations in weather conditions.

CENTRAL FAN SYSTEMS

A central fan system includes any conditioning system by which either outdoor air, return air, or combinations of outdoor and return air, are conditioned at a central point and then distributed through duct work to the various sections of the space being conditioned.

Heating Cycle

Central fan ventilating systems may be sub-divided, first into split systems, by which air is supplied for ventilating purposes only and heat is supplied in winter from another source such as direct radiation; and second, into combined systems, in which the functions of ventilation and heating are both performed by the central fan system.

A control system for a central fan ventilating system using all outdoor air and discharging air at a predetermined temperature is illustrated in Fig. 2. Thermostat T_1 located in the outdoor air intake is set just above freezing, and controls valve V_1 on the first heating coil. This valve must

be completely open or completely closed to avoid danger of freezing. The by-pass damper around the heaters and the other two valves V_2 and V_3 are controlled by thermostat T_2 located in the discharge duct from the fan. If the temperature of the discharge air increases, through the action of T_2 the damper is moved automatically to admit more cold air. Should this not reduce the temperature sufficiently, the valves V_2 and V_3 on the heating coils will be closed gradually and in sequence until the correct temperature is reached. The control of the damper and valves V_2 and V_3 must be gradual or there will be a wide fluctuation in temperature.

In ventilating systems it is customary to supply air to the ventilated spaces at an inlet temperature approximately equal to the temperature maintained in the rooms. The radiators therefore are designed to take care of all the heat losses from the room and in order to maintain controlled room temperatures it is necessary to control the radiators independently of the ventilation control.

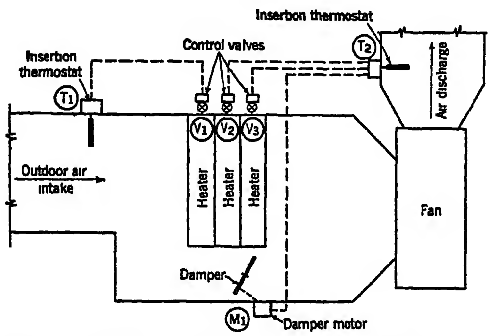


FIG. 2. CONTROL OF A SPLIT SYSTEM OF VENTILATION

In some installations, such as theatres and auditoriums, it is difficult to install sufficient direct heating surface to offset the heat losses from the room. There are also installations where a short heating-up period is allowed before occupancy of the room, and in these cases it is necessary to use the entire heating capacity of the ventilating system for this purpose. An additional thermostat may be installed in the room which will take the control away from the fan discharge thermostat (T_2 in Fig. 2) and utilize the full heating capacity when the room is below normal temperature.

In central fan systems, air washers are often used and in such cases, due to the effect of temperatures on humidity, additional control is required. An arrangement with control of the second tempering heating coil from the air washer temperature and with the usual control of the first tempering heating coil from the outside temperature is shown in Fig. 3. This permits the air to be kept cool while passing through the washer so that too much moisture will not be absorbed. Control of the reheating units and by-pass damper by an insertion thermostat in the fan discharge, and the application of a pilot thermostat to a system of this sort is illustrated in Fig. 3.

Where a number of rooms are to be heated and ventilated through one central fan system it is customary to provide tempering heating units, automatically controlled to provide a minimum temperature for ventilation only and additional heating units to supply the heating requirements. These reheating units may be located in the various branch ducts to the different rooms, each under control of its individual room thermostat, or individual ducts may be run to the various rooms from the central unit. In this case reheater coils are provided to maintain a predetermined temperature in a warm air chamber. Each room duct is connected to this warm air chamber and to the tempered air supply, and through the action of a room thermostat on a gradual-acting double mixing damper the proper proportions of warm and tempered air are secured to maintain desired conditions in the room.

In all types of central fan systems, the outdoor air damper is usually opened and closed by a damper motor controlled from a manual switch

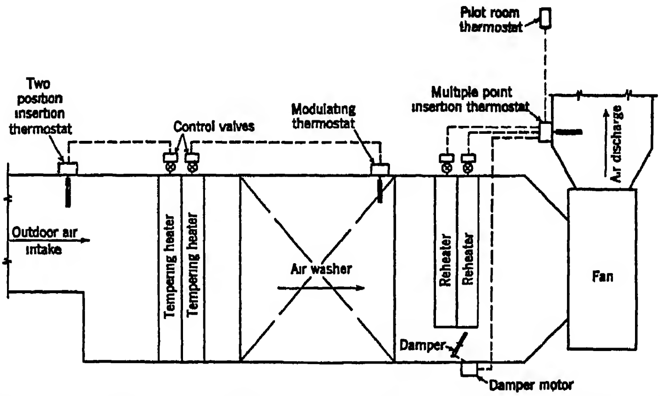


Fig 3. Control of Ventilating System with Air Washer Using Pilot Thermostat

or by a relay in the fan motor circuit, so arranged that when the fan motor is started, the relay causes the damper motor to open the outdoor air damper.

Recirculating and vent dampers may also be opened and closed by means of damper motors controlled from remote locations. Generally these damper motors are positive acting and are either completely open or closed. However, in some cases, where part outdoor air and part recirculated air is desired, it is advantageous to control the dampers so that definite proportions of damper opening area exist. In some installations the control of outdoor air and recirculating dampers is under the command of a thermostat at the intake to the conditioner, in which case the proportions of outdoor and recirculated air are fixed by the resultant temperature of their mixture. This arrangement tends to reduce the amount of outdoor air used as the outside temperature is lowered.

The operation of a central fan system during the heating cycle often results in unfavorably low relative humidity and the provision and control of humidity becomes an important factor of the system. If water spray humidification is used, control may be effected by a humidity con-

troller actuating a control valve in the water supply to the sprays. If steam humidification is used, either of the steam jet type or of the steam heated evaporating type, the flow of steam may be controlled from the humidity controller in the ventilated space. Where an air washer is used, approximate control of humidity may be obtained by maintaining the air temperature in the air washer at a predetermined desired dew-point temperature.

For example, the dew-point temperature at 70 F and 40 per cent relative humidity is 45 F. Therefore, if the air temperature is maintained at 45 F as it leaves an air washer (assuming it is fully saturated) and then is heated to 70 F, it will have a relative humidity of 40 per cent. If it is desired to maintain these conditions in a given space, the air temperature can be raised to any necessary point, say 120 F (at which the relative humidity will be only 9 per cent). When the heat in the air has been dissipated, the space temperature being maintained at 70 F, the relative humidity will be 40 per cent.

Whenever moisture is being added to the air during the heating cycle by the use of a spray or any other means, a considerable amount of care must be used in order to prevent frost from collecting on the windows due to the air being reduced below its dew-point at the inside surface of the windows.

Cooling Cycle

Central fan cooling systems are divided into two general groups based upon the methods employed to control the temperature and humidity of the treated space. Cooling normally involves the removal of moisture from the air, and to accomplish this end the temperature of the air must be lowered below the dew-point. The air at this low temperature must then be treated or introduced into the room in such manner as to avoid uncomfortable cold drafts.

In the first group the air is supplied from the conditioner after being cooled and dehumidified to a fixed temperature and humidity and then before entering the treated space is reheated. This is accomplished either by passing the air through coils heated with steam, hot water, or other heating medium, or the air from the conditioner is mixed with recirculated air before entering the conditioned space.

In the second group are those systems which use the treated space as a mixing chamber, the air being supplied to it at the temperature and humidity leaving the conditioner and depending upon diffusion in the conditioned space to give ultimately the correct conditions. In these systems the temperature and humidity of the treated space are measured and govern, through control of the cooling means, the temperature and the humidity of the air leaving the conditioner.

In Fig. 4 is represented one of the most simple central fan types of cooling system. Thermostat *T* measures the temperature within the treated space and operates to start and stop the refrigeration compressor or to control the supply of refrigerant to the cooling unit as required to maintain a fixed temperature in the space.

There are three general methods for the control of relative humidity in central fan cooling systems, which are

1. By provision for limiting the relative humidity in addition to the temperature at a definite point. When this method is used, either temperature or humidity may demand operation of the cooling source regardless of whether or not the other factor has been exceeded. The use of a high limit humidity control in this manner is desirable during conditions of high relative humidity but its operation may cause excessive cooling unless some method of reheating is employed

2 By the maintenance of a fixed effective temperature By this method, a definite relation is maintained between temperature and humidity, and sensible cooling is done whenever possible instead of the removal of latent heat in the form of moisture

3 By the maintenance of a fixed dew-point in the air discharge This method usually provides for the control of relative humidity within the space being conditioned between reasonable limits, but does not take into consideration any change in the latent heat load, as compared to the sensible heat load

The necessity for varying inside temperature conditions in accordance with changes in outdoor conditions on many types of installations is important. A control system is shown in Fig. 5 where the temperature of the treated space is adjusted according to the outdoor temperature.

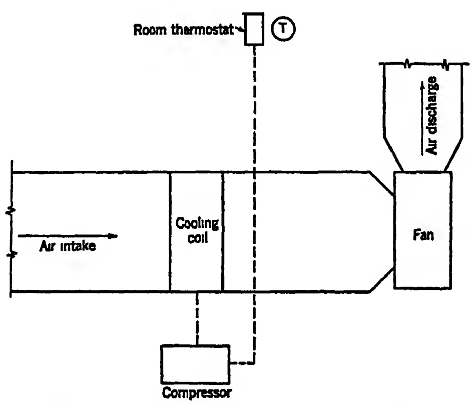


FIG. 4. DIAGRAM OF SIMPLE COOLING SYSTEM CONTROL

Thermostat T_1 measures the outdoor temperature and thereby automatically determines the inside dry-bulb temperature control point. Thermostat T_2 in the conditioned space measures the temperature of that space and controls the refrigerant to the cooling coil so as to maintain the temperature in the space being conditioned at the point which has been set up by thermostat T_1.

It is usually found desirable to adjust the indoor temperature between available limits with the outdoor temperature all of which is fully described in Chapter 3. Various combinations of control may be applied to cooling systems to secure desired relationship between outdoor temperature and resultant indoor temperature and humidity.

All Year Systems

An all year central fan conditioning system consists of the combination of a ventilating system and a cooling system.

During certain seasons of the year, it is sometimes possible to control

the dew-point of the air discharged from an air washer by regulating the relative quantities of outdoor and return air. The use of this method for controlling the outdoor and return air dampers may also provide for automatic change-over from the heating to cooling cycles, providing thereby for the maintenance of a fixed dew-point temperature in the air-discharge during both cycles.

Complete automatic control of all year systems incorporates an automatic change-over between the cooling and heating cycles. If the installation necessitates operation of manual switch or other device to change over between the heating and cooling cycles, then the control system is semi-automatic. The full automatic change-over between cycles becomes particularly desirable in the early and late portions of the cooling and heating seasons when heating is required during the early and late portion of the day and cooling may be required during the middle of the day.

A system for the control of an all year conditioning system providing for automatic change-over from the cooling to heating cycles is illustrated in Fig. 6.

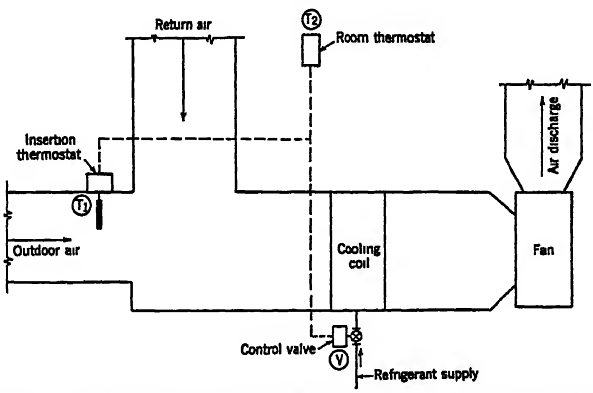


FIG. 5. DIAGRAM OF COMPENSATED COOLING SYSTEM CONTROL

During the heating cycle, thermostat T_1 in the return air or room measures the temperature of the conditioned space and modulates control valve V_1 which, in turn, modulates the flow of steam to the heating coil so as to maintain a fixed temperature in the space. Humidity control H_1 measures the relative humidity in the space being conditioned and opens control valve V_2 so as to admit water to the sprays whenever moisture is required in the air.

During the cooling cycle, thermostat T_2 in the return air measures the temperature in the space being conditioned and modulates control valve V_3 which, in turn, modulates the flow of water to the cooling coil so as to maintain a fixed temperature in the space. Humidity control H_2 measures the relative humidity in the space being conditioned and then assumes command of control valve V_3 whenever the relative humidity exceeds a predetermined amount.

During the heating cycle thermostat T_3 acts as a low limit. It assumes command of control valve V_1 whenever it is necessary to prevent the air

discharge temperature from falling below a minimum point. Thermostat T_3 may also be arranged to act as a low limit during the cooling cycle if the conditions of the installation make it desirable.

Thermostat T_4 installed in the inlet to the conditioner controls damper motor M_1 which in turn regulates the relative quantity of outdoor and return air admitted to the system. This damper action may be provided with a minimum setting of the outdoor air damper so that a minimum fixed requirement of outdoor air will be insured for ventilating purposes.

Humidity control H_3 measures the outdoor air relative humidity and prevents the outdoor air damper from opening beyond its minimum

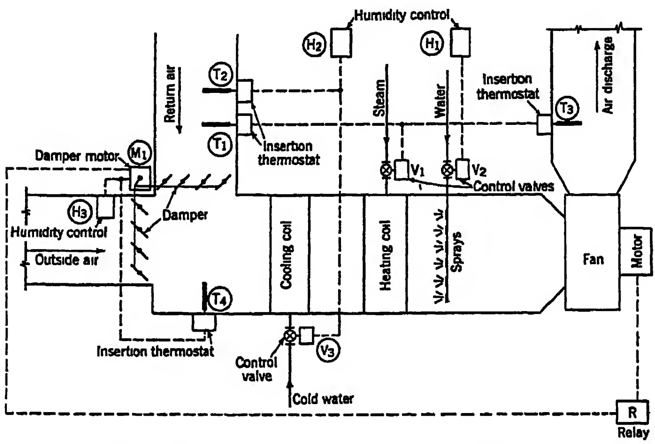


FIG 6 DIAGRAM OF COMPLETE AUTOMATIC CONTROL ALL YEAR AIR CONDITIONING SYSTEM

position whenever the outdoor air relative humidity exceeds a predetermined point.

When the fan is stopped, relay R positions damper motor M_1 so as to close the outdoor air damper.

Thermostat T_1 must be set at a lower temperature than thermostat T_2 in order that each may assume command upon the fall or rise respectively of the temperature of the return air. As an example, T_1 might be set at 72 F and T_2 at 76 F. When the temperature of the return air approaches 72 F, it would indicate that a change had taken place from the cooling to the heating cycle and when the return air approaches 76 F, it would indicate that a change has taken place from the heating to the cooling cycle.

UNIT SYSTEMS

A unit system provides for the same functions as a central fan system except that the actual conditioning is usually done within the space being conditioned instead of at some central location outside of the space. The

automatic control problems, therefore, become exactly the same as for central fan conditioning systems except that compactness, ease of installation and control cost often assume somewhat more importance.

Because of the usual segregated location of unit equipment throughout a building and its consequent lack of competent supervision, complete automatic control is essential to its satisfactory operation.

Unit Heaters

In its simplest form, unit heater control consists of a room thermostat the function of which is to start the unit heater motor when heat is required and shut it off when the demand is satisfied. With this limited control, it is possible in some instances that, with no steam available at the heater, the operation of the fan at the command of the thermostat would cause objectionable drafts. To prevent this occurrence, limit controls are available which will prevent the operation of the fan at the command of the room thermostat except when steam is available, as determined by the temperature of the steam or return pipe or the pressure of the steam supply.

In some cases it is desirable to operate the unit heaters continuously for circulation of air where, due to the type of installation, cold drafts will not result therefrom. In such instances the room thermostat regulates the supply of steam to the unit through a control valve in the steam supply line and the unit heater motor operation is manually controlled.

Where several unit heaters serve a limited area, they may be grouped for purposes of automatic control, and several heaters placed in operation at the command of one thermostat. By properly grouping the units which will operate together, the benefit of zone control can often be obtained with a minimum of control equipment. Where such group operation is utilized, the thermostat and limit control usually function through a relay, as the combined load of the several motors may exceed the current capacity of the thermostatic control device.

Cooling Units

The recommended form of temperature control for the cooling unit contemplates the continuous operation of the cooling unit fan with automatic two position regulation of the compressor or cooling coil as determined by a room thermostat or by a temperature controller measuring the temperature of the return air as it is taken into the cooling unit. Such operation insures continuous circulation of the air in the room served by the cooling unit, and in addition to providing the cooling effect due to the moving air, this circulation overcomes the tendency of air to stratify. Thus, as this temperature tends to rise, the temperature controller will open the valve supplying either refrigerant or cold water to the cooling unit coil or start the compressor.

Cooling units may also be controlled by arranging the room thermostat to start and stop the fan motor or by a combination of motor and refrigerant control.

A humidity controller may be used in conjunction with the thermostat as a high limit control to permit the cooling and dehumidifying of the air

whenever the relative humidity rises above some predetermined point such as 60 per cent even though the thermostat is satisfied. This control is desirable on damp days or in conditions where the humidity load may become excessive, but its operation will result in excessive cooling unless some means of reheating is provided.

Unit Ventilators

There are various types of unit ventilators available but in general all types are designed to draw air from the outside or to mix outside and recirculated air, heat it and introduce it into the room under control of a thermostat.

In the application of control to unit ventilators the essential requirement is that the action be graduated to prevent sudden changes in the temperature of the discharged air and where direct radiation is used in conjunction with the unit that the cycle of control be so arranged that steam will be admitted to the direct radiation only when the unit is unable to carry the heating load. This arrangement prevents the unit from delivering air at low temperatures to offset the overheating effect of the direct radiation and results in the delivery of a higher percentage of tempered air.

There are two general types of control applied to unit ventilators as follows.

1. The mixing or by-pass damper type of unit is provided with a damper, equipped with a damper motor, which, under control of the thermostat, passes air through and around the heating element in such proportion as to maintain a uniform room temperature, the two streams of cold and tempered air being mixed and diffused at the ceiling A control valve may also be used on the steam supply to the heating element of the unit and should be arranged to throttle the steam supply when the damper approaches a position to by-pass all of the air

The outside air damper of this type of unit is usually provided with a damper motor and controlled by a remote manual switch to assume either a fully open or fully closed position

2. The recirculating type of unit ventilator is equipped with a control valve on the steam supply to the heating element of the unit and with a damper motor on the outside air-recirculating air damper, both under the control of the room thermostat. Some units are so arranged that a mixture of outside air and recirculated air passes through the heating element and others so that only the recirculated air is heated.

The fundamental requirements of control as applied to this type of unit is that the steam supply to the direct radiation, the steam supply to the unit ventilator and the mixing of outside and recirculated air be accomplished in a definite cycle or sequence to meet the requirements of the particular unit used and differs from the mixing damper type of unit in that the percentage of outside air and recirculated air delivered by the unit is determined by room temperature. The damper motor is sometimes arranged so that a fixed minimum quantity of outside air is delivered continuously as soon as the room has reached a predetermined temperature A limit thermostat, either in the mixing chamber or in the discharge of the unit, is sometimes used in conjunction with the room thermostat, so arranged that the action on either the control valve or the dampers, or both, is stopped when a predetermined minimum temperature has been reached in the unit discharge, to prevent delivery of air at a lower temperature.

For additional information on the control of unit ventilators when installed and operated under various types of applications refer to Chapter 13.

All Year Conditioning Units

It is desirable to provide for automatic change-over between the cooling and heating cycles in the control system for all year conditioning units because of the probable necessity of changing over a large number of units if done manually.

A control system for an all year conditioning unit providing for the automatic change-over is shown in Fig. 7. Operation of the control equipment is as follows:

1. *During the Heating Cycle.* Combination controller T_1 measures the temperature in the space being conditioned and opens control valve V_2 so as to admit steam to the heating coil whenever heat is required so as to maintain a fixed temperature in the space. Combination controller T_1 also measures the relative humidity in the conditioned space and opens

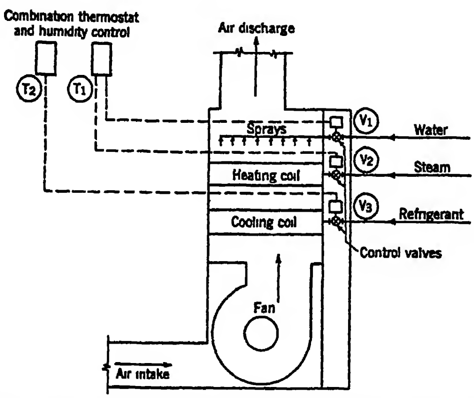


FIG. 7. ALL YEAR AIR CONDITIONING UNIT WITH COMPLETE AUTOMATIC CONTROL

control valve V_1 so as to admit water to the sprays whenever moisture is required in the space.

2. *During the Cooling Cycle.* Combination controller T_2 measures the temperature and humidity in the conditioned space and opens refrigerant control valve V_3, thereby admitting refrigerant to the cooling coil whenever cooling is required to maintain the temperature or relative humidity within predetermined maximum limits.

The temperature control point of controller T_1 must be set at a lower point than that of controller T_2 in order to provide for the automatic change-over between the cooling and heating cycles. As an example, controller T_1 might be set at 72 F and controller T_2 at 76 F. As the temperature in the space approaches 72 F, it would indicate a change from the cooling to the heating cycle and when the temperature in the space approaches 76 F, it would indicate a change from the heating to the cooling cycle, and the corresponding controllers would assume command. In the same way, the relative humidity control point of controller T_1 would be set at a lower point than that of controller T_2. As an example, T_1 might be set at 35 per cent and T_2 at 60 per cent.

CONTROL OF AUTOMATIC FUEL APPLIANCES

It is essential that automatic controls be used with oil burners, gas burners, and stokers in order to maintain even temperatures and provide safe and economical operation of the heating plant. There are many types of burners and many types of automatic control, and it is essential that the proper type of control equipment be selected to fulfill the requirements of the burner equipment and its application.

Combustion regulation equipment should be used on the larger commercial and industrial applications to control the secondary air supply and thereby provide for economical operation. This type of control will usually consist of a pressure regulator which measures and controls the pressure over the fire and which thereby indirectly regulates the carbon dioxide percentage in the flue gas.

On all automatically fired steam boilers it is advisable to provide control equipment which will stop the burner operation in case the boiler water line falls below a predetermined level of safety.

Thermostats used to control automatic fuel appliances may be provided with clock mechanisms which will operate to maintain lower temperatures during night hours for economy of fuel.

Oil Burner Controls

In the normal oil burner installation as encountered in residential and small commercial installations, the burner operation is frequently regulated by electric controls and primarily governed by a room thermostat. It is essential that a limiting control be incorporated in the control system to prevent the temperature of the heating medium from exceeding any predetermined safe maximum. The type of limit control selected will depend on the type of the heating system. In a warm air furnace installation, a limit control would be used, reacting to the temperature of the heated air in the bonnet of the furnace; in a hot water system a control reacting to the temperature of the water in the boiler; and in a steam system a control reacting to the pressure of the steam in the boiler.

In addition to the normal control of the burner from the room thermostat and limit control, it is necessary that a combustion safety device be used to prevent operation of the burner under hazardous conditions. The oil fire is automatically ignited by means of gas, electric spark or incandescent element and the combustion safety control acting through a sequence device permits the burner operation only when the fire is properly established as the burner starts up. A further function of the combustion safety control is to react to any major disturbance in the flame during the running operation, shutting down the burner and preventing the discharge of unburned fuel if for any reason the flame is extinguished.

Gas Burner Controls

In the case of the domestic burner, full automatic operation is the normal requirement and the burner is started and stopped at the command of a room thermostat which, in turn, opens and closes a control valve in the gas supply line. For purposes of preventing abnormally high temperatures in the bonnet of gas fired furnaces or in the temperature of

the water in gas fired hot water heating boilers or excessive pressures in gas fired steam boilers, temperature and pressure limit controls are used. Ignition is normally secured through the use of a gas pilot flame and a safety device is provided, utilizing the heat of the pilot flame in such a manner that if the pilot light is extinguished for any reason, the main gas valve cannot be opened. For satisfactory and economical operation, all automatically fired gas burners should be equipped with pressure regulators on the gas supply line.

Stoker Controls

Domestic stokers are normally placed under command of a room thermostat for primary operation subject also to the command of a limit control to prevent their operation when conditions in the boiler or furnace exceed predetermined safe maximums. Utilizing coal as fuel, automatic ignition is not provided and the stokers, once ignited, maintain their fire, merely changing the rate of combustion by changing the draft and the rate at which the coal is fed. Thus, at the command of the room thermostat the stoker motor is started, driving a forced draft fan and fuel feeding mechanism. The rate of combustion is thus increased and this operation continues until the thermostat has been satisfied when the motor is stopped and the fuel in the combustion chamber continues to burn at a slow rate with reduced draft.

At certain seasons of the year, the operation of the stoker under the requirements of the thermostat may be so infrequent that there is a possibility of the fuel in the combustion chamber burning out or the fire going out between operations. To prevent this occurrence, automatic controls may be utilized to operate the stoker independently of thermostat requirements, sufficiently to sustain the fire either through a timing device functioning for short periods at predetermined intervals or through a temperature control device reacting to minimum stack or boiler temperatures. Control may also be utilized to prevent stoker operation and the delivery of coal into the combustion chamber in the event that the fire has gone completely out. This control is governed normally by the stack temperature and shuts down the stoker after a predetermined minimum stack temperature is reached.

RESIDENTIAL CONTROL SYSTEMS

The control installation in a residence may vary from the simple regulation of a coal-fired heating plant to the completely automatic all year air conditioning system. Residential installations with automatic fuel burning appliances, such as oil burners, gas burners or stokers, are normally equipped with single room thermostat, limit and safety controls as outlined above under Control of Automatic Fuel Appliances.

Coal-Fired Heating Plant

Control in the normal coal-fired domestic heating plant consists of regulating the combustion rate in accordance with requirements. This function is accomplished by a spring or electric-driven damper motor which under the command of a room thermostat and through chain linkage, operates the draft and check dampers of a boiler or warm air

furnace. Such installation should be protected against excessive temperature or pressure by means of a limit control serving to check the fire when temperature or pressure conditions at the boiler or furnace reach a predetermined maximum.

All Year Domestic Hot Water Supply

Hot water or steam heating boilers with automatic fuel burning appliances can be used for all year heating of domestic water supply. The fuel burning appliance in this case is controlled from the temperature of water or pressure of steam in the boiler to maintain uniform boiler conditions and domestic hot water is heated by means of an indirect heater. The heating of the residence is normally governed by means of a thermostat which operates a control valve in the flow line of a gravity hot water or a steam system, or controls the operation of a circulating pump in a forced circulation hot water system.

Air Conditioning Systems

Residential air conditioning systems are of various types normally including a heating source and a motor driven fan for circulating air. In addition, such installations may involve spray-head equipment, the purpose of which may be only to supply humidity, or which, is some instances, are of greater capacity and serve not only to humidify but to wash the air passing through them. It is also common practice to include dry filters to aid in air cleaning. Such installations distribute suitably heated and humidified air during the heating cycle, and during the summer or cooling cycle may be used effectively as conditioners if the washer unit is supplied with water at suitable temperature or if such an installation is equipped with other refrigeration means.

During the heating cycle the regulation of temperatures is normally one or the other of the problems previously discussed in connection with the various types of heating sources described, such as the oil burner, gas burner, stoker or the coal-fired heating plant under automatic control. Regulation of the humidity during the heating cycle is normally accomplished by opening and closing a solenoid water valve supplying water to the spray-heads, the solenoid valve being under control of a room type humidity control. In the average installation the fan is permitted to run only during such intervals as the thermostat is calling for heat or at the command of a limit control to prevent the overheating of the bonnet of a warm air furnace. The limit control should also prevent the operation of the fan at the command of the thermostat until the circulating air temperature has increased to a predetermined point.

When cooling equipment is provided in such installations, control during the cooling cycle will be an adaptation of the control principles described for central fan systems selected for the type of cooling equipment utilized.

The selection of automatic control equipment for residential air conditioning systems is just as important as for commercial installations. Fewer controls are generally used and systems are usually less complicated except in the case of a very large residence installation when the control system may become as complete as the commercial installation.

CONTROL OF REFRIGERATION EQUIPMENT

The most common means of providing cooling for air conditioning may be divided into four general classifications as follows:

Compressor Type Refrigeration

Refrigeration compressors may furnish refrigerant to direct expansion cooling coils through which air is being passed, or to coils in cooling tanks through which water is passed which is then pumped to air washers or cooling coils through which the air is passed.

In either case the compressor motor may be started and stopped in order to meet the demand for refrigeration or a pressure controller may be used to regulate the low side or suction pressure of the compressor. When the latter method is used, the flow of refrigerant to cooling coils may be regulated by the opening and closing of a solenoid refrigerant valve at the command of a temperature controller or thermostat.

A high pressure cutout as an individual unit or in combination with either a temperature or pressure controller provides a safety feature against the development of excessive pressures on the high side of the compressor.

Refrigeration by Ice

When ice is used for the cooling or dehumidification of air, it is usually placed in bunkers and water is sprayed over it. This water, after being cooled, may be used in air washers or surface cooling coils and is usually returned to the bunker for additional cooling after being used.

Control of the water temperature leaving the cold water tank may be maintained by a temperature controller, which measures the temperature of the water in the tank and modulates a control valve in a by-pass which permits a portion of the return water to return directly to the tank instead of passing through the sprays.

Vacuum Refrigeration

A vacuum refrigerating system consists of an evaporator, compressor, condenser and auxiliaries. The refrigerant used is water, and water vapor (steam) is the power medium.

Water which has been passed through an air washer or cooling coil is sprayed directly into the evaporator or water cooler where it is cooled by its own evaporation. A condenser is attached directly to the compressor discharge and its function is to recondense the water vapor drawn from the evaporator, plus the steam which supplies the energy for compression.

The temperature of the cold water leaving the flash chamber should be measured by a temperature controller which will in turn operate a two position or positive control valve installed in the steam line to the jet so as to permit steam to flow only when cooling is required. If city water is used in the condenser, the amount of water should be modulated according to the demand as measured at the condenser outlet by means of a temperature controller and control valve.

Refrigeration by Well Water

When well water is available in sufficient quantities at low temperatures during the cooling season, it may be pumped directly to air washers or cooling coils Control is usually effected through control valves on the water supply to the cooling unit actuated by temperature or humidity controllers, or both, located either at the outlet of the conditioner or in the conditioned space.

INDUSTRIAL PROCESSES

There are many industrial processes requiring automatic temperature and humidity regulation. The control equipment operates on the same principles that have been described, but it is often especially designed for each particular process. Each installation, or the installation for each process, is likely to be a problem peculiar to that process.

PROBLEMS IN PRACTICE

1 ● What important functions of heating, ventilating, and air conditioning systems do automatic controls fulfill?

Controls are applied to maintain adequate requirements for human comfort and efficiency; to maintain requirements for industrial processes, to obtain economy in operation, and to provide necessary safety measures.

2 ● How may temperature control be obtained in a room heated by a unit heater?

With constant steam supply, the unit heater motor may be started or stopped by a thermostat, either directly or through a relay. With intermittent steam supply, operation of the motor by thermostat can be limited to the time that steam is available, by using a reverse-acting temperature or pressure limit switch.

3 ● How may temperature control be obtained in a room cooled by a self-contained mechanical unit?

The fan operation may be controlled by a manual switch, while a room thermostat in conjunction with a solenoid valve may regulate the flow of the refrigerant to the coil. The thermostatic circuit might be operative only when the fans are running; and the compressor might be controlled by refrigerant pressure.

4 ● How may temperature control be obtained in a room heated by an automatically-fired warm air furnace?

A room thermostat might control the combustion unit, and a limit switch in the top of the furnace unit, when at a low setting of its control might operate the fan whenever there is a rise of temperature, and when at a high setting of its control it might shut off the combustion unit. A room humidity control operating a solenoid valve on the water supply to the humidifier, or operating a relay on the recirculating pump motor to the humidifier, may be connected in parallel with the fan motor. Humidification may be supplied only when heat is supplied and when the humidity control acts in conjunction with a time switch.

5 ● How may humidity be controlled in a unit humidifier for a steam or hot water heating plant?

Since heat is required for evaporation, a temperature limit switch, preferably of the immersion type, may be placed in the heating supply riser to cause the unit to be inoperative when heat is not available. A room humidity control will operate a solenoid valve on the water supply to the sprays. Both the solenoid valve and the humidity control may be electrically wired in parallel with a fan motor, and be subject to the temperature limit switch.

Chapter 15

AIR POLLUTION

Sources of Air Pollution, Effects of Air Pollution on Health, Pulmonary Effects, Occlusion of Solar Radiation, Industrial Air Pollution, Abatement of Atmospheric Pollution, Smoke Abatement, Dust and Cinder Abatement

THIS chapter considers the hygienic aspects of atmospheric pollution and the methods by which this pollution may be lessened. Information concerning the cleaning of air brought into buildings for ventilating purposes will be found in Chapter 16, and a discussion of the exhausting of dusts and toxic gases from factories and industrial plants is considered in Chapter 21.

The impurities which contribute to atmospheric pollution include carbon from the combustion of fuels, particles of earth, sand, ash, rubber tires, leather, animal excretion, stone, wood, rust, paper, threads of cotton, wool, and silk, bits of animal and vegetable matter, and pollen. Microscopic examination of the impurities in city air shows that a large percentage of the particles are carbon. (See Fig. 1, Chapter 16, for size of impurities in air.)

CLASSIFICATION OF AIR IMPURITIES

The most conspicuous sources of atmospheric pollution may be arbitrarily classified according to the size of the particles as dusts, fumes, and smoke. *Dusts* are particles of solid matter varying from 1.0 to 150 microns in size. *Fumes* include particles resulting from chemical processing, combustion, explosion, and distillation, ranging from 0.1 to 1.0 micron in size. *Smoke* is composed of fine soot or carbon particles, less than 0.1 micron in size, which result from incomplete combustion of carbonaceous materials, such as coal, oil, tar, and tobacco. In addition to carbon and soot, smoke contains unconsumed hydrocarbon gases, sulphur dioxide, sulphuric acid, carbon monoxide, and other industrial gases capable of injuring property, vegetation, and health.

The lines of demarcation in these three classifications are neither sharp nor positive, but the distinction is descriptive of the nature and origin of the particles, and their physical action. Dusts settle without appreciable agglomeration, fumes tend to aggregate, smoke to diffuse. Particles larger than one micron will eventually settle out by gravitation; particles smaller will remain in suspension as permanent impurities unless they agglomerate to sizes larger than one micron.

The term *fly-ash* is usually applied to the extremely small particles of ash, and the term *cinder* to the larger particles of coke and ash which are discharged with the gases of combustion from burning coal.

AIR POLLUTION AND HEALTH

Many kinds of dusts and gases are capable of producing pathological changes which may cause ill health. The harmful effects depend largely upon the chemical and physical nature of the impurities, and the concentration, length of time, and conditions under which they are breathed. Dust particles must be minute in size to be inhaled at all, although fairly large particles may gain access to the upper air passages.

The human body possesses remarkable filtering media for protecting the lungs. Small hairs which line the nasal passages, and a multitude of microscopic hairs, called *cilia*, in the epithelial lining in the bronchial tubes intercept many of the dust particles before they reach the lungs.

The constant inhalation of dusts in city air irritates the mucous membranes of the nose, throat, and lungs, and eventually may produce discomfort and a series of minor respiratory disorders. The pigmented lung of the city dweller is an example of the pathological change produced over a period of years. This condition may be of no clinical importance, but an exaggeration of it in the coal miner results in anthracosis or dark spots on the lung due to the presence of pigment in the lymph channels which impairs the functioning of the lung cells under stress.

Effects of Solids

Bronchitis is the chief condition associated with exposure to thick dust, and follows upon inhalation of practically any kind of insoluble and non-colloidal dust. Atmospheric dust in itself cannot be blamed for causing tuberculosis, but it appears to have a marked influence in aggravating the disease once it has started. There is, however, quite reliable evidence that carbon pigment, one of the atmospheric dusts, tends to wall off local tuberculosis rather than to further its spread.

The sulphurous fumes and tarry matter in smoke are probably more dangerous than the carbon. In foggy weather the accumulation of these substances in the lower strata may be such as to cause irritation of the eyes, nose, and respiratory passages, leading to asthmatic breathing and bronchitis and, in extreme cases, to death. The Meuse Valley fog disaster will probably become a classic example in the history of gaseous air pollution. Released in a rare combination of atmospheric calm and dense fog, it is believed that sulphur dioxide and other toxic gases from the industrial region of the valley caused 63 sudden deaths, and injuries to several hundred persons. Physical examination showed difficult breathing, rapid pulse, cyanosis, cardiac dilation, and a redness and inflammation of the mucosa of the nose, mouth, throat, trachea, and bronchi.

Carbon monoxide from automobiles and from chimney gases constitutes another important source of aerial pollution in busy cities. During heavy traffic hours and under atmospheric conditions favorable to concentration, the air of congested streets is found to contain enough *CO*

to menace the health of those exposed over a period of several hours, particularly if their activities call for deep and rapid breathing. In open air under ordinary conditions the concentration of *CO* in city air is believed to be insufficient to affect the average city dweller or pedestrian.

Occlusion of Solar Radiation

The loss of light, particularly the occlusion of solar ultra-violet light due to smoke and soot, is beginning to be recognized as a health problem in many industrial cities. Measurements of solar radiation in Baltimore[1] by actinic methods show that the ultra-violet light in the country was 50 per cent greater than in the city. In New York City[2] a loss as great as 50 per cent in visible light was found by the photo-electric cell method.

The effect of air pollution on the health of city dwellers is difficult to determine, owing to the slowness of its manifestations. The aesthetic and economic objections to air pollution are so definite, and the effect of air-borne pollen can be shown so readily as the cause of hay fever and other allergic diseases, that means and expenses of prevention or elimination of this pollution have seemed justifiable to the public.

AIR POLLUTION IN INDUSTRY

In many industrial processes, sufficient amounts of dusts, fumes, and vapors are liberated to be injurious to the health of workers. Some dusts are poisonous (lead, mercury, arsenic, manganese, and cadmium) and some act as irritants (silica, steel, iron, and granite). Certain dusts may produce catarrhal conditions and increase susceptibility to such diseases as bronchitis, pneumonia, and tuberculosis. Silicious dust is especially harmful because it has a direct damaging action upon the tissue of the lungs, but organic dusts, both animal and vegetable (hair, pollen, textile, and fiber), do not seem to affect the lungs at all, although they may cause considerable discomfort in the upper respiratory passages to persons sensitive to them.

Industrial gases and fumes act specifically upon the mucous membranes, the lungs, blood, skin, and eyes. Some extremely poisonous gases act after very short exposures. Among these are carbon monoxide, hydrogen sulphide, ammonia, chlorine, bromine, arsine, and cyanogen.

The industrial processes which liberate harmful substances are too manifold and the effects too diverse to be considered here, where discussion is limited to the commonest and most serious with which the ventilating engineer may be confronted, namely, carbon monoxide, lead, and silica. For a more thorough treatise on the subject reference should be made to books by Hamilton[3], Rosenau[4], and Henderson and Haggard[5].

Carbon Monoxide Poisoning

Carbon monoxide is a common form of poisonous industrial gas, met with in mines, foundries, coke-oven sheds, garages, and houses. Its action

[1]Effects of Atmospheric Pollution upon Incidence of Solar Ultra-Violet Light, by J. H Shrader, M. H Coblentz and F A. Korff (*American Journal of Public Health*, p 7, Vol. 19 1929)
[2]Studies in Illumination, by J E Ives (*U S Public Health Service Bulletin No 197*, 1930).
[3]Industrial Poisons in the United States, by Alice Hamilton
[4]Preventive Medicine and Hygiene, by Milton J Rosenau
[5]Noxious Gases, by Y. Henderson and H Haggard

is due to the fact that the combining power of carbon monoxide with the haemoglobin of the red blood corpuscles is about 300 times greater than that of oxygen. Since the resulting stable combination destroys the power of the haemoglobin to unite with oxygen in the lungs and to supply it to the tissues, the effects are due to lack of oxygen, and the symptoms are those of anoxemia, namely, dizziness, headaches, sleepiness, fatigue, and, in extreme cases, paralysis and death. The dangerous saturation level of the blood with carbon monoxide is about 50 per cent. Even as little as 0.07 per cent in the air will render, in half an hour, one quarter of the red corpuscles incapable of uniting with oxygen. One to two parts per 10,000 parts of air is set as a safe limit of pollution which may be breathed for a long time without producing perceptible symptoms.

Silicosis

Silicosis is a chronic disease of the lungs which results from the local physio-chemical action of hydrated silica upon the pulmonary tissue, causing progressive lymphatic fibrosis, and rendering the tissue susceptible to tuberculosis. The disease is slow in evolution, requiring usually a number of years of exposure. It occurs principally among granite workers, sand blasters, metal miners, metal polishers, potters, and millstone workers.

Lead Poisoning

Lead poisoning is the most insidious and most common of all industrial diseases. It occurs principally among lead workers and smelters, lead miners, potters, painters, typesetters, stereotypers, plumbers, and workers with glass, gold and silver. Lead, in practically all forms, is a cumulative poison which is absorbed by way of the blood stream, chiefly from the respiratory tract, but also from the digestive tract and from the skin. The effect may be either an acute or chronic poisoning. The principal symptoms are colic, constipation, anemia, headache, anorexia, a bluish line along the edges of the gums, rheumatic pains, and, in extreme conditions, paralysis, blindness, insanity, and death.

It has been found[6] that 2 mg per day is the smallest dose, by inhalation, which in the course of years may result in lead poisoning. Regular inhalation during the usual working hours of air containing less than 0.2 mg of lead per cubic meter does not seem to produce serious lead poisoning in individuals of representative industrial groups[7].

Prevention

The prevention of industrial hazards from dusts and poisonous gases is largely a ventilation problem consisting of keeping the impurities in air down to a safe concentration. As yet there are no generally accepted standards on which to base the design of the ventilation equipment. Approximate data on the toxicity of various gases and fumes met with in industrial establishments are given in Table 1. Column 5, giving the maximum allowable concentrations for prolonged exposures, was compiled from experiments in which most exposures lasted not more than a

[6]Lead Poisoning, by Thomas Morrison Legge (*Journal Royal Society Arts*, 1929, Vol 77, p 1023)

[7]What is a Dangerous Quantity of Lead Dust in Air, by C. M Salls (*Industrial Hygiene Bulletin*, New York State Department of Labor, 1925)

week, and it is reasonable to assume that over more prolonged exposures such concentrations would cause pernicious effects.

Much is known concerning the physiological and pathological effects induced by various types and concentrations of atmospheric pollutants. In the absence of an accepted standard for safe breathing, and because of the slow, cumulative effects of certain kinds of air contaminants, the best procedure is the periodic medical examination of individuals, and the

TABLE 1. TOXICITY OF GASES AND FUMES IN PARTS PER 10,000 PARTS OF AIR[a]

VAPOR OR GAS	RAPIDLY FATAL	MAXIMUM CONCENTRATION FOR FROM ½ TO 1 HOUR	MAXIMUM CONCENTRATION FOR 1 HOUR	MAXIMUM ALLOWABLE FOR PROLONGED EXPOSURE
Carbon monoxide	40	15–20	10	1
Carbon dioxide	800–1000	-------	-------	-------
Hydrocyanic acid	30	1½	½	⅕
Ammonia	50–100	25	3	1
Hydrochloric acid gas	10–20	½	-------	1/10
Chlorine	10	½	-------	1/100
Hydrofluoric acid gas	2	1/10	-------	1/33
Sulphur dioxide	4–5	½–1	-------	1/10
Hydrogen sulphide	10–30	5–7	2–3	1
Carbon bisulphide	-------	11	5	½
Phosphene	20	4–6	1–2	-------
Arsine	2½	½	½	-------
Phosgene	Over ¼	¼	-------	1/100
Nitrous fumes	2½–7½	1–1½	-------	⅓
Benzene	190	-------	31–47	1½–3
Toluene and xylene	190	-------	31–47	-------
Aniline	-------	-------	1–1½	1/10
Nitrobenzene	-------	-------	1/100	1/500
Petrol	243	100–220	-------	-------
Carbon tetrachloride	480	240	40	16
Chloroform	250	140	50	2
Tetrachlorethane	73	-------	-------	1½
Trichlorethylene	370	-------	-------	-------
Methyl chloride	1500–3000	200–400	70	5–10
Methyl bromide	200–400	20–40	10	2
Lead vapor	-------	-------	-------	5–6

[a]Original data compiled by Y Henderson and H. Haggard (See *Noxious Gases*, 1927) Data revised by T M Legge. (See *Lessons Learned from Industrial Gases and Fumes, Institute of Chemistry of Great Britain and Ireland*, London, 1930)

routine measurement and study of the concentration and the physical and chemical characteristics of the dusts to which those individuals are exposed.

SMOKE AND AIR POLLUTION ABATEMENT

Successful abatement of atmospheric pollution requires the combined efforts of the combustion engineer, the public health officer, and the public itself. The complete electrification of industry and railroads, and the separation of industrial and residential communities would aid materially in the effective solution of the problem.

In the large cities where the nuisance from smoke, dust and cinders is

the most serious, limited areas obtain some relief by the use of district heating. The boilers in these plants are of large size designed and operated to burn the fuel without smoke, and some of them are equipped with dust catching devices. The gases of combustion are usually discharged at a much higher level than is possible in the case of buildings that operate their own boiler plants.

In general, time, temperature and turbulence are the essential requirements for smokeless combustion. Anything that can be done to increase any one of these factors will reduce the quantity of smoke discharged. Especial care must be taken in hand-firing bituminous coals (See Chapter 27.)

Checker or alternate firing, in which the fuel is fired alternately on separate parts of the grate, maintains a higher furnace temperature and thereby decreases the amount of smoke.

Coking and firing, in which the fuel is first fired close to the firing door and the coke pushed back into the furnace just before firing again, produces the same effect. The volatiles as they are distilled thus have to pass over the hot fuel bed where they will be burned if they are mixed with sufficient air and are not cooled too quickly by the heat-absorbing surfaces of the boiler.

Steam or compressed air jets, admitted over the fire, create turbulence in the furnace and bring the volatiles of the fuel more quickly into contact with the air required for combustion. These jets are especially helpful for the first few minutes after each firing. *Frequent firings* of small charges shorten the smoking period and reduce the density. *Thinner fuel beds* on the grate increase the effective combustion space in the furnace, supply more air for combustion, and are sometimes effective in reducing the smoke emitted, but care should be taken that holes are not formed in the fire. A *lower volatile coal* or a higher gravity oil always produces less smoke than a high volatile coal or low gravity oil used in the same furnace and fired in the same manner.

The installation of more modern or better designed fuel burning equipment, or a change in the construction of the furnace, will often reduce smoke. The installation of a Dutch oven which will increase the furnace volume and raise the furnace temperature often produces satisfactory results.

In the case of new installations, the problem of smoke abatement can be solved by the selection of the proper fuel-burning equipment and furnace design for the particular fuel to be burned and by the proper operation of that equipment. Constant vigilance is necessary to make certain that the equipment is properly operated. In old installations the solution of the problem presents many difficulties, and a considerable investment in special apparatus is necessary.

Legislative measures at the present time are largely concerned with the smoke discharged from the chimneys of boiler plants. Practically all of the ordinances limit the number of minutes in any one hour that smoke of a specified density, as measured by comparison with a Ringelmann Chart (Chapter 43), may be discharged.

These ordinances do not cover the smoke discharged at low levels by automobiles, and, although they have been instrumental in reducing the

smoke emitted by boiler plants, they have, in many instances, increased the output of chimney dust and cinders due to the use of more excess air and to greater turbulence in the furnaces.

Legislative measures in general have not as yet covered the noxious gases, such as sulphur dioxide and sulphuric acid mist, which are discharged with the gases of combustion. Where high sulphur coals are burned, these sulphur gases present a serious problem.

DUST AND CINDERS

The impurities in the air other than smoke come from so many sources that they are difficult to control. Only those which are produced in large quantities at a comparatively few points, such as the dust, cinders and fly-ash discharged to the atmosphere along with the gases of combustion from burning solid fuel, can be readily controlled.

Dusts and cinders in flue gas may be caught by various devices on the market, such as fabric filters, dust traps, settling chambers, centrifugal separators, electrical precipitators, and gas scrubbers, described in later paragraphs.

The cinder particles are usually larger in size than the dust particles; they are gray or black in color, and are abrasive. Being of a larger size, the range within which they may annoy is limited.

The dust particles are usually extremely fine; they are light gray or yellow in color, and are not as abrasive as cinder particles. Being extremely fine, they are readily distributed over a large area by air currents.

The nuisance created by the solid particles in the air is dependent on the size and physical characteristics of the individual particles. The difficulty of catching the dust and cinder particles is principally a function of the size and specific gravity of the particles.

Lower rates of combustion per square foot of grate area will reduce the quantity of solid matter discharged from the chimney with the gases of combustion. The burning of coke, coking coal, and sized coal from which the extremely fine coal has been removed will not as a general rule produce as much dust and cinders as will result from the burning of non-coking coals and slack coal when they are burned on a grate.

Modern boiler installations are usually designed for high capacity per square foot of ground area because such designs give the lowest cost of construction per unit of capacity. Designs of this type discharge a large quantity of dust and cinders with the gases of combustion, and if pollution of the atmosphere is to be prevented, some type of catcher must be installed.

Dust and Cinder Catchers [8]

The various types of dust and cinder catchers available today can be divided into six general classes:

1. Settling chambers.
2. Dust and cinder traps.
3. Centrifugal separators.

[8]Smoke and Dust Abatement, by M D Engle (A.S.H.V.E TRANSACTIONS ,Vol. 37, 1931)

4. Electrostatic precipitators
5. Gas scrubbers.
6. Fabric filters.

The selection of the proper type of catcher calls for a careful study of the material to be caught and the draft and space available. After installation, constant vigilance is necessary to keep the catchers in proper working condition if satisfactory operation is to be obtained.

If possible, the dust or cinder catcher should be installed on the inlet side of the induced draft fans because the dust and cinders in the gases seriously erode the wheels of the fans, the inlet connections and the scrolls. Where the induced draft fans operate at high tip speeds and no catchers are installed, it is not uncommon for the fans to require major repairs within one year and complete replacement within five years.

Settling Chambers

Probably the oldest form of dust catcher is the settling chamber, which generally consists of a large-sized, gas-tight space into which the dust-laden gases are discharged before being delivered to the chimney. The velocity of the gas should be reduced to a point where the larger and heavier particles will be precipitated by gravity. For good operation, the velocity of the gas should be reduced to a maximum of 2 fps. The bottoms of the chambers should be provided with dump plates through which the collected dust can be removed. Because these chambers are not effective in removing the finer dust particles they have been practically superseded by smaller and less costly devices.

Traps, Catchers, Precipitators

Various types of traps have been devised. In general they all depend upon breaking the gas up into thin strata and subjecting those thin strata to several abrupt changes in direction. The dust is thrown out of the gas stream into specially shaped pockets, or impinged against a roughened surface. The trapping pockets are drained into a hopper below with a small quantity of gas and the dust settles out by gravity due to the low velocity in the hopper. In the roughened surface type, various sections of the trap are closed off at intervals by means of dampers and the dust is shaken off the roughened surface into a hopper below.

These devices work very well in catching large size dust and cinders and trap much of the fine dust. They have been used most extensively on stoker-fired installations. They have the advantages of low pressure drop, relatively small space requirements, and low first cost.

Centrifugal catchers obtain separation by projecting the particles tangentially out of the gas stream. The effectiveness of this type of catcher varies directly as the specific weight of the dust and as the square of the tangential velocity, and inversely as the radius of rotation.

Electrostatic precipitators are used for catching fine dust. These precipitators consist of dust-tight chambers in which are suspended reinforced concrete slabs on about 10-in. centers. Between the slabs are suspended bare metal rods. High-voltage undirectional current is applied to the reinforcing rods in the concrete slabs acting as positive electrodes, the bare rods acting as negative electrodes. The dust-laden

gas flows horizontally through the precipitator and the dust particles migrate toward the concrete slabs to which they adhere and then fall or are scraped off into the dust hoppers below.

Gas Scrubbers

Wet scrubbers have been used for many years for removing dust from gases. A number of different types of scrubbers are now being built for removing dust from boiler flue gases. One type depends upon saturating the gas and washing the dust out of suspension by a spray of water. For best results with this type, the water should be atomized into as fine a spray as possible.

Another type depends upon splitting the gas into thin strata and subjecting these strata to a number of abrupt changes in direction, throwing the dust against the wet surfaces. The main problem in developing a satisfactory wet dust catcher is to find suitable materials of construction that will resist the corrosive action of the wash water for a reasonable length of time.

Fabric Filters

Filters of many kinds have been used with variable success. The filter bags are made of cotton, wool or asbestos fabric. The fabrics used in these filters do not withstand the temperatures at which gases are usually discharged from the boilers, and hence the gases must be cooled by some means. Surface coolers or water sprays can be used for reducing the gas temperatures.

One of the serious objections to all of these dust catchers is the relatively high cost of installation and maintenance, and the space required for installation.

Disposal of Dust and Cinders

Even after the dust and cinders have been caught, the disposal of the material caught presents a serious problem. The cinders discharged with the gases from stoker-fired boilers are usually very high in carbon and contain from 50 to 80 per cent as much heat per pound as the coal which is being burned. It is possible, and usually economical, to burn these cinders. They cannot be satisfactorily mixed with the coal in the stoker hopper but they can be blown into the furnace over the stoker fuel bed and burned satisfactorily. If a sufficient quantity of cinders is caught, a small unit pulverizer can be installed to prepare them for burning over the stoker fuel bed. The same pulverizer can be used for coal at times of peak load and will materially increase the capacity of the fuel-burning equipment for the boiler to which it is connected.

No satisfactory market has been developed for the dust caught from pulverized coal installations, but the possibilities are being investigated and it seems likely that in the future this material will have a market value that will go a long way toward paying the fixed charges on the cost of catching it.

The distribution of dust in the gas entering and leaving the dust and cinder catchers is not uniform and is different in practically every in-

stallation, and varies widely with changes in furnace conditions. In order to obtain a representative sample it is necessary to traverse the inlet and outlet of the catcher with a sampling tube which faces into the gas flow. The velocity of the gas into the sampling tube must be the same as the velocity of the gas in the duct at the instant the sample is taken. The swirls and eddy currents in the ducts make it difficult to obtain consistent readings, but if the test is conducted by some one of experience, an indication of the approximate efficiency can be obtained.

Nature's Dust Catcher

Nature has provided means for catching solid particles in the air and depositing them upon the earth. A dust particle forms the nucleus for each rain drop and the rain picks up dust as it falls from the clouds to the earth. In fact, without dust in the air to form the nuclei for rain drops it would never rain, and the earth would be continually enveloped in a cloud of vapor.

PROBLEMS IN PRACTICE

1 ● Classify the detrimental aspects of air pollution as it effects large industrial communities.

Air pollution may be classified (*a*) medical, as it affects the physiological functions of people, (*b*) botanical, as it affects vegetation, trees, plants, shrubs and flowers, and (*c*) physical, as it affects the discoloration and deterioration of buildings, and the nuisance of soiled interior furnishings, clothes, merchandise, etc.

2 ● Distinguish between dusts, fumes, and smokes.

Solid particles ranging in size from 1.0 micron to 150 microns are called *dusts* (micron = $\frac{1}{25,000}$ in).

Particles resulting from sundry chemical reactions and ranging from 0 1 to 1 0 micron in size are called *fumes.*

Carbon particles less than 0 1 micron in size which generally arise from the incomplete combustion of such materials as coal, oil, or tobacco are called *smokes*

3 ● What are some of the more important physical properties of these various groups of foreign bodies which are of importance in ventilation?

In slowly moving air, dusts tend to settle out by gravity without agglomerating to form larger particles, fumes have the tendency to form larger particles which will settle when they attain the size of approximately 1.0 micron; while smokes tend to diffuse and remain in the air as permanent impurities.

4 ● Why is atmospheric pollution an important engineering problem?

a. Certain impurities, when present in too great concentrations, cause ill health or even death.

b. High concentrations of solids occlude solar radiations.

c. Some materials cause permanent injury to parts of buildings, as sulphur fumes corrode exposed metal.

d. Extra cleaning expense is incurred in dusty localities

e. Internal combustion engines are damaged by abrasive dusts

5 ● How may the hazards of dust-producing industrial operations best be curtailed?

By providing mechanical exhaust ventilation sufficient to keep dust concentration at a safe level (see Table 1) and then removing foreign bodies to reduce the pollution of outside air.

6 ● How may the pollution of the atmosphere be lessened?

By compelling industrial plants to install dust catching and smoke controlling devices. In many cities the domestic heating plant is one of the most serious offenders, but these plants are too small to justify the installation of dust catchers. Public education in improved firing methods would be of considerable help in this field

7 ● Compare the dry and wet types of dust catchers.

The dry types are very effective in removing the larger dust particles but the smaller particles generally pass through other kinds than the electric precipitator. The dry types also require considerable space and therefore sometimes introduce resistance to the flow of air. The wet types are effective in removing some of the smaller dusts and the water-soluble gases The principal disadvantage of the washer is its short life caused by the corrosive action of the wash water.

8 ● What size particles are detrimental to health?

While fairly large particles may enter the upper air passages, those found in the lungs are seldom more than 10 microns in size, and comparatively few of them are more than 5 microns. It is agreed that particles between ½ and 2 microns may be harmful; some authorities place the upper limit at about 5 microns, and some incline to extend the lower limit to 0 1 of a micron.

9 ● Is the shape of the particle of any significance?

Hard particles with sharp corners or edges have a cutting effect on the delicate mucous membranes of the upper respiratory tract which may lower the resistance of the nose and throat to acute infections. This is aggravated by the irritating effects of some chemical compounds which may be taken in with the air and which act to reduce resistance.

10 ● What are the principal meteorological effects of smoke and dust?

a. The reduction in the amount of light received Measurements have shown that visible light may be as much as 50 per cent less intense in a smoky section of a city than in a section that is free from smoke Ultra-violet light is reduced as much or more, and in some cases is cut out entirely for a time.

b. Smoke and dust aid in the formation and prolongation of fogs. City fogs accumulate smoke and become darker in color and very objectionable. The sun requires a longer time to disperse them, and when the water is evaporated, there is a rain of smoke and soot particles that have been entrained.

11 ● Why has not smoke abatement been more effective?

Because communities have not been made sufficiently aware of the possibilities of burning high volatile fuels smokelessly and of separating cinder and ash from the stack gases to a degree that will prevent a nuisance

12 ● Is the abatement of dust and cinders important?

Yes. Only a small percentage of the solid emission from stacks is smoke, in the accepted popular sense; the remainder is fly-ash and cinders. While black smoke is disagreeable and its tarry matter and carbon particles soil anything with which they come in contact, the cinders and some of the ash are hard and destructive. They also, together with dusts from industrial processes, make up the hard, sharp, irritating, air-borne solids that are breathed by individuals not working in a dusty mill or factory.

13 ● Are air-borne impurities causative factors in hay fever, bronchial asthma, and allergic disorders?

Yes. Recent medical investigations indicate that 90 per cent of seasonal hay fever and 40 per cent of bronchial asthma are caused by air-borne pollens, tree dusts, and other allergic irritants.

14 ● Name some essential requirements for the smokeless combustion of fuels.

Time, temperature, and turbulence. A study of these factors is usually of value in overcoming a smoke nuisance.

15 ● What is the Ringelmann Chart Method of comparing smoke densities?

See Chapter 43 The Ringelmann Chart consists of four cards ruled with lines having different degrees of blackness These cards, together with a white card and a black one, are hung in a horizontal row 50 ft from the observer At this distance the lines become invisible and the cards appear to be different shades of gray, ranging from white to black The observer, by matching the cards against the shades of smoke coming from a stack, is able to estimate the blackness of the smoke as compared with the chart

Chapter 16

AIR CLEANING DEVICES

Air Cleaner Requirements, Classifications, Air Washers and Scrubbers, Viscous Type Filters, Unit Filters, Automatic Filters, Dry Air Filters, Air Filter Installations

THE removal of impurities from air brought into a building, or from air recirculated in a building for ventilating or air conditioning purposes is the function of any air cleaning or filtering device. These impurities include carbon (soot) from the incomplete combustion of fuels burned in furnaces and automobile engines, particles of earth, sand, ash, automobile tires, leather, animal excretion, stone, wood, rust and paper, threads of cotton, wool and silk, bits of animal and vegetable matter, bacteria and pollen. Microscopic examination shows that the character of the impurities varies with the locality, but as a rule carbon forms the greater part of them while the total is somewhat proportional to the state of industrial activity and the wind intensity. Additional information on sources of air pollution will be found in Chapter 15.

Observations have shown that the largest percentage of atmospheric impurities are less than 5 microns in size. (One micron equals 0.001 millimeter or approximately 0.00004 in.) The size and composition of each individual particle determines its buoyancy and consequently the length of time it will remain in suspension. The chart, Fig. 1, shows graphically the sizes of impurities found in the air, and other related data.

To estimate the probable dust load for air filter installations, the following approximate averages of atmospheric dust concentration may be used (7000 grains equal 1 lb):

Rural and suburban districts	0.2 to 0.4 grains per 1000 cu ft
Metropolitan districts	0.4 to 0.8 grains per 1000 cu ft
Industrial districts	0.8 to 1.5 grains per 1000 cu ft

AIR CLEANER REQUIREMENTS

To fulfill the essential requirements of clean air, an air cleaner should:

1 Be efficient in the removal of harmful and objectionable impurities in the air, such as dust, dirt, pollens, bacteria.

2 Be efficient over a considerable range of air velocities

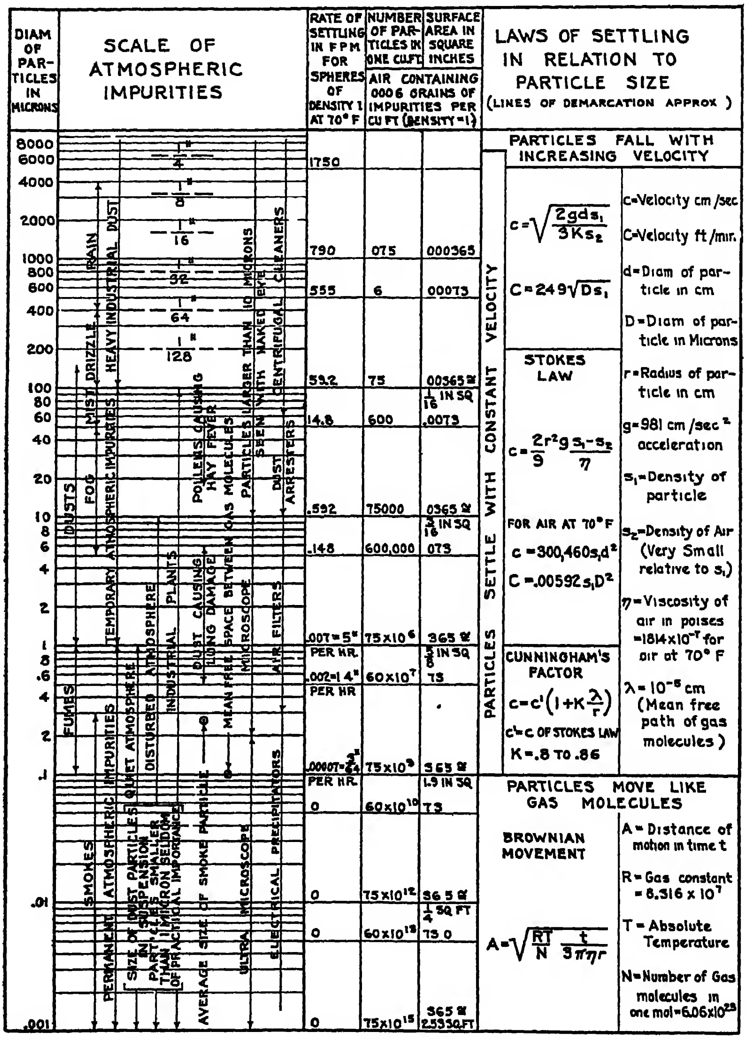
3 Have a low frictional resistance to air flow; that is, the pressure drop across the filter, measured in inches of water, should be as low as possible.

4. Have a large dust-holding capacity without excessive increase of resistance, or have ability to operate so as to keep the resistance constant automatically.

5. Be easy to clean and handle, cleans itself automatically, or else be inexpensive enough to replace when dirty.

6. Leave the air passing through the cleaner free from entrained moisture or charging liquids used in the cleaner

The AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS has developed a code[1] which explains how such devices are rated by (1) capacity in cubic feet of air handled per minute, (2) resistance in inches



Compiled by W. G. Frank and Copyrighted.

FIG 1. SIZES AND CHARACTERISTICS OF AIR-BORNE SOLIDS

of water at rated capacity, (3) dust arrestance, the percentage relationship expressing dust removal efficiency at rated capacity, (4) reconditioning power, the energy necessary to operate the mechanism of

[1]A S.H V.E Standard Code for Testing and Rating Air Cleaning Devices Used in General Ventilation Work (A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933)

an automatic air cleaning device, and (5) dust holding capacity, the amount by weight of standard dust which a non-automatic air cleaning device will retain before reconditioning is necessary.

CLASSIFICATION OF AIR CLEANERS

According to the Code, the following four classifications are given the devices:

Class A. Automatic Type: In general all air cleaning devices which use power to automatically recondition the filter medium and maintain a non-varying resistance to air flow.

Class B. Low Resistance Non-Automatic Type: Air cleaning devices for warm air furnaces, unit ventilating machines and similar apparatus and installations in which a maximum of not more than 0 18 in. water gage is available to move air through the air cleaning device

Class C. Medium Resistance Non-Automatic Type: Air cleaning devices for systems in which a maximum of not more than 0 5 in. water gage is available to move air through the air cleaning device.

Class D. High Resistance Non-Automatic Type: Air cleaning devices for the air intake of compressors, internal combustion engines, and the like, where a pressure of 1.0 in or more water gage is available to move air through the air cleaning device.

Air cleaners may also be classified as follows:

1. According to principle of air cleaning.
   a. Air washers.
   b. Viscous air filters.
      (1) Unit type.
      (2) Automatic type.
   c. Dry air filters.

2 According to application.
   a. For central fan systems of ventilation and air conditioning. Filters of the automatic or semi-automatic type, as well as the non-automatic viscous unit or dry type are usually recommended and are installed in a central plenum chamber.
   b. For unit ventilators. Filters of viscous unit or dry type, installed at inlet of individual units.
   c. For window installations. Self-contained units consisting of fan and filter, usually dry or viscous type, adapted to be placed in the ordinary window.
   d. For warm-air furnaces. Unit type viscous or dry filters placed in small plenum chamber of warm-air house heating systems.
   e For compressors and Diesel engines. Unit type viscous or dry filters, installed at air intake of compressors and Diesel engines.
   f. For compressed air lines. Unit type viscous or dry filters.

With the growing congestion of large cities and an industrial growth throughout the entire country, the percentages of foreign material in the air, such as soot or carbon, which are unaffected by an air washer type of air cleaner, have increased. This has brought about the development of the viscous and dry type air filters which are part of many ventilating and air conditioning systems.

AIR WASHERS AND SCRUBBERS

Scrubbers have not been used very extensively in the past for cleaning air for ventilating purposes. However, new types have been developed

which appear to have possibilities for cases where the air to be cleaned is extremely dirty or where a higher degree of cleanliness is desired than can be obtained with an air washer. Information on air washers will be found in Chapter 12

VISCOUS TYPE FILTERS

The principle of air cleaning used in viscous filters is that of *adhesive impingement*. Dust and dirt in the air, especially soot and carbons, are trapped and retained by successive impingements on coated surfaces. While the arrangements of filtering media and the kind of materials used are almost unlimited, there are certain rather definite requirements for a practical commercial filter.

Investigations in this country and abroad demonstrate that the first impingement of dust laden air on a viscous coated surface removes about 60 per cent of the dust, the next impingement takes 60 per cent of what then remains—that is, 24 per cent—and the next impingement removes 9.6 per cent. To secure maximum efficiency, it is necessary to divide the air into innumerable fine streams, as the more intimately and freely the air is brought into contact with the viscous-coated media the better will be the cleaning.

The binding liquid used with viscous filters should have the following properties:

1 Its surface tension should be such as to produce a homogeneous film-like coating on the filter medium.

2. The viscosity should vary only slightly with normal changes of temperature

3 It should be germicidal in its action to prevent the development of mold spores and bacteria on the filter media

4. The liquid should have a high affinity for dust at low temperatures

5 The liquid should have high capilarity, or ability to wet and retain the dust

6 Evaporation should not exceed 1 per cent.

7 It should be fireproof.

8 It should be odorless.

Viscous Unit Filters

In the unit type viscous filter, the filtering media are arranged in units of convenient size to facilitate installation, maintenance, and cleaning. Each unit consists of an interchangeable cell or replaceable filter pad and a substantial frame which may be bolted to the frames of other like units to form a partition between the source of dusty air and the fan inlet. Where necessary reconditioning equipment should be installed near each group of unit filters, with hot water and sewer connections provided.

To secure greater dust holding capacity and a practically constant resistance and air volume, the filter media are usually placed in the direction of air flow, with progressively finer filter densities determined by the percentage of dust impinged. This arrangement provides relatively large spaces for the collection of dirt in the front of the filter where the bulk of the dust is taken out without undue increase in resistance, while at the back of the filter the openings are smaller to secure high efficiency in the removal of the finer dust particles.

The resistance of a well-designed unit filter of the adhesive impinge-

ment type usually depends upon the velocity at which the air is handled and upon whether the unit is clean or dirty. The cleaning efficiency of the unit is usually highest after it has accumulated a certain portion of its maximum load of dirt because some dust collected in the cell acts as an efficient medium for the further seizing of solids from the air. By periodically cleaning a predetermined number of cells, the resistance and capacity of a built-up filter may be held at any desired figure. The frequency of cleaning any unit filter installation depends upon the dust concentration

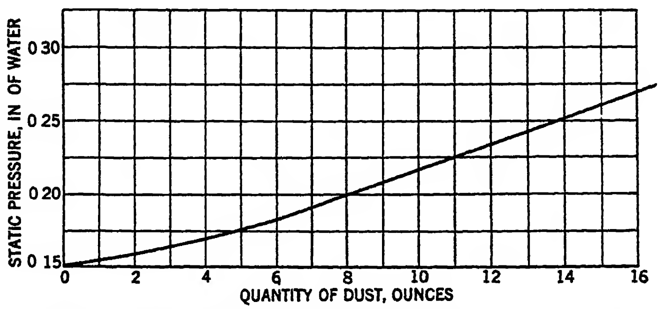


FIG. 2. CHART SHOWING CHANGE IN RESISTANCE DUE TO DUST ACCUMULATION

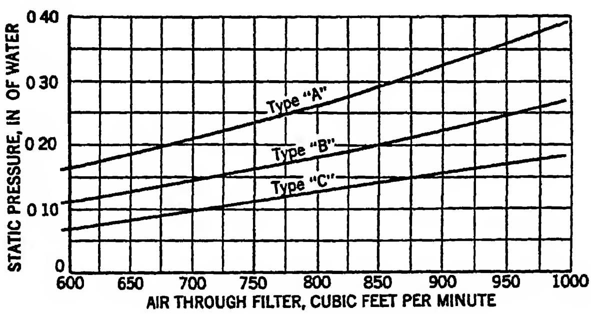


FIG. 3. RESISTANCE TO AIR-FLOW OF TYPICAL UNIT AIR FILTERS

of air being cleaned, and on the amount of dirt which can be accumulated in the filter medium without causing excessive resistance. (Figs. 2, 3 and 4.)

It is difficult to satisfactorily compare the cleaning efficiencies of various filter types unless the efficiency ratings are determined under laboratory conditions in accordance with some definite test procedure such as that developed by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS[2]. Efficiency tests made in the field with *atmospheric dust* are subject to so many variables that consistent comparisons are difficult. Of course there is no *standard atmospheric dust*, as atmospheric dust varies widely in composition and concentrations in different localities. Wide variations are also found due to different seasons of the year as well as

[2]Loc Cit. Note 1.

the time of day and the direction of the wind. A chart showing the increase in resistance of a unit filter of the viscous impingement type, when tested with the standard test dust described in the code[8], is given in Fig. 2. The resistance to air flow of three typical clean viscous impingement type filters having different media densities is shown in Fig. 3. Type A is a dense pack used in bacteria control; Type B is a medium pack used for general ventilation work and Type C is a low resistance unit for use where low resistance is the important factor and maximum cleaning efficiencies are not essential. The operating characteristics which might be expected under various dust concentrations with air filters having different dust-holding capacities are illustrated in Fig. 4.

Filters consisting of inexpensive frames of cardboard or similar material filled with viscous-coated glass wool, steel wool or the like are available.

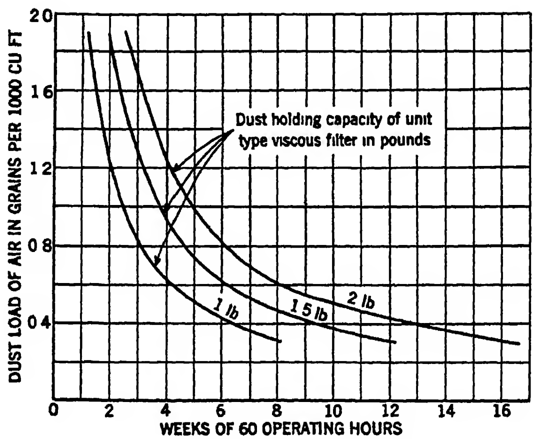


FIG. 4. MAINTENANCE CHART FOR UNIT TYPE VISCOUS FILTERS

Because of their construction these units may be discarded when dirty and replaced with new units at relatively little expense. They are used in general ventilation work and with warm air furnaces and other installations where low first cost and low resistance to air flow are essential. The operating characteristics of these units conform in general with those of the rigid frame type.

Viscous Automatic Filters

The principle of air cleaning used in the viscous automatic filters is the same as in the unit filters. The removal of the accumulated dust, however, is done automatically instead of by hand. The automatic cleaning and recoating of these filters is based on the principle that the viscous fluid itself will perform the cleaning function, thereby eliminating a separate washing agent. The dust collected by the filter thus is deposited finally in the bottom of the viscous fluid reservoir from which it may be removed by different methods, depending on the design of the filter.

[8]Loc Cit. Note 1.

There are three general types of automatic filters. They are differentiated from each other according to the process of self-cleaning and renewing of the viscous coating used by each type, as follows:

1. The filter medium has the form of an endless curtain suspended vertically, with its lower portion submerged in a viscous fluid reservoir. The curtain rotates slowly through this bath, thus performing the cleaning and recoating of the filter medium.

2. The filter screen is arranged in the form of shelves or cylinders, and the viscous fluid is flushed through all parts of the medium in a direction opposite to the air flow.

3. The filter medium is arranged vertically and is stationary. The viscous fluid is flushed from above over the medium, while the air flow is stopped.

The washing and renewing process in automatic filters usually is intermittent. It is accomplished by an electric motor or by other motive power and is controlled by manual or by automatic timing devices. The operating cycle is of a predetermined frequency and should be so timed as to insure a constant static pressure drop across the filter. The customary resistance to air flow is ⅜-in. water gage at an air velocity of 500 fpm, measured at the filter entrance. Automatic viscous filters are made up in units which are delivered either fully assembled or in parts to be assembled at the point of installation.

DRY AIR FILTERS

Dry air filters, in which dust is impinged upon or filtered through screens made of felt, cloth, or cellulose, are available in various types. These filters require no adhesive liquid, but depend on the straining or screening action of the filtering medium. Because of the close texture of the filtering media used in most of the dry filters, the surface velocity, or velocity of the air entering the media, ranges between 10 and 50 fpm, depending on the nature and texture of the fabric. This necessitates a relatively large screen surface, and the filter media are usually arranged in the form of pockets to bring the frontal area within customary space requirements.

As in viscous unit filters, an average constant resistance and air volume may be obtained by periodic reconditioning or renewal of the filter screens. Since some materials suitable for dry filtering media are affected considerably by moisture which tends to cause a rapid increase in resistance, they should be treated or processed to minimize the effect of changes in humidity.

Filters using felt and similar materials as filter media usually depend upon vacuum cleaning for reconditioning. A special nozzle, operated from a portable or stationary vacuum cleaner, is shaped to reach all parts of the filter pockets. Permanent filter media should be capable of withstanding repeated vacuum cleanings without loss in dust removal efficiency. While most dry filters are cleaned by replacing an inexpensive filter sheet, the useful life of these sheets often may be lengthened by vibrating or vacuum cleaning.

METHODS OF INSTALLATION

The published performance data for all air filters are based on *straight through* unrestricted air flow. Filters should be installed so that the face area is at right angles to the air flow whenever possible. Eddy currents

and dead air spaces should be avoided and air should be distributed uniformly over the entire filter surface, using baffles or diffusers if necessary.

The most important requirements of a satisfactory and efficiently operating air filter installation are:

1. The filter must be of ample size for the amount of air it is expected to handle An overload of 10 to 15 per cent is regarded as the maximum allowable When air volume is subject to increase, a larger filter should be installed

2. The filter must be suited to the operating conditions, such as degree of air cleanliness required, amount of dust in the entering air, type of duty, allowable pressure drop, operating temperatures, and maintenance facilities.

3. The filter type should be the most economical for the specific application. The first cost of the installation should be balanced against depreciation as well as expense and convenience of maintenance

The following recommendations apply to filters and washers installed with central fan systems:

1. Duct connections to and from the filter should change size or shape gradually to insure even air distribution over the entire filter area.

2. Sufficient space should be provided in front as well as behind the filter to make it accessible for inspection and service. A distance of two feet may be regarded as the minimum.

3 Access doors of convenient size should be provided in the sheet metal connections leading to and from the filters.

4. All doors on the clean air side should be lined with felt to prevent infiltration of unclean air. All connections and seams of the sheet metal ducts on the clean air side should be as air-tight as possible.

5. Electric lights should be installed in the chamber in front of and behind the air filter.

6. Air washers should, whenever possible, be installed between the tempering and heating coils to protect them from extreme cold in winter time.

7. Filters installed close to air inlet should be protected from the weather by suitable louvers, in front of which a large mesh wire screen should be provided.

8 Filters should have permanent indicators to give a warning when the filter resistance reaches too high a value.

REFERENCES

Determining the Efficiency of Air Cleaners, by A M Goodloe (A S.H V E TRANSACTIONS, Vol 30, 1924).

Design and Application of Oil-Coated Air Filters, by H. C. Murphy (A.S H.V E TRANSACTIONS, Vol 33, 1927)

A Study of Dust Determinators, by F. B Rowley and John Beal (A.S H V E TRANSACTIONS, Vol 34, 1928).

Determining the Quantity of Dust in Air by Impingement, by F. B. Rowley and John Beal (A S H V E TRANSACTIONS, Vol 35, 1929)

Operation and Maintenance of Air Filters, by W G Frank (*Heating, Piping and Air Conditioning*, May, 1931).

Size and Characteristics of Air-Borne Impurities, by W. G Frank (*Heating, Piping and Air Conditioning*, January, 1932).

Fundamental Principles in the Design of Dry Air Filters, by Otto Wechsberg (A.S H.V.E JOURNAL SECTION, *Heating, Piping and Air Conditioning*, April, 1933).

Testing and Rating of Air Cleaning Devices Used for General Ventilation Work, by Samuel R Lewis (A S H V E TRANSACTIONS, Vol 39, 1933)

The Dust Problem in Air Conditioning, by F. B Rowley (A S H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, June, 1935).

PROBLEMS IN PRACTICE

1 ● Assume a fan and duct system which handled 10,000 cfm through clean filters with a system resistance of 0.8 in. of water and that after the filters have become dirty the system resistance increases to 1.0 in. of water, and that the fan speed remains unchanged. Is there any way of predicting the volume of air delivered after the filter becomes dirty?

Yes. If the performance curves for the particular make of fan are available, the new volume may be determined from the resistance pressure curve. (Figs 1, 2, 3 and 4, Chapter 17)

2 ● What are the advantages of viscous filters?

The principal advantage of the viscous filter is its large dust holding capacity. The dust accumulation is distributed through the depth of the filtering medium rather than upon the surface as in the dry types, which makes it possible for viscous filters to handle heavy dust concentrations without excessive resistance Since its efficiency and resistance are based on maximum air velocities of from 300 to 500 fpm through the filter, the viscous filter consumes the minimum amount of space for a given air volume.

3 ● What are the advantages of dry filters?

Dry filters are more efficient in the removal of fine dust particles from the air, and some types will eliminate even as much as 60 per cent of the smoke particles. Dry filters also are easily and conveniently maintained by vacuum cleaning, vibrating, or renewing the filtering medium

4 ● If an air washer is used for cooling and humidity control in an air conditioning system, is a filter needed?

An air filter is desirable in conjunction with an air washer because of the large amount of soot in the air which, due to its greasy and amorphous nature, is not readily trapped in an air washer Filters should be placed between the washer and the air intake so that all the dirt will be collected at one point to simplify maintenance and to protect all the equipment in the system.

5 ● Is an air filter needed with an extended surface type heat exchanger?

An air filter is essential with an extended surface heat exchanger in order to maintain its efficiency, for without this protection dust particles will adhere to the exposed surfaces, and gradually build up a deposit to the point where the efficiency will be impaired and the resistance increased by restricting the air passage

6 ● What is the proper location of a filter in relation to the fan?

A filter will operate equally well whether placed on the suction or discharge side of the fan. It has become standard practice, however, to locate the filter on the fan inlet side because there it has: (1) simpler duct connections, (2) reduced static pressure losses, (3) more even air distribution over the entire filter area Where an exceptionally high efficiency in dust removal must be maintained, it is often advisable to place the filter on the discharge side of the fan so there can be no infiltration of unclean air.

7 ● What instruments and apparatus are required for determining the pollen concentration in air by means of the settling method?

A microscope with a field of known area and a glass slide coated with a viscous material

8 ● Describe the procedure for determining the pollen concentration in air by means of the settling method.

A glass slide coated with a viscous material is placed for a period of 24 hours in a horizontal position in the atmosphere to be tested The slide is then removed and placed under the microscope, and pollen counts are made of approximately 25 fields over the area of the glass slide Having determined the count over a definite area, as for example,

1 sq cm, and finding the settling rate of the average particles from the chart, Fig. 1, the concentration in parts per cubic yard can be calculated.

9 ● The resistance to air flow of a unit air filter is found to be 0.4 in. of water. The volume of air passing through the filter is 1000 cfm at a velocity of 200 fpm. What would be the filter area required in order to reduce the pressure drop across the filter from 0.4 in. of water to 0.16 in. of water?

Referring to Fig. 3: The resistance is substantially proportional to the square of the velocity, or

$$\frac{R_1}{R_2} = \frac{V_1^2}{V_2^2}$$

$$\frac{0.4}{0.16} = \frac{200^2}{V_2^2}$$

$$V_2^2 = 16{,}000$$

$$V_2 = 126.5 \text{ fpm}$$

$$Q = AV$$

$$1000 = 126.5\,A$$

$$A = \frac{1000}{126.5} = 7.91 \text{ sq ft}$$

The filter area would be increased from 5 sq ft to 7.91 sq ft.

10 ● A ventilating system complete with filters has a fan which, when operating at 400 rpm and delivering air at 1 in. of water total static pressure, requires an input of 3 horsepower. After the system operates for a time, the pressure drop across the filter caused by the clogging action of the collected dust and dirt increases from 0.1 in. of water to 0.4 in. of water. To maintain the original rate of air delivery with the increased static pressure, at what speed must the fan be run and what horsepower will be required?

Static pressure after clogging of filter $= 1 + (0.4 - 0.1) = 1.3$ in. of water.

The static pressure varies as the square of the fan speed. Therefore, if X is the fan speed after the static pressure increases

$$\frac{1.3}{1} = \left(\frac{X}{400}\right)^2$$

$$X = 456 \text{ rpm.}$$

The horsepower varies as the cube of the fan speed. Therefore, if Y is the horsepower after the static pressure increases:

$$\frac{Y}{3} = \left(\frac{456}{400}\right)^3$$

$$Y = 4.44 \text{ horsepower.}$$

To maintain the original rate of air delivery with the increased static pressure, the fan speed must be increased from 400 to 456 rpm, and the horsepower from 3 to 4.44.

Chapter 17

FANS

Classification, Performance, Fan Efficiency, Characteristic Curves, System Characteristics, Selection of Fans, Controls, Fan Designations, Motive Power

IN heating and ventilating practice, fans are used to produce air flow except where positive displacement is required, in which case compressors or rotary blowers are used. Fans are classified according to the direction of air flow as (1) *axial flow* or *propeller* type if the flow is parallel with the axis, and (2) *radial flow* or *centrifugal* type if the flow is parallel with the radius of rotation.

Axial flow fans are made with various numbers of blades of a variety of forms. The blades may be of uniform thickness (sheet metal), either flat or cambered, or may be of varying thickness of so-called aerofoil section (airplane propeller type). Where an axial flow fan is intended for operation at comparatively high pressures the hub sometimes is enlarged in the form of a disc and the fan is known as a *disc fan.*

Radial flow or *centrifugal fans* include steel plate fans, pressure blowers, cone fans, and the so-called multiblade fans. All the foregoing types have variations which may be obtained by modification of the proportions or change in the curvature and angularity of the blades. The angularity of the blades determines the operating characteristics of a fan; a forward curved blade is found in a fan having slow speed operating characteristics, while a backward curved blade is found in a fan having high speed operating characteristics.

A wide variation exists in the demands which have to be met by fan installations. A fan may be required to move large quantities of air against little or no resistance or it may be required to move small quantities against high resistances. Between these two extremes innumerable specific requirements must be met. In general, fans of all types in each general class can be made to perform the same duty, although mechanical difficulties, noise or lack of efficiency may limit the use to one or another type. The most common field of service for fans of the propeller type is in moving air against moderate resistances, especially where no long ducts or heavy friction must be overcome and where noise is not objectionable, whereas centrifugal fans are commonly employed for operation at the comparatively higher pressures and where extreme quietness is necessary.

FAN PERFORMANCE

Fans of all types follow certain laws of performance which are useful in determining the effect of changes in the conditions of operation. These

laws apply to installations comprising any type of fan, any given piping system and constant air density, and are as follows:

1. The air capacity varies directly as the fan speed.
2. The pressure (static, velocity, and total) varies as the square of the fan speed.
3. The power demand varies as the cube of the fan speed.

Example 1. A certain fan delivers 12,000 cfm at a static pressure of 1 in of water when operating at a speed of 400 rpm and requires an input of 4 hp. If in the same installation 15,000 cfm are desired, what will be the speed, static pressure, and power?

$$\text{Speed} = 400 \times \frac{15{,}000}{12{,}000} = 500 \text{ rpm}$$

$$\text{Static pressure} = 1 \times \left(\frac{500}{400}\right)^2 = 1.56 \text{ in}$$

$$\text{Power} = 4 \times \left(\frac{500}{400}\right)^3 = 7.81 \text{ hp}$$

When the density of the air varies the following laws apply:

4. At constant speed and capacity the pressure and power vary directly as the density.

Example 2. A certain fan delivers 12,000 cfm at 70 F and normal barometric pressure (density 0 07492 lb per cubic foot) at a static pressure of 1 in. of water when operating at 400 rpm, and requires 4 hp. If the air temperature is increased to 200 F (density 0 06015 lb) and the speed of the fan remains the same, what will be the static pressure and power?

$$\text{Static pressure} = 1 \times \frac{0.06015}{0.07492} = 0\ 80 \text{ in}$$

$$\text{Power} = 4 \times \frac{0.06015}{0.07492} = 3.20 \text{ hp}$$

5 At constant pressure the speed, capacity and power vary inversely as the square root of the density.

Example 3. If the speed of the fan of Example 2 is increased so as to produce a static pressure of 1 in. of water at the 200 F temperature, what will be the speed, capacity, and power?

$$\text{Speed} = 400 \times \sqrt{\frac{0.07492}{0.06015}} = 446 \text{ rpm}$$

$$\text{Capacity} = 12{,}000 \times \sqrt{\frac{0.07492}{0.06015}} = 13{,}392 \text{ cfm (measured at 200 F)}$$

$$\text{Power} = 4 \times \sqrt{\frac{0.07492}{0.06015}} = 4.46 \text{ hp}$$

6 For a constant weight of air.

(*a*) The speed, capacity, and pressure vary inversely as the density

(*b*) The horsepower varies inversely as the square of the density

Example 4. If the speed of the fan of the previous examples is increased so as to deliver the same weight of air at 200 F as at 70 F, what will be the speed, capacity, static pressure, and power?

$$\text{Speed} = 400 \times \frac{0\ 07492}{0\ 06015} = 498 \text{ rpm}$$

$$\text{Capacity} = 12{,}000 \times \frac{0.07492}{0\ 06015} = 14{,}945 \text{ cfm (measured at 200 F)}$$

$$\text{Static pressure} = 1 \times \frac{0.07492}{0.06015} = 1.25 \text{ in}$$

$$\text{Power} = 4 \times \left(\frac{0.07492}{0.06015}\right)^2 = 6.20 \text{ hp}$$

FAN EFFICIENCY

The efficiency of a fan may be defined as the ratio of the horsepower output to the horsepower input.

The horsepower output is expressed by the formula:

$$\text{Air Horsepower}^1 = \frac{\text{cfm} \times \text{total pressure in inches of water}}{6356} \tag{1}$$

When the static pressure is used in the computation it is assumed that this represents the useful pressure and that the velocity pressure is lost in the piping system and in the air which leaves the system. Since in most installations a higher velocity exists at the fan outlet than at the point of delivery into the atmosphere, some of the velocity pressure at the fan outlet may be utilized by conversion to static pressure within the system, but owing to the uncertainty of friction losses which occur at the places where changes in velocity take place, the amount of velocity pressure which is actually utilized is seldom known, and the static pressure alone may best represent the useful pressure.

The efficiency based upon static pressure is known as the static efficiency and may be expressed as follows.

$$\text{Static efficiency}^1 = \frac{\text{cfm} \times \text{static pressure in inches of water}}{6356 \times \text{Horsepower input}} \tag{2}$$

Different fans may develop the same capacity against the same static pressure and with the same power input, and therefore operate at the same static efficiency, while maintaining different outlet velocities. Where a high outlet velocity is desirable or can be utilized effectively, the static efficiency fails to be a satisfactory measurement of the performance. In many applications of propeller fans, air is circulated without encountering resistance and no static pressure is developed. The static efficiency is zero and its calculation is meaningless. Because of such situations where the static efficiency fails to indicate the true performance, many engineers prefer to base the calculation of efficiency upon the total or dynamic pressure. This efficiency is variously known as the total, dynamic, or mechanical efficiency, and may be expressed as follows:

$$\text{Mechanical or Total efficiency}^1 = \frac{\text{cfm} \times \text{total pressure in inches of water}}{6356 \times \text{Horsepower input}} \tag{3}$$

CHARACTERISTIC CURVES

In the operation of a fan at a fixed speed the static and total efficiencies vary with any change in the resistance which is imposed. With different designs the peak of efficiency occurs when the fans deliver different per-

[1]See Standard Test Code for Disc and Propeller Fans Centrifugal Fans and Blowers, Edition of 1932

centages of their wide-open capacity. Variations in efficiency accompany variations in pressures and power consumption which are characteristic of the individual designs and which are influenced particularly by the shape and angularity of the blades. Such variations in pressure, power, and efficiency are shown by characteristic curves.

Characteristic curves of fans are determined by tests performed in accordance with the Standard Test Code for Disc and Propeller Fans, Centrifugal Fans and Blowers[2] as adopted by the American Society of Heating and Ventilating Engineers and the *National Association of Fan Manufacturers*. The results of tests are plotted in different ways: the

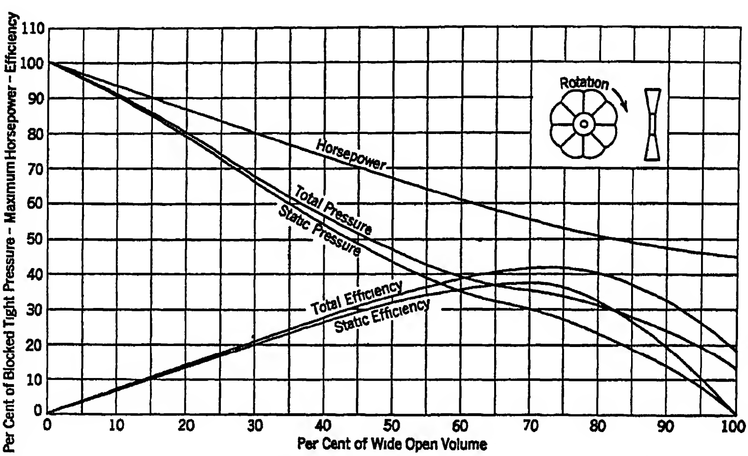


Fig. 1. Operating Characteristics of an Axial Flow Fan

abscissae may be the ratio of delivery, assuming full open discharge as 100 per cent, and the ordinates may be static pressure, dynamic pressure, horsepower and efficiency. Pressures may be expressed in per cent of the maximum pressure in the manner shown in the illustrations in this chapter, but in engineering calculations they are sometimes expressed in proportion to the pressures due to the peripheral velocity.

It should be noted that characteristic curves of fan performance are plotted for a constant speed. Some variation in values of efficiency may occur at different speeds but such variation is usually slight within a wide range of speeds. Fans of similar design but of different size will also show some difference in efficiency. Figs. 1 to 4 show characteristic curves for different types of fans using blades of various shapes, but without reference to the design of housing employed. The efficiency curves are therefore not serviceable for making rigid comparisons of efficiencies obtainable with blades of the various shapes but are intended merely to show reasonable values and more particularly to show the manner in which variations occur with changes in fan capacity.

[2]A.S H.V.E. Transactions Vol. 29, 1923. Amended June, 1931.

Axial flow fan characteristics are indicated by Figs. 1 and 2. These fans, when properly designed, have a satisfactory efficiency at low resistance, comparing favorably in this respect with centrifugal fans. They are low in cost and economical in operation and occupy relatively little space. Although this type of fan can operate against considerable resistance, the noise often becomes objectionable, so that it does not always compare favorably with centrifugal fans for such service. With most of the designs which employ blades of uniform thickness the power increases rapidly with an increase in resistance.

The curves (Fig. 1) show the rapid reduction in capacity and increase in power as the resistance increases The low efficiency when overcoming

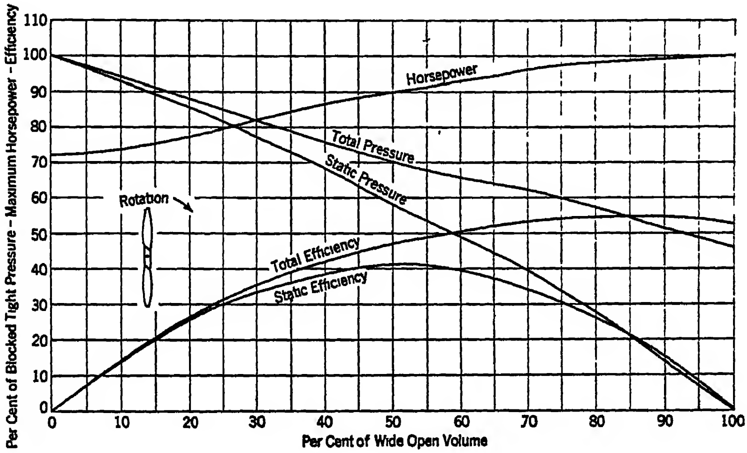


FIG. 2. OPERATING CHARACTERISTICS OF AN AIRPLANE PROPELLER FAN

heavy resistance is due to the low speed of the blades near the hub as compared to the relatively high peripheral or tip speed. The air driven by the blade area near the rim can pass back through the less effective blade area at the hub more easily than it can overcome the duct resistance.

Fig. 2 shows the performance of the *airplane propeller fan* in which the blades are similar in shape to those of an airplane propeller but of varying number according to the pressure to be developed. This fan usually operates at a higher speed than does the former type of propeller fan, and with a different power characteristic, the power remaining fairly constant throughout the range of pressures, being somewhat less at the higher than at the lower pressures. The flatness of the horsepower curve indicates the advantage of this type of fan in preventing overloading of motors where fluctuations in pressure occur. Variations in the diameter, width, pitch, camber, and the thickness of the blades provide a considerable degree of flexibility in design, so that the peak of total efficiency may be made to occur at wide-open volume or at various percentages of that volume.

Another advantage of this type of axial flow fan is its low resistance to air passage when standing still. There are some installations in which such a characteristic is desirable.

The straight blade (paddle-wheel) or partially backward curved blade type of fan is practically obsolete for ventilation. Its use is largely confined to such applications as conveyors for material, or for gases containing foreign material, fumes and vapors. The open construction and the few large flat blades of these wheels render them resistant to corrosion and tend to prevent material from collecting on the blades. This type of fan has a good efficiency, but the power steadily increases as the static

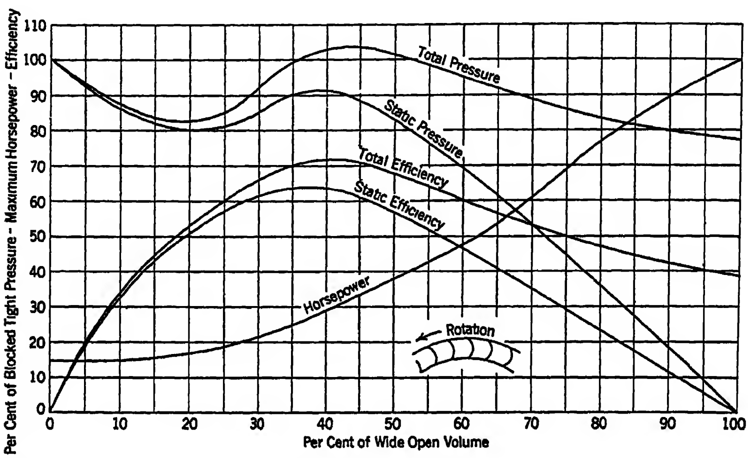


Fig. 3 Operating Characteristics of a Fan with Blades Curved Forward

pressure falls off, which requires that the motor be selected with a moderate reserve in power to take care of possible error in calculation of duct resistance.

The *forward curved multiblade fan* is the type most commonly used in heating and ventilating work, as it has a low peripheral speed, a large capacity, and is quiet in operation. The point of maximum efficiency for this fan occurs near the point of maximum static pressure. The static pressure drops consistently from the point of maximum efficiency to full open operation. The power curve rises continually from low to peak capacity, but if reasonable care is exercised in figuring resistance there is no danger of overloading the motor.

The outstanding characteristics of the *full backward curve multiblade type fan* are the steep pressure curves, the non-overloading power curve, and the high speed. (See Fig. 4.) This fan operates at a peripheral speed of approximately 250 per cent of the forward curve multiblade type for like results. The pressure curves begin to drop at very low capacity and continue to fall rapidly to full outlet opening. The steep pressure curves tend to produce constant capacity under changing pressures. Where wide fluctuations in demand occur, this type of fan is desirable to prevent overloading of motors. The maximum power requirement occurs at about the maximum efficiency. Consequently a motor selected to carry the load at this point will be of sufficient capacity to drive the fan over its full range of capacities at a given speed. The high speed of this type

makes it adaptable for direct connected electric motor drives. The high speed may necessitate somewhat heavier construction and more operating attention or service. The dimensional bulk for a given duty often is 150 per cent of that of a forward curve multiblade type fan.

Between the extremes of the forward and the full backward curve blade type centrifugal fans a number of modified designs exist, differing in the

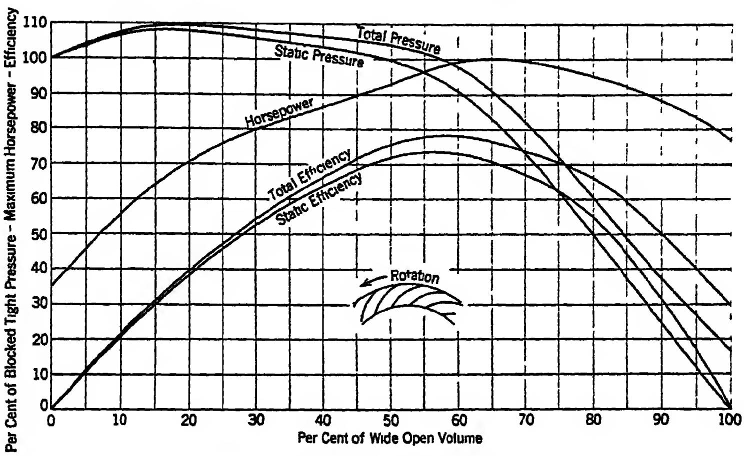


FIG. 4. OPERATING CHARACTERISTICS OF A FAN WITH BLADES CURVED BACKWARD

angularity or in the shape of the blades. Common among these designs are the straight radial blade type, the radial tip type, and the double curve blade type with a forward angle at the heel and a slight backward angle at the tip of the blade. Characteristic curves of these types show varying degrees of resemblance to the curves of Figs. 3 and 4, according to the degree of similarity to one or the other of the two designs of fan considered.

SYSTEM CHARACTERISTICS

A given fan performs as determined by the real characteristic of the system to which it is attached. When a different performance of a fan is desired, it is necessary to either change the speed of the fan (as *A* to *B* or *C* to *D* in Fig. 5), or to change the system (as by moving a damper from *A* to *C* in Fig. 5). If the speed of the fan is changed, the new point of operation is the intersection of the constant speed static pressure—cubic feet per minute curve for the new speed with the system characteristic. If the system is changed, the new point of operation is the intersection of the constant speed static pressure, cubic feet per minute curve with the new system characteristic.

Heating and ventilating systems follow the simple parabolic law quite closely but other types of systems follow some other more or less complex relation. The more complex systems can be separated into their component parts whose individual characteristics are known and the summation of the characteristics of the several parts of a system will give the composite characteristic of the system.

SELECTION OF FANS

The following information is required to select the proper type of fan:

1. Cubic feet of air per minute to be moved.
2. Static pressure required to move the air through the system.
3. Type of motive power available.
4. Whether fans are to operate singly or in parallel on any one duct.
5. What degree of noise is permissible.
6. Nature of the load, such as variable air quantities or pressures

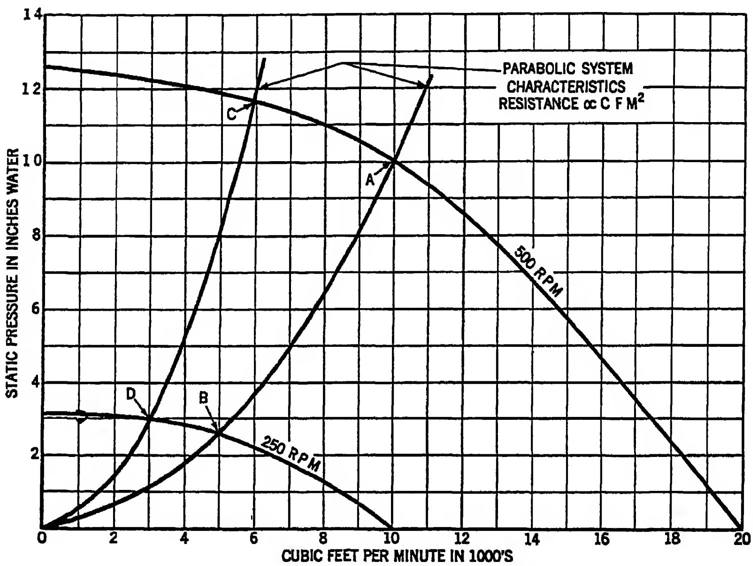


Fig. 5. Illustration of Operating Points of a Given Fan at Two Speeds on the Same and Different Systems

Knowing the requirements of the system, the main points to be considered for fan selection are (1) efficiency, (2) speed, (3) noise, (4) size and weight, and (5) cost.

In order to facilitate the choice of apparatus, the various fan manufacturers supply fan tables or curves which usually show the following factors for each size of fan operating against a wide range of static pressures:

1. Volume of air in cubic feet per minute (68 F, 50 per cent relative humidity, 0.07488 lb per cubic foot).
2. Outlet velocity.
3. Revolutions per minute.
4. Brake power.
5. Tip or peripheral speed.
6. Static pressure.

The most efficient operating point of the fan is usually shown by either bold-face or italicized figures in the capacity tables.

Fans for Ventilating and Air Conditioning Systems

Two important factors in selecting fans for ventilating systems are efficiency (which affects the cost of operation) and noise. First cost and space available are secondary. The fans should be selected to operate at maximum efficiency without noise. Because noise in a ventilating system is irritating and a cause for complaint, fans must be selected of proper size in order to reduce it to a minimum. Noise may be caused by other factors than the fan, namely, high velocity in the duct work, unsatisfactory location of the fan room, improper construction of floors and walls, and poor installation. Where noise is chargeable directly to the fan, it is caused either by excessive peripheral speeds, or the fan is of insufficient size. It should be remembered, however, that the tip speed required for a specified capacity and pressure varies with the type of blade, and that a tip speed which may be excessive for the forward curved type is not necessarily so for the backward or slightly backward type. A noisy fan usually is one which is operated at a point considerably beyond maximum efficiency.

Table 1. Good Operating Velocities and Tip Speeds for Forward Curved Multiblade Ventilating Fans

Static Pressure Inches of Water	Outlet Velocity Feet per Minute	Tip Speed Feet per Minute
¼	1000–1100	1520–1700
⅜	1000–1100	1760–1900
½	1000–1200	1970–2150
⅝	1100–1300	2225–2450
¾	1200–1400	2480–2700
⅞	1300–1600	2660–2910
1	1500–1800	2820–3120
1¼	1600–1900	3162–3450
1½	1800–2100	3480–3810
1¾	1900–2200	3760–4205
2	2000–2400	4000–4500
2¼	2200–2600	4250–4740
2½	2300–2600	4475–4970
3	2500–2800	4900–5365

For a given static pressure there is a corresponding outlet velocity and peripheral speed wherein maximum efficiency is obtained. If a fan is selected to operate at this point, the cost of operation and the noise can be held within control.

To aid in selecting fans as near as possible to the point of maximum efficiency, there are listed in Tables 1 and 2 for each static pressure corresponding outlet velocities and tip speeds which will give satisfactory results. The proper tip speed for a given static pressure varies with the design of wheel and with the number of blades or vanes in the wheel.

Lower outlet velocities than those listed in Table 1 may be employed, but care must be exercised to avoid selecting a fan for operation below its useful range. The useful range of the fans of Table 2 extends over the full length of the performance curve.

In exhaust ventilating systems where the air column moves toward the fan, noise due to the higher tip speeds and outlet velocities will not be so readily transmitted back through the air column to the building as when the air column is moving toward the rooms. Therefore higher outlet velocities may be used, but this will be at the expense of increased horsepower.

Amply large fans should always be used for both exhaust and supply systems, as there may be and usually is leakage despite the most careful workmanship, necessitating the delivery of more air at the fans than is exhausted from or supplied through the openings in the various rooms.

Long runs of distributing ducts, heaters, and air washers require definite increments of the total pressure which a supply fan in a ventilating system must overcome. These static pressures should be considered when selecting the fan characteristics, speed, and power.

Table 2 Good Operating Velocities and Tip Speeds for Multiblade Ventilating Fans with Backward Tipped and Double Curved Blades

Static Pressure Inches of Water	Outlet Velocity Feet per Minute	Tip Speed Feet per Minute
1/4	800–1100	2600–3100
3/8	800–1150	3000–3500
1/2	900–1300	3400–4000
5/8	1000–1500	3800–4500
3/4	1100–1650	4200–5000
7/8	1200–1750	4500–5300
1	1200–1900	4800–5750
1 1/4	1300–2100	5300–6350
1 1/2	1400–2300	5750–6950
1 3/4	1500–2500	6200–7550
2	1600–2700	6650–8050
2 1/4	1700–2800	7050–8550
2 1/2	1800–2950	7450–9000
3	2000–3200	8200–9850

Fans picked within the limits of Table 1 will operate close to the point of maximum efficiency. No attempt has been made to select these limits for quiet operation, since this is a relative term and varies with the type and location of the installation.

The connection of a fan to a metallic duct system should be made by canvas or a similar flexible material so as to prevent the transmission of fan vibration or noises. Where noise prevention is a factor the fan and its driver should have floating foundations.

Fans for Drying

Both axial flow and centrifugal types of fans are used for drying work. Propeller fans are well adapted to the removal of moisture-laden air when operating against low resistance and when handling air at low temperatures. Motors on these fans usually are of the fully-enclosed moisture-proof types so that saturated air or air containing foreign material will not injure the motors.

Unit heaters employing axial flow fans are widely used in the drying

field. In drying, these fans may be used with unit heaters where not too much duct work is required and where air is to be delivered against pressure, since the noise developed from the high peripheral speed of these fans is not ordinarily objectionable in process work.

Centrifugal fans of the multiblade type generally are selected to supply air for drying, as they are capable of delivering large volumes of air against all pressures likely to be encountered.

Belt driven fans usually are to be preferred to direct-connected fans since efficient motor speeds do not usually coincide with efficient fan speeds. Replacement of a standard motor is quick and easy if it is belted.

Wherever drying is done throughout the year and where air requirements change as the drying conditions change, the drying can be speeded up or reduced through control of the fan capacity. This may be done by changing the fan speed or by varying the outlet area with dampers. A throttled outlet reduces the volume and reduces the power.

Due to the low speeds of forward curved multiblade or paddle-wheel type fans, these can be direct-connected to reciprocating steam engines, and the exhaust steam from the engines may be used in the heating apparatus. In selecting engine driven fans for drying processes, where a large quantity of exhaust steam is used in the heaters, a smaller fan and greater power consumption may be used, because power economy is not essential under this condition.

Where static pressure in a dryer varies, and where several fans must operate in parallel, fans are to be preferred which have a continuously rising pressure characteristic, such as is given by backward-curved or double-curved blades. This type of fan is well adapted for direct-connected motors of the higher speeds. (See Chapter 41 on Drying Systems.)

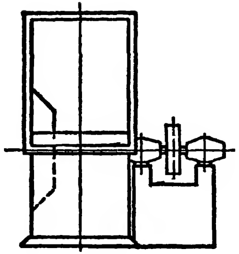
Fans for Dust Collecting and Conveying

The application of fans for handling refuse, dust, and fumes generated by machine equipment is covered in Chapter 21. Information is given regarding the methods for determining air quantities, the velocity required for carrying various materials and the method of determining maintained resistance or total static pressure at which the fan is to operate. The selection of a proper size fan is at times governed by the future requirements of the plant. In many instances, additional future capacity is anticipated and should be provided for.

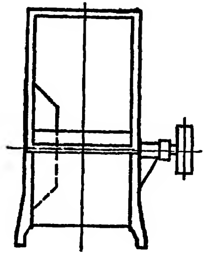
Having determined the necessary volume of air and the maintained resistance or static pressure required, the proper size fan may be selected from the fan manufacturers' performance charts or capacity tables. The fan chosen should be the size that will provide the required ultimate quantities with the minimum power consumption.

FAN VOLUME CONTROL

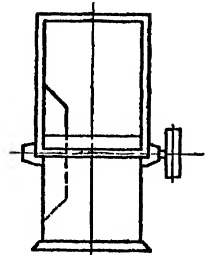
Some method of volume control of fans usually is desirable. This may be done by varying the peripheral velocity or by interposing resistance, as by throttling-dampers. Both methods, since they reduce the volume of air, reduce the power required. In many installations adjustments of volume are desirable during varying hours of the day. In others an



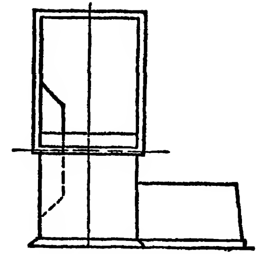
Arr. 1.
For belt drive.
Wheel overhung. Bearings on pedestal.



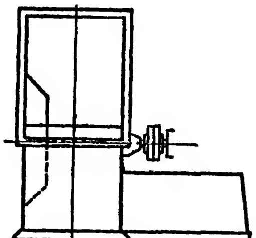
Arr. 2.
For belt drive.
Pulley and wheel overhung Bearings in bracket on fan housing Made only in smaller sizes for reversible discharge.



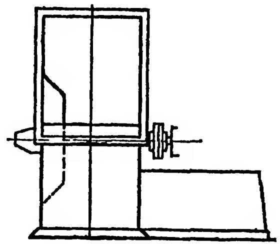
Arr. 3.
For belt drive.
Pulley overhung. Bearings supported on fan housing



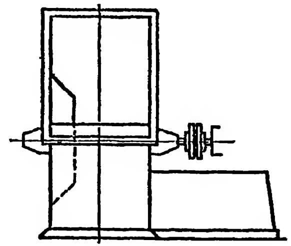
Arr. 4.
For direct drive.
Wheel overhung No bearings on fan Wheel mounted on motor or engine shaft Pedestal for motor or engine



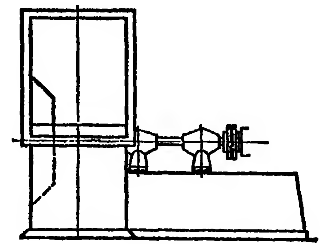
Arr. 5.
For direct drive.
Wheel overhung. Includes housing, wheel, shaft, one intermediate bearing, flanged coupling and pedestal only for motor or engine.



Arr. 6.
For direct drive.
Three-bearing arrangement with fan bearing at inlet side Includes housing, wheel, shaft, one bearing (in inlet), rigid coupling, and pedestal only for motor or engine



Arr. 7.
For direct drive
Similar to Arr 6, but with two bearings on fan, and flexible instead of rigid coupling



Arr. 8.
For direct drive.
Similar to Arr 5, but with two bearings on pedestal with motor, and flexible instead of rigid coupling

FIG. 6. ARRANGEMENT OF FAN DRIVES

increased supply of air in summer over that needed for winter is demanded. Experience is required in deciding whether speed-control or damper-control shall be used for specific cases. Where noise is a factor, it may be exceedingly desirable to reduce the speed at times, while on the other hand, any fan which has its normal speed reduced as much as 50 per cent without *change in resistance* will move only 50 per cent of the air.

FAN DESIGNATIONS

Facing the driving side of the fan, blower, or blast wheel, if the proper direction of rotation is clockwise, the fan, blower, or blast wheel will be designated as *clockwise*. If the proper direction of rotation is counter-clockwise, the designation will be *counter-clockwise*. (The driving side of a single inlet fan is considered to be the side opposite the inlet regardless of the actual location of the drive.)[3]

This method of designation will apply to all centrifugal fans, single or double width, and single or double inlet. Do not use the word "hand," but specify "clockwise" or "counter-clockwise."

The discharge of a fan will be determined by the direction of the line of air discharge and its relation to the fan shaft, as follows:

Bottom horizontal: If the line of air discharge is horizontal and below the shaft.

Top horizontal: If the line of air discharge is horizontal and above the shaft.

Up blast: If the line of air discharge is vertically up.

Down blast: If the line of air discharge is vertically down.

All intermediate discharges will be indicated as angular discharge as follows:

Either top or bottom angular up discharge or top or bottom angular down discharge, the smallest angle made by the line of air discharge with the horizontal being specified.

In order to prevent misunderstandings, which cause delays and losses, the arrangements of fan drives adopted by the *National Association of Fan Manufacturers* and indicated in Fig. 6 are suggested.

If double width, double inlet fans are selected, care must be taken that both inlets have the same free area. If one inlet of a fan is obstructed more than the other, the fan will not operate properly, as one half of the wheel will deliver more air than the other half. The *backward curved* and *double curved* types with backward tip operate satisfactorily in double or in parallel operation.

MOTIVE POWER

It is no easy matter to predetermine the exact resistance to be encountered by a fan or, having determined this resistance, to insure that no changes in construction or operation shall ensue which may increase air resistance, thus requiring more fan speed and power to deliver the required volume, or which may reduce air resistance, thus causing delivery of more air and a consequent increase of power even at constant speed.

It is recommended, therefore, for centrifugal type fans that the rated power to be supplied shall exceed the rated fan power by a liberal margin, when *forward curved* types are used. When *backward* or *double curved* blade types are used, motors with ratings very close to that of the fan horsepower demand can be employed, provided the fan has a limiting horsepower characteristic.

[3]Recommendations adopted by the *National Association of Fan Manufacturers.*

Justification for liberal power provision exists also in the possibility of varying demand due to changes in ventilation requirements, intensity of occupation, and weather conditions.

The motive power of fans should be determined in accordance with the Standard Test Code for Disc and Propeller Fans, Centrifugal Fans and Blowers, as adopted by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS and the *National Association of Fan Manufacturers.*

Fans may be driven by electric motors, steam engines (either horizontal or vertical), gasoline or oil engines, and turbines, but as previously stated the drive commonly used is the electric motor.

REFERENCES

Heating, Ventilating and Air Conditioning, by Harding and Willard, Revised Edition, 1932

Fan Engineering, Buffalo Forge Company

Fans, by Theodore Baumeister, Jr

Theories and Practices of Centrifugal Ventilating Machines, by D Murgue, translated by A L Stevenson

Mechanical Engineers' Handbook, by Kent

Mechanical Engineers' Handbook, by Lionel S Marks

Constructive Mechanism and the Centrifugal Fan, by George D Beals.

Coal Miners Pocket Book.

The Fan, by Charles H. Innes.

Mine Ventilation, by J. J. Walsh (A S H V E TRANSACTIONS, Vol 23, 1917).

Fan Blower Design, by H F. Hagen (A S.H.V.E. TRANSACTIONS, Vol. 28, 1922).

The Centrifugal Fan, by Frank L. Busey.

Section X, A S H V.E. Code of Minimum Requirements for the Heating and Ventilation of Buildings (Edition of 1929)

PROBLEMS IN PRACTICE

1 ● What information must be supplied to the manufacturer when ordering a centrifugal fan?

a. Size of fan (catalog number).

b. Type of fan.

c. Width of fan (single or double)

d. Number of inlets (single or double).

e. Fan performance and kind of application.

f Direction of rotation (clockwise or counter-clockwise).

g Direction of discharge (top horizontal, down blast, etc).

h. Drive arrangement (see Fig. 6)

i. Style of housing (full, three-quarters, etc.).

2 ● In selecting fans for quiet operation in public buildings:

a. Should the outlet velocity of the fan be limited?

b. Should the tip speed of the fan be limited?

a. Because all commercial fans operating at pressures suitable for this class of work would be considered noisy if the fan were to discharge directly into the room, and

because the duct system on the fan discharge is depended upon to absorb a reasonable amount of fan noise, it is desirable to have a moderate run of duct work with some bends and elbows included as sound deadeners. Where this duct is of necessity very short, the outlet velocity must be kept down to the lower limits recommended in this chapter or else an efficient sound absorber must be used The experience of the engineer must be his guide in determining the allowable outlet velocity in each individual case.

b Tip speed should not ordinarily be limited, because different types of fan blades have entirely different allowable tip speeds for quiet operation. A fan having a backward blade at the tip can run at much higher tip speed than can a forward curved or a straight blade fan, with the same degree of quietness

3 ● Is a direct connected or a belted fan preferable in public building work?

Where space is at a premium, direct connection is best Next in space economy is the short V-belt drive. The flat belt drive fan requires the greatest floor space. In this class of work, pressures are usually so low that even with the high speed fans the motor cost is greater for direct connected units than for belt drive fans

4 ● a. What type fans are used in industrial work?
b. What outlet velocity is suitable?

a. All of the centrifugal types are suitable, the disc and propeller types are suitable for low pressure work, or they are often used as exhausters.

b. The outlet velocities on fans for industrial work can be much higher than can those in public building work, where quietness is essential Fans should be selected with outlet velocities as recommended in this chapter, using the upper limit of velocities.

5 ● Are direct connected or belted fans preferred in industrial work?

In industrial applications, fans are often advantageously direct connected to motors. The pressures are usually high enough to use standard motor speeds. The high speed types of fans have limiting horsepower characteristics so that little margin in power must be provided in the driving motor. Belted fans may be used, but where high power is required a special arrangement is often necessary for shaft and bearings on account of the weight of the sheave and the belt pull.

6 ● A forward curved multiblade fan which requires 5.4 bhp is delivering 22,800 cfm at 70 F against a resistance pressure of 1 in. of water at an outlet velocity of 1440 fpm:

a. What is the static efficiency?

b. What is the total efficiency?

a 66.3 per cent (see Equation 2)

b. 74 5 per cent (see Equation 3).

7 ● If the above fan has a 54-in. diameter wheel and operates at 193 rpm, will it be suitable for a ventilating installation where a minimum of noise is desirable?

Yes The tip speed will be 2720 fpm and this, together with the 1440 fpm outlet velocity, falls within the limits given in Table 1 for 1-in. resistance pressure.

8 ● What objectionable feature is inherent in the ordinary propeller fan when it is operating at high resistance pressures?

It must operate at a high speed with consequent noise

9 ● At what point should a fan be selected for operation, and why?

At its point of maximum efficiency because the cost of operation and the noise produced will be least.

10 ● In Fig. 3, a static pressure of 85 per cent of blocked tight pressure corresponds to three different volumes, namely 11 per cent, 30 per cent and 48 per cent of wide open volume. What will determine which volume the fan delivers?

The fan can operate only at the intersection of its pressure-volume curve and the system characteristic The type of system, together with the specification of the volume at a certain static pressure, completely defines the system characteristic.

As illustrated in Fig 5, a given system characteristic will intersect the fan curve in only one point.

If the 85 per cent value for static pressure is specified for the 48 per cent value of volume, it is at once obvious that the same system will not have the same resistance at any other volume.

Chapter 18

SOUND CONTROL

Decibel Defined, Apparatus for Measuring Noise, The Sound Control Problem, Acceptable Noise Levels, Controlling Vibration from Machine Mountings, Controlling Noise through Room Wall Surfaces, Controlling Noise Transmitted Through Ducts

IN ventilating and air conditioning a building or a room, the effect of the mechanical system employed must be considered on the acoustics of the space conditioned. It is important to consider also that the use of air conditioning often permits keeping the windows closed, thus giving relief from certain external noises, but at the same time increasing the necessity of providing adequate sound control.

It is not assumed that the ventilating and air conditioning engineer will attempt to improve the acoustics of the space that is being conditioned, but the designer should have at least enough fundamental knowledge of the acoustical effects of the system which is being designed to be sure that no damaging effects occur to the existing acoustical properties. It is assumed that in a given space the architect and acoustical engineer have produced a room or rooms which are satisfactory for speech, music, or other uses. The ventilating engineer's sole function is to ventilate and air condition these rooms properly so that they will be physically comfortable without adding any acoustical hazards.

UNIT OF NOISE MEASUREMENT

In the United States and England the unit of noise measurement is the *decibel* (db). In Germany this unit is called the *phon*. The decibel is defined by the relation $N = 10 \log \frac{I_1}{I_0}$, where N is the number of decibels by which the intensity flux I_1 exceeds the intensity flux I_0. The intensity flux is the measure of the energy contained in a sound wave and is defined in terms of microwatts per square centimeter of wave front in a freely traveling plane wave. It is usually more convenient to select an arbitrary reference intensity for I_0 and express all other intensities in terms of decibels above that level. For this purpose the threshold of audibility for the average human ear at a frequency of 1000 cycles per second has been selected. This reference threshold is 10^{-16} watts per square centimeter or 10^{-10} microwatts per square centimeter. This reference level also corresponds to a pressure of 0.0002 dynes per square centimeter.

A stated sound level in decibels, unless otherwise defined, will thus be related to a threshold of 10^{-16} watts. For example, a level of 60 db above this reference threshold is 10^{-10} watts. In a similar manner, when sound

measurements are given in actual intensity or energy units, they can be converted to decibels by this relation.

Since the decibel is a ratio, it can only be employed when related to a reference threshold level as given. Noise levels, which vary with frequency as well as intensity, must not only be related to this reference threshold level, but also to a reference frequency, which is taken as 1000 cycles. These terms and procedures may be found in tentative standards[1] published by the *American Standards Association.*

APPARATUS FOR MEASURING NOISE

Since the relative loudness to the ear, rather than the actual physical intensity, is the quantity in which engineers are usually interested, it has been found necessary to allow for the varying sensitivity of the ear at different frequencies in designing noise measuring equipment. The most satisfactory method of measuring noise is by means of a sound level meter which usually consists of a microphone, a high gain audio-amplifier, and a rectifying milliammeter which will read directly in decibels. This meter is calibrated to give readings above the threshold of audibility and usually contains a weighing network to make it less sensitive at those frequencies where the ear is less sensitive For complete specifications relative to the approved type of sound level meters refer to the information[2] published by the *American Standards Association.*

GENERAL PROBLEM OF SOUND CONTROL

As previously stated, the function of the ventilating and air conditioning engineer is to add no acoustical hazard to the conditions already present in the room or building and the problem can be stated as:

a To determine the noise level existing without the equipment

b To ascertain the noise level which would exist if the equipment were installed without sound control

c To provide as a part of the installation sufficient sound control appliances to reduce the noise level substantially to that found in (*a*).

To accomplish this the engineer should have information of three kinds:

1 A knowledge of the noise levels currently considered acceptable in various rooms in order that he may have a basis on which to proceed

2. A knowledge of the nature and intensity of the noise created by the various parts of the equipment

3 A knowledge of how, when necessary, to vary and control the noise level between the equipment and the conditioned space.

In addition, the engineer should have information available to deal with noises which may enter the room due to openings made into it to accommodate the equipment, such as cross talk between rooms connected with common ducts and noise transmitted to portions of duct system outside the conditioned space and through to its interior.

While the general problem may be logically outlined and the items of

[1]American Tentative Standards for Noise Measurement, *American Standards Association*

[2]American Tentative Standards for Sound Level Meters for Measurement of Noise and Other Sounds, *American Standards Association*

knowledge necessary to its solution can be listed, the available information at present is lacking in certain respects. However, attention may be directed to that information which is currently available, and to furthermore outline a solution of the noise problem based on these data.

ACCEPTABLE NOISE LEVELS

Measurements of noise levels have been observed by several investigators in various rooms and locations. The information compiled in Table 1 is based on these data, which represent the best opinion on the

TABLE 1 TYPICAL NOISE LEVELS

Rooms	Noise Level in Decibels to be Anticipated		
	Min	Representative	Max
Sound Film Studios	10	14	20
Radio Broadcasting Studios	10	14	20
Planetarium	15	20	25
Residence Apartments, etc	25	35	40
Theatres, Legitimate	25	30	35
Theatres, Motion Picture	30	35	40
Auditoriums, Concert Halls, etc	25	30	40
Churches	25	30	35
Executive Offices, Treated Private Offices	25	33	40
Private Offices, Untreated	35	45	50
General Offices	45	55	60
Hospitals	25	40	55
Class Rooms	30	35	45
Libraries, Museums, Art Galleries	30	40	45
Public Building, Court Houses, Post Offices, etc.	45	55	60
Small Stores	40	50	60
Upper Floors Department Stores	40	50	55
Stores, General, Including Main Floor Dept Stores	50	60	70
Hotel Dining Rooms	40	50	60
Restaurants and Cafeterias	50	60	70
Banking Rooms	50	55	60
Factories	60	70	80
Office Machine Rooms	60	70	80
Vehicles			
Railroad Coach	60[a]	70	80
Pullman Car	55[a]	65	75
Automobile	50	65	80
Vehicular Tunnel	75	85	95
Airplane	80	85	100

[a]For train standing in station a level of about 45 db is the maximum which can ordinarily be tolerated

subject now available. All levels are given in decibels above a reference threshold of 10^{-16} watts (corresponding to a pressure of 0.0002—dynes per square centimeter). Minimum, representative, and maximum levels are given for each application. These values are intended to indicate the variation which may be expected in different locations of the same type, but not the time variation which may be expected in each location.

The values shown in Table 1 are typical of those found currently in

existing spaces. They are, however, the noise levels of the room and not the noise levels of the ventilating or air conditioning equipment. If the noise level at the room of the equipment is kept at the levels shown in the table the equipment will not add to the acoustical hazard existing without it, provided the equipment noise is heard alone, but if both are heard together the total noise level in the room will be increased about 3 db. This is usually considered an acceptable result.

In some cases it is desirable to keep the equipment noise level at the room at such a value that it actually will not increase the noise level in the room to any measurable degree. This can usually be accomplished if the equipment noise at the room can be kept 10 db below the noise level shown in the table.

NOISE CREATED BY EQUIPMENT

Information concerning the noise levels created by ventilating and air conditioning equipment such as fans, motors, air washers, and similar items is not yet on a basis which permits tabular presentation although certain manufacturers are prepared to offer such data and do state the noise producing properties of their products.

Absence of this information makes it necessary to resort to indirect means in solving certain problems and also prevents a direct logical solution.

KINDS OF NOISE

To solve a sound problem of this type it is desirable to consider separately the several means by which noise reaches the room. This avoids to some extent the necessity of knowing the noise level at the source and places the emphasis on ascertaining the level at the point where the sound enters the room rather than on its point of origin.

The noise introduced into a room or building by ventilating or air conditioning equipment may be divided into two kinds depending on how it reaches the room as:

1. Noise transmitted through the building construction.
2. Noise transmitted through the ducts

It is convenient to further subdivide these two methods of delivery as.

1 Noise transmitted through the building construction.
   a. From machine mountings as vibration.
   b. From equipment through room wall surfaces.

2. Noise transmitted through the ducts
   a. From equipment such as sprays, fans, etc.
   b. From outside, and transmitted through duct walls into air stream
   c. From air current, including eddying noises.
   d. Cross talk and cross noises between rooms connected by the same duct system

The next step in the solution of this problem is to present data and discuss methods whereby solutions to the noise problem can be obtained when the allowable room noise level and the path through which the noise reaches the room are known.

NOISE THROUGH BUILDING CONSTRUCTION

It is impossible to select ventilating equipment which will operate without producing some mechanical noise, and since the equipment must be mounted in a building, it is probable that a part of this noise will be transmitted to the building itself to such a degree as to make noisy conditions in the rooms which are to be air conditioned. Much of this noise may be transmitted by the duct if it is rigidly connected to the fan outlet. It is common practice to make the connection between the fan and the duct with a canvas sleeve which effectively restricts noise at this point. Noise may also enter the building through the mounting of the motor and the fan. Flexible mountings should be provided in all installations but these mountings must be carefully designed so that they will actually reduce the contact between the machinery and the supporting floor. If a flexible material is used, it is desirable to investigate the installation so that it is not short-circuited by through bolts which are improperly insulated and by electrical conduit which is not properly broken and is attached both to the equipment and to the building. The flexible mounting, if it is improperly engineered, may actually increase the contact between the equipment and the floor upon which it is supported. In general, the flexible material should be loaded as heavily as possible without impairing its load-carrying capacity.

Controlling Vibration from Machine Mountings

The theory of the insulation of vibration was first worked out by Soderberg[3]. If a machine of mass m be supported by an elastic pad the amount of vibratory force communicated by the machine to the floor or foundation upon which it rests will be determined by the elastic and viscous properties of the pad. The ratio of the vibratory force communicated to the floor or foundation with the machine resting upon the pad, and with the machine resting directly upon the floor, is given by the following equation:

$$\tau^1 = \sqrt{\frac{r^2 + \dfrac{1}{4\pi^2 n^2 c^2}}{r^2 + \left(2\pi n m - \dfrac{1}{2\pi n c}\right)^2}} \tag{1}$$

where

τ^1 = the so-called *transmissibility* of the support.

c = the compliance (that is, the reciprocal of the force constant).

r = the mechanical resistance owing to the viscous forces within the support

n = the frequency of vibration generated by the machine which is to be insulated, such as the commutation frequency of a motor or the blade frequency of a fan.

m = the mass of the machine to be insulated.

It should be noted that not only must vibrations within the audible range of frequencies be considered, but those in the sub-audible range as well, since these may cause objectionable vibrations. All the possible frequencies should be considered in the calculation. Sometimes beat effects are introduced by slight irregularities of belts or pulleys that have much lower frequencies than those of the rotating elements.

---

[3]C. R. Soderberg, *The Electric Journal* (January, 1924), and succeeding articles. See also V O. Knudsen, *Physical Review*, Vol. 32, 1928, p 324, and A. L. Kimball, Journal *Acoustical Society of America*, Vol. 2, 1930, p 297

If r, the mechanical resistance, is very small, formula 1 may be written

$$\tau = \frac{1}{\frac{n^2}{n_0^2} - 1} \qquad (2)$$

where n_0 is the natural frequency of the machine upon the elastic pad,

$$n_0 = \frac{1}{2\pi}\sqrt{\frac{1}{mc}}$$

In most cases of design of resilient machine mounting the effect of frictional resistance is small, and Equation 2 may be used. In such cases it is only necessary to know the natural frequency of the elastic pad or platform used under the desired loading and the transmissibility for any vibrational frequency of the machine may be obtained. However, this formula gives the theoretical maximum insulation which may be obtained and should be used with a liberal factor of safety. (A factor of 2 is common practice.)

If the pad is to be of any value in the prevention of solid-borne vibrations, the value of τ' must be considerably smaller than unity. If the fundamental frequency of vibration generated by the machine happens to coincide with the natural frequency of the mass of the machine resting on the elastic pad, a condition of resonance will be established, and the machine will exert a greater force upon the foundation than it would if the pad were completely removed. It is necessary, therefore, that the elastic support be sufficiently compliant, and the mass of the machine sufficiently heavy, that the natural frequency of the mass m upon its elastic support will be low in comparison with the frequencies which are generated by the machine. Thus, if the principal vibrations in the machine be of the order of 100 vibrations per second, the natural frequency of the machine mounted on its elastic support should not exceed about 50 vibrations per second, and for best results preferably 20.

When the forced frequency is low, it is frequently impossible to insulate for the fundamental forced frequency due to connecting pipe work and other relevant factors. In cases of this kind an effective installation of sound insulation may be obtained with a mounting which functions far above the fundamental forced frequency. For example, a compressor operating at 500 rpm has a forced frequency of 8.3 vibrations per second. By designing a mounting having a natural frequency of 20 to 25 vibrations per second, it is possible to isolate practically all of the noise.

If a slab of insulating material be placed under the entire foundation of a machine, as is often done in practice, it may happen that the natural frequency of the machine on its elastic support will be nearly the same as the frequencies which are to be insulated, in which case the elastic support will be worse than nothing. In general, as Equation 1 shows, both m and c should be as large as possible if the vibrations of the machine are to be effectively insulated from the solid structure of the building.

The elastic support under the machine acts as a low-pass filter which passes all frequencies below about two times the natural frequency of the machine mounted on its elastic support, but prevents all frequencies

above about $\sqrt{\frac{mc}{\pi}}$ from reaching the solid structure of the building. The principal influence of the internal mechanical resistance r is to limit the vibration at the resonant frequency. It is generally advisable, therefore, to use materials which have an appreciable internal resistance.

The values of c and r can be determined for any specimen of flexible material and, when known, can be used to determine the insulation value of any particular set-up. The value of c can be obtained by making static measurements of the amount of displacement of the compressed support for each additional unit of the compressing force. If this be done for a specimen of the flexible material of a certain thickness and area of cross section, the compliance can be determined for any other thickness or area from the relation that c will be directly proportional to the thickness and inversely proportional to the area of the flexible support. When the internal resistance r is not too large, it can be determined by observing the successive amplitudes of the free vibrations of a mass m which rests upon a specimen of the flexible material, and solving for r by the usual log-decrement method. Or, if the damping be so great that the free motion of m is non-oscillatory, r can be obtained from measurements on the experimentally-determined resonance curve of the forced vibrations of m, or from measurements of the rate of return of m when it is given an initial displacement.

If the resistance of a certain specimen of material, as cork, felt, or rubber, has been determined by any of these methods, the resistance for any other thickness or area of the material can be determined approxi-

TABLE 2. COMPLIANCE AND RESISTANCE DATA FOR TYPICAL SPECIMENS OF FLEXIBLE MATERIALS[a]

The compliances and resistances given in the table are for specimens 1 in. thick and 1 sq cm in cross-section

MATERIAL	DESCRIPTION OF MATERIAL	APPROXIMATE UPPER SAFE LOADING IN POUNDS PER SQUARE INCH	COMPLIANCE c IN CENTIMETERS PER DYNE	RESISTANCE r IN ABSOLUTE UNITS
Corkboard	1.10 lb per board foot	12	0.25×10^{-6}	0.15×10^{5}
Corkboard	0.70 lb per board foot	8	0.50×10^{-6}	0.25×10^{5}
Fiber Board	1.35 lb per board foot	4 to 6	0.60×10^{-6}	0.50×10^{5}
Fiber Board	Carpet lining	10	0.40×10^{-6}	----------
Fiber Board	Insulating board	12	0.18×10^{-6}	----------
Fiber Board	Insulating board	15	0.16×10^{-6}	----------
Fiber Board	Insulating board	15	0.12×10^{-6}	----------
Anti-Vibro-Block	----------	5	0.60×10^{-6}	1.5×10^{5}
Sponge Rubber	25 lb per cubic foot	1 to 3	3.0×10^{-6}	----------
Soft India Rubber	55 lb per cubic foot	3 to 6	1.2×10^{-6}	----------

[a]From *Architectural Acoustics*, by V. O. Knudsen, p. 278

mately because the resistance will be inversely proportional to the thickness and directly proportional to the area of cross-section of the flexible support. Thus, if the values of c and r for a flexible material be known, it is possible to calculate, by means of Equation 1, the amount of insulation that will be obtained from the use of this material as a flexible support for a piece of equipment having a mass m. For the routine calculations in practice, r may be neglected with only a slight sacrifice of accuracy. Table 2 gives the values of c and r for a number of commonly used flexible materials.

Example 1. A machine weighing 1000 lb has a base area of 20 sq ft Assume that the principal vibration of the machine has a frequency of 100 cycles per second (most machinery vibrations are less than 150 vibrations per second, and the assumed frequency of 100 is quite representative of typical machines). Suppose that a 1-in slab of corkboard weighing 1.10 lb per board foot be placed between the machine and the floor. The loading on the cork will then be only 50 lb per square foot, or slightly more than ⅓ lb per square inch (It is assumed that the compliance c in centimeters per dyne for a specimen 1 in. thick and 1 sq cm in cross-section is 0.25×10^{-6} and the resistance r in mechanical ohms is $0\ 15 \times 10^{5}$.)

The *transmissibility* is calculated in the following manner·

Mass of machine in grams = $1000 \times 454 = 4.54 \times 10^{5}$
Area of base in square centimeters = $20 \times 144 \times 2.54 \times 2\ 54 = 1.86 \times 10^{4}$.

Therefore, the compliance of the entire support, 1 in. thick and 20 sq ft in cross section, is $0.25 \times 10^{-6} \times \frac{1}{1.86 \times 10^{4}} = 0.134 \times 10^{-10}$ cm per dyne, and the resistance of the entire support is $0.15 \times 10^{5} \times 1\ 86 \times 10^{4} = 0.28 \times 10^{9}$ mechanical ohms (or absolute units) Therefore,

$$\tau^{I} = \sqrt{\frac{(0\ 28 \times 10^{9})^2 + \frac{1}{4\pi^2 \times 100^2 \times (0.134 \times 10^{-10})^2}}{(0.28 \times 10^{9})^2 + \left((2\pi \times 100 \times 4.54 \times 10^{5} - \frac{1}{2\pi \times 100 \times (0\ 134 \times 10^{-10}}\right)^2}}$$

$$= \sqrt{\frac{0.0784 \times 10^{18} + \frac{10^{18}}{4\pi^2 \times 10^2 \times 0.018}}{0.0784 \times 10^{18} + \left(2\pi \times 4.54 \times 10^{7} - \frac{10^{8}}{2\pi \times 0.134}\right)^2}} = 0\ 935$$

Consequently, it is seen that the *transmissibility* is nearly equal to unity, and that the support therefore is not satisfactory for insulating 100 or fewer vibrations per second

If the amount of cork be reduced so that it is loaded to 10 lb per square inch, the total area of the supporting cork will be only 100 sq in. or 645 sq cm The compliance of the entire support will now be $0.25 \times 10^{-6} \times \frac{1}{645} = 0.39 \times 10^{-9}$ cm per dyne, and the resistance will be $0.15 \times 10^{5} \times 645 = 0\ 97 \times 10^{7}$ mechanical ohms (or absolute units). Therefore

$$\tau^{II} = \sqrt{\frac{(0.97 \times 10^{7})^2 + \frac{1}{4\pi^2 \times 100^2 \times (0\ 39 \times 10^{-9})^2}}{(0.97 \times 10^{7})^2 + \left(2\pi \times 100 \times 4.54 \times 10^{5} - \frac{1}{2\pi \times 100 \times (0\ 39 \times 10^{-9})}\right)^2}}$$

$$= \sqrt{\frac{0\ 94 \times 10^{14} + \frac{10^{14}}{4\pi^2 \times 0\ 1521}}{0\ 94 \times 10^{14} + \left(2\pi \times 4\ 54 \times 10^{7} - \frac{10^{7}}{2\pi \times 0.39)}\right)^2}} = 0\ 0375$$

It is seen, therefore, that with the bearing surface on the cork reduced to 100 sq in. (that is, with the cork loaded to 10 lb per square inch), the *transmissibility* is reduced to 0.0375, or the amplitude of vibration transmitted to the floor will be only about 1/27 of what it would be if the machine were mounted directly upon the floor. These two numerical examples will serve to show not only the manner of making the calculations, but also the importance of selecting the proper type and design of flexible supports for insulating the vibrations of a machine from the rigid structure of a building.

Controlling Noise Through Room Wall Surfaces

The ventilating equipment is usually housed in a separate room where the noise produced by the mechanical operation of the equipment can be isolated from the rest of the building. If the vibration of the machinery is absorbed by flexible mounting and is not transmitted to the building,

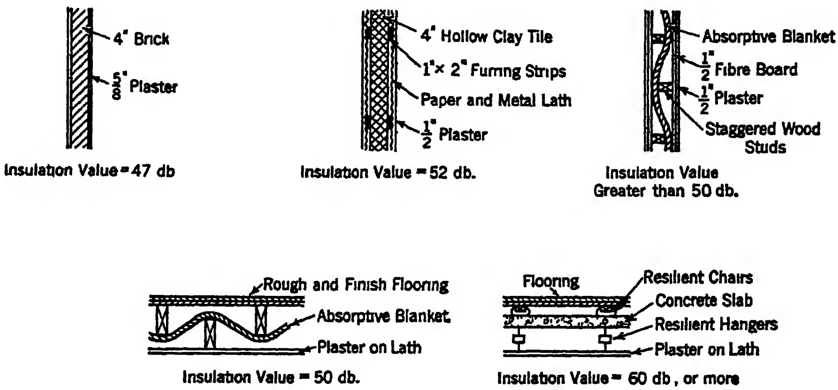


FIG. 1. THREE WALL SECTIONS AND TWO FLOOR AND CEILING SECTIONS WHICH ARE SUITABLE FOR THE INSULATION OF EQUIPMENT ROOMS[a]

[a]Acoustical Problems in the Heating and Ventilating of Buildings, by V. O. Knudsen (A S H V E TRANSACTIONS, Vol. 37, 1931)

the only noise to be eliminated by the walls of the room will be the air-borne mechanical noise. Acoustical measurements on average brick, tile, lath, and plaster walls indicate that the usual wall of these types is sufficient to satisfactorily attenuate this air-borne mechanical noise.

Three wall sections and two floor and ceiling sections which are satisfactory for the wall insulation of the equipment room are shown in Fig. 1.

Attention should be given to the equipment room door, since this door may leak badly and allow sound to escape into parts of the building which should be quiet. Where the equipment noise is particularly severe, double doors should be used and in all cases, the doors of the equipment room should be fitted with tight thresholds and weather-stripping. The door itself may transmit considerable sound if it is thin but it will not transmit a tenth as much as will be transmitted by a ¼-in. crack between the door and the threshold.

In cases where the equipment noise is extraordinarily high, it may be

necessary to treat acoustically the walls and ceiling of the equipment room. If the equipment room is not entirely closed, partition walls may be necessary.

NOISE TRANSMITTED THROUGH THE DUCTS

After noise reaches the air stream in the ducts it can be controlled by lining the ducts on the inside with a sufficient quantity of sound absorbing material. Lagging material of similar characteristics placed on the outside of ducts serves to prevent noise originating outside the ducts being carried inside the ducts and into the air stream.

A case where outside lagging is desirable occurs when ducts originate at the fan in the equipment room and pass through this room on the way to the room being conditioned or ventilated. Unless the ducts are lined some of the mechanical noise from the equipment room air may be transmitted through the wall of the duct, thus reaching the air stream and be carried into the room. In such cases, that portion of the duct which is exposed to the sounds in the equipment room should be lagged with material such as cork, pipe covering or other sound damping material to prevent the sound from entering the duct at this point. Numerical data are not available to permit a simple and practical calculating procedure to determine thickness of covering which should be used for this purpose.

Inside lining material used in the case previously mentioned would serve as an absorber of the sound transmitted through the duct walls, and thus act as a means of preventing the transfer of noise into the air stream.

Inside lining may also be used in ducts to absorb noise which reaches the air stream from equipment such as fans, sprays and coils; noise due to eddying currents set up by elbows, dampers and similar obstructions; and noise transmitted from room to room where there is a common duct system.

To use the lining effectively it must be properly located, well installed and be applied in sufficient quantity to reduce the noise level of the air stream to the level desired.

At present there are no wholly rational or generally recognized methods of calculating the amount of duct lining necessary to accomplish a given reduction of noise level in the air travelling in a duct system; consequently some empirical method has to be used. One empirical method is to use direct trial and error. Another empirical method uses a duct lining factor evaluated by experience. In the present state of the data on sound control for ducts, the latter method is convenient for making estimates, but attention is specifically called to its empirical nature and to the necessity of exercising judgment in applying it.

Use of Duct Lining Factor

A duct lining factor (f) giving numerical values for use at various equipment noise levels is shown in Fig. 2. When properly used with Table 1 this chart (Fig. 2) provides a solution which may be both useful and simple. It is important to understand that the levels referred to in this chart are the *average* noise levels set up in the room by the ventilating or air conditioning equipment. In the case of a piece of equipment which generates a noise level of 95 db, when the noise is measured immediately

next to the machine, there might be a reduction of 15 db in passing through the duct, and a further difference of 15 db between the noise at the outlet supply grille and the average level in the room, leaving an effective level of 65 db in the room. Reductions of noise level ranging from 5 to 25 db through duct systems have been encountered without the use of sound absorbing linings and the drop from supply opening to average room level may vary from 5 to 20 db.

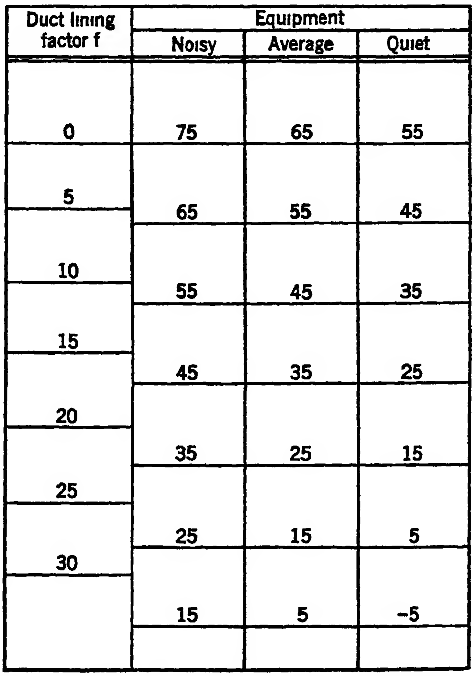


FIG. 2 CHART FOR DETERMINING NOISE REDUCTION IN DECIBELS FROM DUCT LINING FACTOR[a]

[a]Values for equipment noise are only general. Wherever possible substitute actual values as supplied by equipment manufacturer or as measured.

To determine whether to use column 1, 2, or 3 in Fig. 2, in forming an estimate of the relative amount of noise generated by the system, the length of the untreated duct system and the number of bends or elbows or splitters should be considered, since the longer and the more complex the system, the more reduction of noise level will occur before the sound reaches the room grilles. Also the sound absorbing power of the room should be taken into account, since in rooms where there is a great deal of absorptive material, such as rugs, draperies, curtains and furniture, there will be a higher loss between the outlet grille noise and the average room level. The ventilating engineer will have to judge whether the conditions deviate from the typical.

Manufacturers ratings on equipment should be considered in connection with the foregoing discussion. The quantity determined involves

the noise level which will be produced in the room and the manufacturer's method of rating must be considered before allowances previously mentioned are accepted.

To use Fig. 2, proceed by consulting Table 1 and determine the probable noise level already existing in the room, and, as suggested, assume that this level is satisfactory for current practice. This gives a noise level in decibels and with this enter the chart of Fig 2. Read across the chart and determine the value of the duct lining factor (f) in the column at the left. Then multiply the smallest cross sectional dimension (inches) of the duct by this factor. The result will be the length of duct in inches to be lined to attenuate an average fan noise. If circular ducts are used, the length to be lined will be (f) $\times$ diameter of duct.

Example 2 A 7 x 30 in. duct is connected to a private office space in a quiet location. Determine the length of lining necessary to attenuate a fan noise satisfactorily.

From Table 1 the noise level in this office will be 35 db.

Length to be lined for noisy equipment is $22 \times 7 = 154$ in.
Length to be lined for average equipment is $17 \times 7 = 119$ in
Length to be lined for quiet equipment is $12 \times 7 = 84$ in

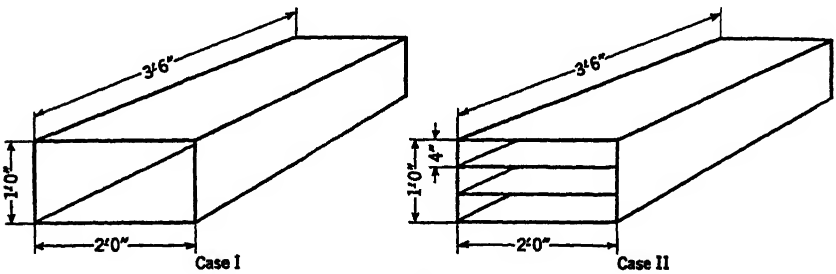


Fig. 3. Diagram of Branch Duct Connection

The sound absorbent properties of duct lining are extremely important and materials which have coefficients as high as possible should be used. This is particularly true of the coefficients at the low frequencies. Fig. 2 is based on materials having a noise quieting coefficient of 0.60 or more. For materials which are less efficient a factor of safety should be added[4].

Only certain sound absorbent materials among those listed in various publications will be found to be suitable for duct lining. In addition to a high sound absorbent coefficient a duct lining material should have a low surface coefficient of friction, high resistance to moisture absorption, and should be fireproof and vermin proof. A number of building codes now specify that any sound absorbent material used for duct lining shall have no fire hazard. There are no existing specifications on moisture resistance but the manufacturer should be required to show that the material will not absorb sufficient moisture to cause deterioration or to decrease the sound absorbing efficiency.

[4]For coefficients of commercial sound absorbent materials see Bulletin *Acoustical Manufacturers' Association*, 919 No. Michigan Ave, Chicago, Ill.

If, as is often the case, the length of duct from the main duct to a grille is shorter than the length of lining indicated by using the factor found, this duct may be sub-divided[5] into smaller ducts, so that the value found may be used as shown in Fig. 3.

Example 3 Assume a branch duct, as shown in Fig 3, is 24 in. wide by 12 in. high and 42 in. long. Use a duct lining factor of 10.

Case I (No splitters).

Length of lining = $f \times$ minimum dimension = $10 \times 12 = 120$ in
In this case the duct should be lined for 120 in. which is obviously impossible.

Case II. (Two splitters).

Results in 3 ducts 24 in. wide and 4 in. high.
Length of lining = $f \times$ minimum dimension = $10 \times 4 = 40$ in
This length of lining fulfills the space limitations of the branch duct which is 42 in. long.

General Suggestions

In some instances where high velocity air is used, a considerable amount of whistle is generated at the grille. This noise is obviously produced after the air leaves the duct and there is no treatment which can be installed in the duct that will reduce this noise. The engineer must take into consideration the type of grille which he intends to use and provide sufficient grille area so that the velocity through the grille is reduced to a point where the grille is not too noisy.

Ducts serving more than one room permit cross talk between the rooms and should be lined with acoustical material. Where the rooms are close together and the ducts short, the ducts should be sub-divided to provide ample acoustical treatment.

Very often in ventilating duct work the engineer feels that it will not be necessary to line ducts if the sound is travelling against the airflow. This, however, is untrue since sound travels so much more rapidly than does the air in even high velocity systems, that it will travel as easily against the airflow as it does with it.

Sounds which are low in pitch are much harder to eliminate from a duct system than sound which is high in pitch, consequently equipment which produces low pitched sounds should be avoided as much as possible.

REFERENCES

How Sound is Controlled, by V. O. Knudsen (*Heating, Piping and Air Conditioning*, October 1931).

Acoustical Problems in the Heating and Ventilating of Buildings, by V. O Knudsen (A.S.H V.E. TRANSACTIONS, Vol. 38, 1932).

Effect of Humidity upon the Absorption of Sound in a Room, by V. O. Knudsen (*Journal, Acoustical Society of America*, July 1931). Also see report presented at the May 1933, meeting of *A.S of A*.

Acoustics and Architecture, by P. E. Sabine

Architectural Acoustics, by V. O. Knudsen.

Acoustical Engineering, by West.

Modern Acoustics, by Davis.

[5]Patents exist covering the sub-dividing of ducts for installing sound absorbent materials

PROBLEMS IN PRACTICE

1 ● Does a soft pad under the ventilating machinery prevent building vibration?

It may or it may not In some cases a soft pad causes more vibration than no pad A flexible mounting should be carefully designed to be effective

2 ● Are especially designed walls necessary in an equipment room to keep noise out of adjoining spaces?

Ordinarily good brick, tile, or concrete walls are satisfactory Window and door openings should be made as tight as possible with weather-stripping, etc

3 ● How should mechanical noise be eliminated from the duct system?

A flexible connection between the fan discharge and the duct should be used The duct should be lined from the fan end for a certain length, depending on the degree of quietness desired.

4 ● Given the choice of two types of equipment, one generating high-pitched sounds and the other low-pitched sounds, which would you choose? Why?

The equipment generating the high-pitched noise should be chosen since high-pitched sounds are more easily absorbed than are low-pitched sounds

5 ● In building an acoustic filter in a short duct 32 by 24 in. which direction should the splitters run?

The splitters should be installed parallel to the longest dimension, since they will provide more acoustical material per splitter

6 ● What should be the characteristics of a good duct-lining material?

a. High noise reduction
b Physical strength
c Easy working and installation
d Fire resistance
e Cleanliness, absence of loose fibers or pieces
f Smooth surface to reduce air friction

7 ● Should a ventilating duct be lagged or covered on the outside?

Yes, in some locations, and particularly in the equipment room and where the duct runs through noisy rooms to serve a quiet room This lagging will prevent air-borne sounds from entering the duct through its sides and causing annoying sound in the quiet room

8 ● How can cross-talk be eliminated when one duct serves two or more rooms?

Install proper filters adjacent to the grilles in each room, using splitters if the duct leads to the rooms are short

9 ● Space limitations and maximum air velocities for the introduction of air to a broadcasting studio restrict the size of duct to 30 by 16 in. and in addition the length of branch duct which is suitable for lining with sound absorption material is limited to 22 ft. Determine the length of duct lining necessary to attenuate an average fan noise and establish a permissable room noise level.

Referring to Fig 2 the noise level for broadcasting tudio is 14 db and the corresponding duct lining factor f is 28. Minimum cross sectional dimension of duct = 16 in

$\frac{16 \times 28}{12}$ = 37 3 ft duct lining required.

Maximum length of duct is 22 ft, therefore it is necessary to divide the duct with a splitter, resulting in a minimum duct dimension = 8 in.

$\frac{8 \times 28}{12}$ = 18 7 ft duct lining required to attenuate and average fan noise.

Chapter 19

AIR DISTRIBUTION

Definitions, Grille Locations, Standards for Satisfactory Conditions, Factors Affecting Distribution for Cooling and Heating, Air Outlet Noises, Selection of Supply Outlets, Balancing System

CORRECT air distribution contributes as much or more to the success of a forced air heating, ventilating, cooling or air conditioning system as does any other single factor Supplying the proper amount of air is one problem: properly distributing it from the point where it leaves the fan is another. The distribution problem may be further divided into: (*a*) distribution *to* the various spaces served by the system, (*b*) distribution *in* these spaces. This discussion is primarily limited to division (*b*), reference being made to the duct system only insofar as it affects the performance of the air distribution outlets.

Definitions

In this discussion, the term *air outlets* or *outlets* will be used to designate a cover for an opening, whether it is a grille or a register.

A *register* is defined as an outlet with a damper, and a *grille* is defined as an outlet without a damper.

The perpendicular distance over which the air will satisfactorily carry measured between the face of the outlet and the opposite wall is the *throw*. In the case of directional flow outlets, this may be less than the actual carry of the air.

The *core area* of the outlet is the area of that portion of the grille inside the frame through which the air can flow.

The ratio of width to height of the core area is termed the *aspect ratio*.

GRILLE LOCATIONS

The location of supply and exhaust outlets is extremely important if a satisfactory installation is to be secured. Very frequently, however, the room or building is planned and constructed with practically no consideration of this problem. The engineer of today is more likely than not to have as his problem a building that was constructed long before any consideration whatever was given to air conditioning it. Consequently, the room shapes, the location of columns and beams, and other details of architecture frequently make it difficult to properly locate the outlets. In general, for a cooling installation, the grilles should be located high enough from the floor to prevent the discharge of air directly upon the occupants of the room, and far enough down from the ceiling to minimize

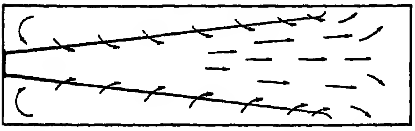


FIG. 1 PLAN VIEW LONG THROW SUPPLY OUTLET

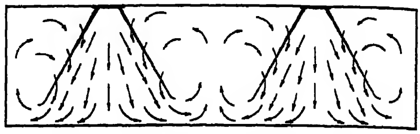


FIG 2. PLAN VIEW SHORT THROW SUPPLY OUTLETS

the possibility of streaking, and to permit induction of air from all sides of the stream. If the stream actually strikes the ceiling, but at a small angle, the throw will be increased somewhat if the ceiling is smooth. If the angle at which the stream hits the ceiling is 20 deg or more, or if the flow along the ceiling is obstructed by panel mouldings or beams, air velocity may be rapidly lost and a decreased throw result. The air stream also should be so directed that it will not strike nearby columns or beams in such a way as to cause misdirection of the air stream or drafts. Where the room is of irregular shape, as an ell, or where it has an alcove in one side, consideration should be given to obtaining satisfactory circulation in these corners. Frequently this cannot be done except by the use of multiple supply outlets. In using multiple outlets, care must be taken that the several air streams do not interfere with each other, until their velocities have been reduced to values which will not cause high turbulence and a drafty condition. Beams and offsets in the ceiling will cause little difficulty when substantially parallel to the direction of flow, unless they are of considerable depth, but when positioned across the air stream, may cause drafts and failure to secure satisfactory circulation in that portion of the room farthest from the outlet. In the case of a heating installation, down-drafts produced by such obstructions may not be serious, because the air will rapidly lose its downward motion, but the possibility of failure to obtain satisfactory circulation still exists.

The location of supply outlets should, if possible, be such as to take advantage of the maximum velocity permissible from a noise standpoint. For instance, the spaces illustrated in Figs. 1 and 2 may be satisfactorily served by either arrangement. However, by taking advantage of the long

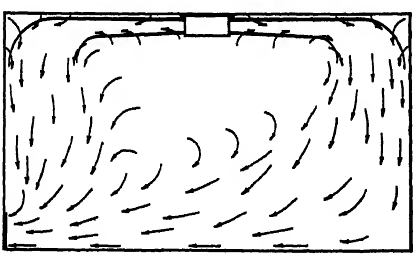


FIG. 3 ELEVATION VIEW CEILING SUPPLY OUTLET WITH RETURN WALL OUTLET

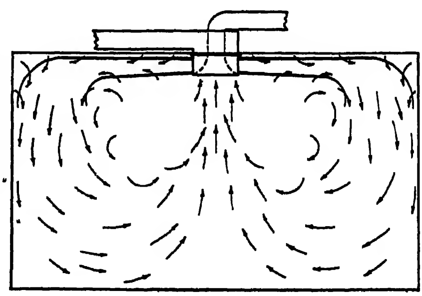


FIG. 4. ELEVATION VIEW CEILING SUPPLY AND RETURN OUTLET

throw, to which the arrangement in Fig. 1 lends itself, fewer outlets are required and additional savings are effected in the sheet metal work.

In solving the problem of properly conditioning a room of irregular shape, where multiple wall supply grilles are objectionable, a ceiling outlet of the type illustrated in Figs. 3 and 4 may very often be the best solution.

In choosing the most desirable location for the return air grille, consideration should be given to its effect on circulation of the air through the room. It is generally true that the return air grille should be placed on the same wall as the supply and near the floor level. This results in a

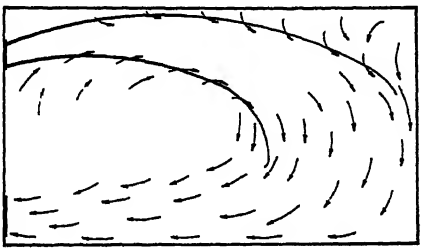


Fig. 5 Elevation View Correctly Located Return Outlet

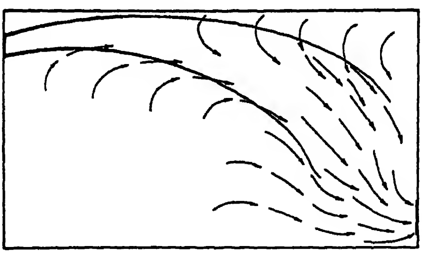


Fig 6. Elevation View of Improperly Located Return Outlet

U-shaped air path (Fig. 5) which covers the room thoroughly. The arrangement shown in Fig. 6 should be avoided, because it tends to create a stagnant section below the supply grille. What would otherwise be an unsatisfactory dead spot in a room may in some instances be taken care of by location of the return air grille near that area (Fig. 7).

STANDARDS FOR SATISFACTORY CONDITIONS

The most satisfactory air condition cannot be definitely stated for any particular individual without conducting a series of tests with that individual as subject; some persons are less sensitive than others to variations in temperature, humidity, air velocity and noise. The best

that can be done is to attempt to set limiting conditions leaning toward the values of these variables which produce a condition of comfort for the greatest number of individuals. On a cooling installation, the allowable deviation from average room temperature, that is, the temperature of puffs of air which may strike a person momentarily, is a function of the room temperature as well as the velocity of the air. For instance, in a room controlled at 72 F, a puff of air at 70 F might be uncomfortable to an individual, even at relatively low velocities, whereas if the average room temperature were 80 F, air at 78 F, even at moderate velocities, might be very satisfactory. However, air at 78 F in an average room temperature of 83 F would be cold. In general, other conditions being equal, for the range of temperatures normally encountered in living quarters on cooling installations, the permissible deviation from average room temperature varies from approximately 1 F at the low end of the range to about 3 F at the high end of the range. In this matter, it is important to consider the particular problem in the light of the type of occupancy. For instance, greater deviations from room temperature and higher velocities may be permitted in a garage or a hotel hallway than would be permissible in an office or living room. The velocity which may be considered the permissible maximum differs with the temperature deviation for a given installation, but an absolute maximum under any conditions might be considered that which would produce a mechanical disturbance, such as the movement of a person's hair or disturbance of papers on a desk. Humidity is an important consideration in the determination of one's feeling of comfort; however, if the room generally is assumed to be at a satisfactory value of relative humidity, the designer is justified in neglecting this factor when considering permissible fluctuations in temperature and velocity in the occupancy zone. This is true because the maximum allowable temperature fluctuation results in an unnoticeable humidity change.

The standards that might be set up for maximum allowable room temperature deviation and air velocity would not be the same for both heating and cooling installations. In the former case, any appreciable temperature deviation is likely to be above rather than below the average room temperature, whereas the reverse is most likely to be true on a cooling installation. Further, because air movement has a cooling effect in itself, the feeling of warmth due to temperatures above room temperature is counteracted to a certain extent so that an individual may be subjected to higher velocities of warm air without the feeling of discomfort occasioned by the same velocities of cool air. In every case, it should be the purpose of the designing engineer to keep the conditions within the zone of occupancy as nearly uniform as possible, securing minimum temperature deviations and low velocities. The air velocity at all points in the room should be at least 25 fpm for good results.

It is impractical to measure momentary temperature differences with any degree of accuracy in the field, but in checking a given installation it will generally be found satisfactory to measure velocity only, since on cooling installations high velocities normally occur with low temperatures, and on heating installations high velocities occur with high temperatures. That is, in the former case, the chilled supply air loses its velocity and undergoes an increase in temperature as it settles into the occupancy

zone, whereas in the latter case the heated supply air loses its velocity and undergoes a decrease in temperature during this process. Therefore, if the average velocities within the occupancy zone are not excessive, one is fairly safe in assuming that the temperature difference is also within permissible limits.

The subject of sound control is covered in Chapter 18 and it is recommended for detailed review before consideration of the problem of air outlet noise. An understanding of the relation between sound intensity and loudness level in decibels, as well as the effect of the presence of sound absorbent materials in the room, is particularly necessary. A more detailed discussion of the nature of this problem appears later; whereas the following comments refer to what constitutes a satisfactory noise condition

Obviously, the nature of the conditioned space is important when considering the allowable outlet noise. In factories, press rooms, and similar

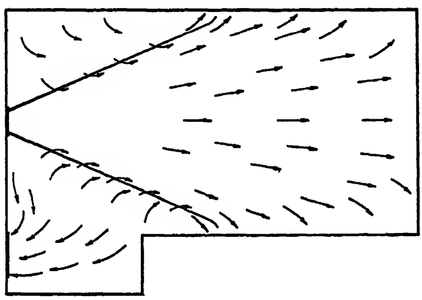


FIG 7 PLAN VIEW CORRECTLY LOCATED RETURN OUTLET ELIMINATING STAGNANT SPACE

spaces where the noise level is 65 db or higher, no complaints of grille noise are likely to be made. On the other hand, some homes, offices, hospitals, and, most of all, radio broadcasting and movie sound studios present a real problem which must be intelligently attacked if a satisfactory installation is to be made. In this chapter the noise of the air outlets (and returns) only is considered, it being assumed that the noise or sound level of the room without the outlet noise includes that which may be contributed by fans, motors, duct work, and other items of conditioning equipment. The control of noise from these sources is another problem (see Chapter 18). Where sound control is important, the actual room sound level without conditioning equipment should be known. If feasible, the contribution of the conditioning equipment, less outlets, should be estimated to secure the working sound level. If this correction is not made, the use of the first value errs in the direction of safety.

It is evident that the point within the room which should concern the designer in this problem is that at which the outlet noise is greatest. A tentative standard *listening point* relative to the outlet is suggested later in this discussion, and it is assumed that the outlet noise data are taken

with reference to this point. If it is desired that the outlet noise result in an inaudible addition to the existing noise level, it is safe to assume the total outlet noise to be 5 db below room level. This results in an increase in total noise of slightly over 1 db, which is unnoticeable. If an increase of 3 db is permissible, the outlet noise level may be equal to the room noise level alone. All outlets in the room must be considered, as will appear later, and the returns may be ignored only if they are so sized that the velocity of air through them is much less than through the outlet.

DISTRIBUTION FACTORS IN ROOM COOLING

In attempting to design a satisfactory air distributing system, it is first necessary to properly locate the grilles in accordance with the recommendations already stated. Assuming that the best locations have been selected, it then becomes necessary to choose the proper grille for that location. The considerations involved are the amount of air to be handled, the velocity permissible from the standpoint of noise, and the distance the air should carry. The distance it will carry, assuming no obstructions, is affected by a number of factors which are listed below:

1 The temperature difference between incoming and room air.
2 Height of grille above floor
3 Face velocity.
4 Core area
5 Core aspect ratio.
6 Design of grille.

The manner in which the above factors affect throw may be generally stated. All other things being constant, a lower temperature of incoming air will result in shorter throw; a greater height above the floor will affect a longer throw; a higher velocity will produce a longer throw; greater area will give longer throw; larger aspect ratio will decrease throw. The variation in throw with type of outlet will, of course, depend upon the design characteristics of the outlet.

In consideration of what constitutes the possible throw of an outlet under a given set of conditions, it is important to remember that the throw may be unsatisfactory for any one of several reasons:

1 It may be so long that it will strike the far side of the room and come down the wall with velocities higher than are permissible,

2. It may be so short that it will fail to carry the full length of the room, and short-circuit to the return air outlet, or

3. It may spill into the center of the room

In the first case, the system fails for lack of uniform distribution and the presence of cold areas. In the second case, the standards as to velocity and temperature difference in the zone of occupancy may be satisfactorily met, but air distribution and circulation throughout the entire room is not accomplished, with the result that the end of the room away from the outlet would not be satisfactorily conditioned. In the third case, the shortcomings of both case one and case two are present. It is evident, therefore, that for a given outlet discharging air at a given velocity, there is a maximum and a minimum length of room which can be satisfactorily

handled. In the latter, the velocity of the air down the far wall is just within the maximum permissible, while in the former, satisfactory circulation is barely accomplished.

In general, the higher the outlet is above the floor, the greater may be the difference between room air and incoming air temperatures.

Assuming that proper supply outlets for a given installation have been selected, unsatisfactory performance may still result due to the construction of the duct work immediately back of the outlets. Performance data on the grilles and registers of various manufacturers should be based upon results obtained with the air approaching the grille perpendicularly and at uniform velocity over the entire duct cross-section. Where this condition does not exist in practice, performance predictions based on published data cannot be expected to be realized. Every precaution should be taken to secure as nearly ideal conditions in the approaching air stream as are possible.

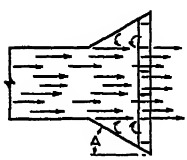


FIG. 8. EFFECTS OF EXPANDING DUCT

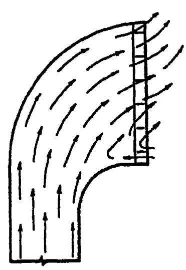


FIG 9 UNEQUAL FACE VELOCITIES

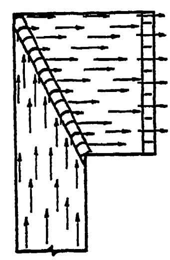


FIG. 10. EFFECT OF TURNING MEMBER

In addition to disturbances due to the construction of the duct work itself are those which may be created by dampers immediately behind the grille. Where either multiple louvre or single blade dampers are used, considerable deflection of the air stream may result, if it is throttled appreciably by these means. This is particularly true when the fins of the register core are perpendicular to the damper blades. If the core has sufficient depth and the fins are parallel to the blades, there is a marked tendency to straighten the air stream, although some deflection may still result.

Any attempt to secure a low face velocity and high duct velocity by the construction of any expanding chamber immediately behind the grille is very likely to be unsuccessful. In order to expand from a small duct to a larger one, and have the air stream fill the duct at the end of the diverging section without turbulence, angle *A* in Fig. 8 should be about 3 deg for four-sided expansion and about 5 deg for two-sided expansion. From this it is apparent that an attempt to secure equivalent results with a short connection would be futile. What actually happens when this is attempted is illustrated by the arrows in Fig. 8. When localized high velocities through the outlet exist from this cause or any other, the noise produced will naturally exceed that which the outlet area and average face velocity would lead one to expect. This fact should be remembered

in considering the use of register dampers, particularly in those cases where there must be considerable throttling with the damper to balance a poorly designed system. Where reduction of noise is important, it is recommended that balancing dampers be placed in the duct *ahead* of the acoustic duct lining.

Similar unequal face velocities, aggravated by a deflection of the air stream, are obtained with the arrangement shown in Fig. 9. The latter may be corrected by inserting a turning member in the elbow back of the outlet face as shown in Fig. 10 The importance of straightening the air stream and affecting uniform distribution over the entire face of the outlet cannot be over-emphasized.

DISTRIBUTION FACTORS IN ROOM HEATING

The problem in the case of a heating installation is substantially the same as in cooling, with a few exceptions. Because the temperature of the incoming air is above that of the room, there is no tendency for it to drop and consequently the throw is not particularly affected by temperature difference in a low ceiling room In general, the air should be deflected downward where the grille is above the occupancy zone, and this is particularly desirable where the ceiling is high. For the same reason, that is, to keep the heat in the occupancy zone and to avoid excessive temperature at the ceiling, it is desirable to have the grille comparatively low on the wall, and just slightly above the occupancy zone If the grille is lower than this, it may create an unsatisfactory condition of very warm air at quite high velocities where it can possibly strike the occupants of the room. Where the velocities are very low, the grilles may even be satisfactorily located below the 6 ft level, although the immediate vicinity of the supply outlets will probably be useless for occupancy because of high temperature. Essentially, the problem is to keep the incoming air up for cooling, and down for heating, until it is thoroughly mixed with the room air. Grilles and registers which are adjustable for deflection upward and downward, either by moving the fins or inverting the grille, are in general use.

AIR OUTLET NOISES

When air is introduced into a room through a grille or register at a constant velocity, sound energy is being introduced into the enclosure at a constant rate. Due to partial reflection at the boundaries of the enclosure, the intensity of sound at any point in the space builds up to some maximum value. In a large room at a point remote from the source of sound (the outlet) the intensity can be shown to be substantially proportional to the rate at which sound energy is generated and inversely proportional to the number of sound absorption units (sabins) in the room. It would thus appear that doubling the sound absorption of the room would halve the intensity and result in a noise level decrease of 3 db. However, it is not satisfactory to consider the grille noise on this basis (wherein the sound power received directly from the source is small compared with that received by reflection) since in practice the occupants of the room may be quite close to the grille. The nearer the listener is to

the sound source, the greater the proportion of the sound intensity which is due to direct transmission.

In the absence of generally accepted standards at this time it is suggested that the loudness level 5 ft from the lower edge of the outlet, measured downward at 45 deg in a plane perpendicular to the outlet at its center, represents about the maximum within the zone of occupancy The cases where persons are nearer to the outlet than this are rare and are ignored in the consideration of this problem Although the effect of sound absorbent material on the intensity at the 5 ft station is not nearly so great as at more remote points in the room, it should not be ignored without consideration of the error involved. An average living room may contain 100 sabins (absorption units). If this be decreased to 50 sabins, the *diffuse* or reflected sound level would be increased 3 db. However, at the 5 ft station the increase would be less than 2 db If the absorption of the room be increased to 200 sabins, one might expect a reduction in diffuse noise of 3 db; but at the 5 ft station the reduction would be less than 1½ db. Furthermore, even though the absorption be increased without limit (as in free space) the reduction would still be less than 2 db because of proximity to the source.

In comparing sound ratings of various grilles, the following must be known if the information is to be intelligently applied:

1 The threshold intensity on which the decibel ratings are based.

2 The distance from the grille at which data were taken

3 If stated as loudness level versus velocity for a given grille, the *core area* (not nominal area) must be known

4 The sound absorbing characteristics of the test room

5 Whether or not corrected for test room loudness level, if not, the room level (without grille noise) must be known.

6 Methods used for recording data.

Data mentioned in this chapter are assumed to have been referred to the following:

1 Threshold intensity = 10^{-16} watts per square centimeter[1]

2 Microphone location 5 ft from lower edge of outlet on a line downward at 45 deg and in a plane bisecting the outlet perpendicularly.

3. Where data are given as loudness level versus velocity, the rating is per square foot of core area

4 The room is assumed to have 100 sabins absorption

5. Plotted data are loudness levels of *outlets only*, correction having been made for test room level

6. Data taken with a direct reading sound-level meter with frequency weighting network intended to approximate the response of the human ear.

If the published ratings are in terms of decibels per square foot, correction must be made for area to secure to total sound level of outlets of more or less than one square foot area. This can be done by use of the following formula:

$$\text{Decibel Addition} = 10 \log_{10} A \qquad (1)$$

[1]American Tentative Standards for Noise Measurement, *American Standards Association*

where: A = core area, square feet

In practice the allowable total sound and the required air flow are usually known, and it is desired to determine the maximum allowable velocity. Since total loudness and air flow are both functions of velocity and area, the solution of the problem by use of the previous analysis implies a trial and error method. It has been found possible to present these data with sufficient practical accuracy as a family of uniform curves as illustrated in Fig. 11. With this chart it is possible to find directly the velocity in feet per minute which will give a predetermined total loudness at a predetermined rate of flow expressed in cubic feet per minute. The values used are arbitrarily chosen for the purpose of discussion and do not necessarily represent data referring to any particular make of grille,

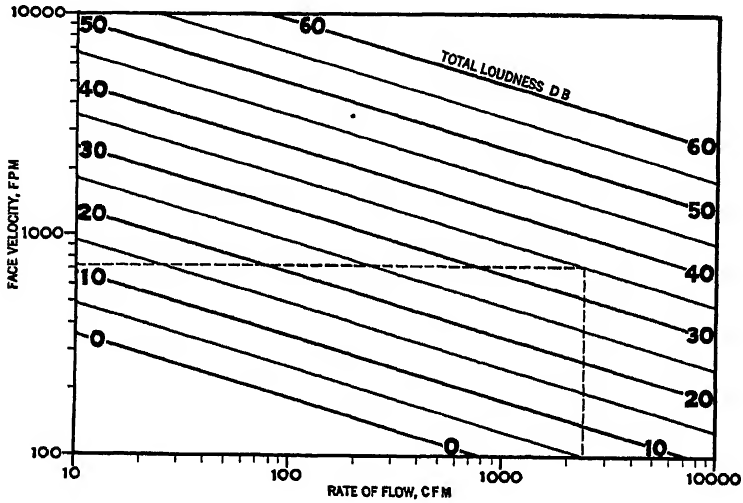


Fig. 11 Air Flow and Loudness Chart

register or air outlet. It is assumed that Fig. 11 is based on a room having 100 sabins of sound absorption. In such a room the sound level due to other sources may be 35 db. As previously stated an outlet having a noise level of 30 db would be substantially inaudible in such a room.

If 1000 cfm are required with a total noise due to outlet of 30 db, a velocity (Fig. 11) of about 675 fpm may be used. From this velocity and the rate of flow, the core area can be computed. This determination was on the basis of a room absorption of 100 sabins. If the absorption is greater, the 675 fpm velocity is safe, since the loudness level will go down. However, correction can be made if desired by the use of the chart of Fig. 12. Thus, if the absorption is 200 sabins, a correction of +1.3 db may be made and the permissible velocity becomes that corresponding to a total loudness level of 31.3 decibels or approximately 750 fpm. If the room is highly reflecting and has an absorption of less than 100, correction is much more important. For instance, for 35 sabins a correction of −3 db

must be made and the maximum velocity corresponding to 27 db total loudness chosen; that is, approximately 550 fpm.

Where more than one outlet must be considered, the problem is more complicated. If a similar outlet is added in a far corner of a highly absorbent room, the change in noise level at the 5 ft station at the first

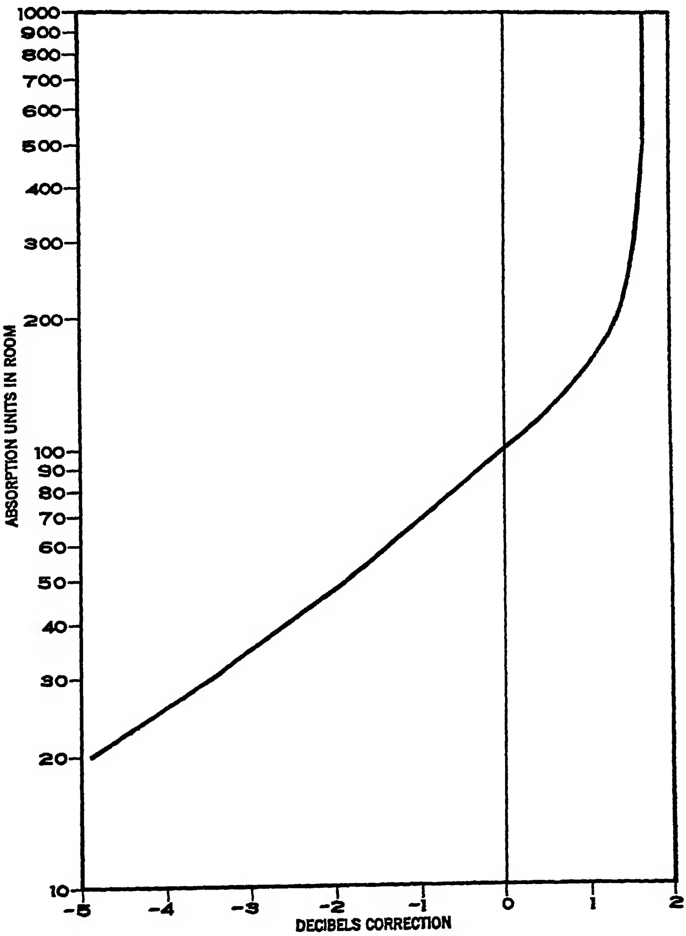


FIG 12 ROOM ABSORPTION CORRECTION CHART

outlet is small; however, if the room is small, or highly reverberant or both, the intensity at the 5 ft station may be almost doubled and the noise level increased nearly 3 db thereby. The simplest method of handling this problem, and one which errs in the direction of safety, is to treat the room as though all the air were being supplied by one outlet. Thus, if two outlets, each supplying 1000 cfm are used, the value 2000 cfm

should be used with Fig. 11. Although this method may place an unwarranted limit on velocity when used in a large room, it is seldom that such a room has a noise level low enough to make this penalty serious or to justify a more complicated though more exact procedure.

In general, return grilles are selected for velocities about half the supply velocity, and when this is done, they may be neglected in sound computations. However, if supply and return grilles are the same size, resulting in the same face velocity, they must be treated as two supply outlets. That is, if 1000 cfm is supplied and exhausted through grilles of the same area, 2000 cfm must be used in the solution with Fig 11.

SELECTION OF SUPPLY OUTLETS

After the heating and cooling load calculations have been made (Chapters 7 and 8), and, or a suitable supply air temperature selected, the

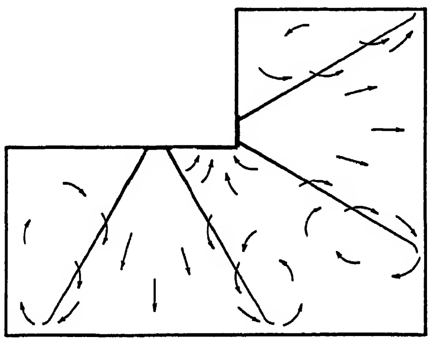


Fig 13. Plan View Typical General Office

volume of air required for each space can be determined. The next step is to determine the velocity at which the air may be introduced into the space quietly and without creating objectionable drafts.

Present-day grille design coupled with the introduction of effective acoustical treatment for minimizing fan and duct noises have made grille face velocities in excess of 1500 fpm feasible, and 600 to 1200 fpm is now used in practice. This range of velocities is approximately three times higher than common practice values of a few years ago.

Since high velocities make for smaller ducts and outlets, and therefore savings in space as well as greater flexibility in locating the duct work to the best advantage, selection of design velocities is a very important step.

The selection of proper velocity requires that the designer have before him reliable data applicable to the particular make of outlet he proposes to use. Even under these circumstances, the problem is one of *cut and try* because permissible velocity may be determined by either noise or throw.

A method for selecting supply outlets is outlined below in the form of a sample cooling problem, using numerical values which have no reference to any particular make of outlet.

1 The load calculations have been made, a suitable temperature differential has been selected (it is to be understood that the data referred to from this point on are based on this temperature differential), and the volume of air required determined Assume that Fig 13 represents a small general office having a noise level of 40 db and that 2500 cfm must be supplied for proper conditioning

2 Select a tentative location for the outlet or outlets, having in mind the type of grille most likely to effect proper distribution In this particular case, two outlets having a wide spread appears to be a logical choice

3 Data from which to determine velocity which corresponds to 2500 cfm and a noise rating at least 5 db below the noise level of the office may be presented in a number of forms, one of which is shown in Fig 11 (Fig 11 represents assumed values only In practice similar data should be obtained from the manufacturer whose outlets are being considered Several similar charts or tables may be necessary to cover any one manufacturer's complete line) From Fig 11 it will be noted that for 2500 cfm the type of grille selected may be used at velocities up to 700 fpm without exceeding 35 db, that is, 5 db below the noise level of office

4 Having determined the velocity, the core area becomes fixed at 3 57 sq ft or 257 sq in per outlet In this problem, the two grilles in question are so close together that consideration of their combined area in determining the permissible velocity from the standpoint of noise introduces little error.

5 The type grille selected has thus far been found satisfactory from a noise standpoint, provided the face velocity does not exceed 700 fpm The next consideration is throw, which may be assumed to be 16 ft, and by reference to a manufacturer's catalogue the proper correlative test data may be checked with the throw assumed It is of course evident that one or more types of grilles may satisfy the requirements, and that in any one type there will be a choice of outlet proportions It will also be evident that the tentative selection of an outlet having a wide spread may be unsatisfactory from the standpoint of throw, in which event a second choice should be made and the procedure repeated

In the case of a heating problem, the method of solution is the same, but the manufacturer's data must, of course, be based on tests with air above room temperature.

TYPES OF SUPPLY OUTLETS

Grille, registers or outlet design for attaining uniform distribution and minimum air resistance consists of various fixed and adjustable arrangements. Some types are designed with directing air blades, fins, bars, louvres, or thin metal strips shaped into a series of grooves or tubes, all of which may be set into a suitable round, square or rectangular frame. In order to attain desired long or short air throws, the emergence of air from the outlet may be directed to straight, deflecting, converging or jet air streams depending upon the outlet design. Designs which direct the air stream to produce an ejector effect within the enclosed space tend to mix the room air with the conditioned air to provide uniform distribution.

Centrally located ceiling or wall type outlets arranged for completely diffusing the air consist of several round; hollow, cone-shaped flaring members placed in the proper relationship to each other. The velocity of emergence of the air from the unit can be made practically uniform over the entire surface of the outlet, and the velocity in any direction may be varied to any desired value by adjusting the position of the cones. One or more of the smaller flaring members act as ejectors and injectors which draw a small proportion of the room air into the air spreader where it mixes with the conditioned air before it is discharged.

An idea for producing even distribution of air consists of a perforated ceiling made of a suitable architectural surface and installed a small distance below the normal ceiling level of the room. In the space provided by this suspended ceiling a plenum chamber is formed into which the conditioned air is introduced. From the plenum space the air is permitted to diffuse through the large number of small ceiling openings into the room

Railroad Cars

In the early practice of air conditioning railroad passenger cars a system of bulkhead distribution was used. This consisted of an inlet opening at each end of the car discharging the air toward the middle with the flow parallel to the long dimension of the car. This installation resulted in drafts in the middle of the car and was considered unsatisfactory. Later designs incorporate a duct delivery system on each side of the car roof directing the air through numerous inlet openings toward the middle of the car where the two air streams come in contact and deflect downward, gradually filtering into the aisle. At the present time, several center duct air distribution systems are used in railroad car applications. In some instances square or circular ceiling outlets connected to a center duct have been used, which distribute the air along the ceiling in widening circles and at right angles to the inlet opening. Another method consists of a continuous slot in the bottom of the duct to which is attached a flat plate so that the air is deflected along the car ceiling. Connected to the central supply system are small individual ducts to each sleeping compartment with adjustable directional outlets to control the desired air flow.

BALANCING SYSTEM

In designing an air conditioning system, it should be the aim of the engineer to so proportion the duct system that proper distribution of air to every outlet will be obtained. Since this is almost impossible to accomplish in practice, it becomes necessary to have means of balancing the system to secure the desired amount of air in each space. There are a number of ways in which this may be accomplished, some of which are listed:

1. Dampers on the supply grilles.
2. Dampers on the return grilles.
3. Dampers in the supply ducts.
4. Dampers in the return ducts.
5. Reducing the effective area of some outlets by blank-offs.
6. Combinations of dampers in both supply and return air.

Dampers on the supply grilles themselves are objectionable because of their effect on the air stream. Dampers on the return grilles are frequently helpful in building up a static pressure in the room to prevent infiltration of outside air, and at the same time reduce the volume of incoming air. However, it is frequently impossible to sufficiently reduce the incoming air by this method alone. A damper in the supply duct some distance

back of the outlet forms a very satisfactory means of regulating the flow without disturbing distribution across the outlet face. A damper in the return air duct has the advantage over one immediately behind the grille in that it does not tend to create high localized velocities through the grille as the latter might do if nearly closed. Blank-offs consisting of pieces of sheet metal covering a portion of the outlet face can frequently be used satisfactorily, although determination of just what is required is a matter of experiment, and the balancing of the system is not nearly so conveniently accomplished as with dampers. Dampers in both supply and return air form the most flexible means of controlling the supply to the room and the static pressure within the room. When feasible, these dampers, particularly those in the supply ducts, should be a substantial distance from the outlet, and ahead of the acoustic duct lining if used. Due consideration should also be given to the use of the several volume control and uniform distribution devices now available. See *Catalog Data Section*.

REFERENCES

Methods of Air Distribution, by L. L. Lewis (A.S.H.V.E. TRANSACTIONS, Vol. 34, 1928).

Air Supply, Distribution and Exhaust Systems, by S. R. Lewis (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, October, 1932).

Characteristics of Registers and Grilles, by J. H. Van Alsburg (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, June, 1935).

PROBLEMS IN PRACTICE

1 ● What important factors are involved in the correct distribution of air to an enclosed space?

Not only is it important to distribute the air from the fan to the various spaces served by the system, but also the air must be properly distributed within the enclosed space to give complete satisfaction.

2 ● Upon what basis should the selection of supply outlets be based?

If possible, the selection should take advantage of the maximum velocity permissible from a noise standpoint.

3 ● What factors in grille design effect the length of air throw?

a. The temperature difference between incoming and room air, *b.* height of grille above floor, *c.* face velocity, *d.* core area, *e.* core aspect ratio, and *f.* design of grille.

4 ● How does the height of a supply outlet affect the temperature differential within a room?

In general, the higher the outlet is above the floor, the greater may be the difference between room air and incoming air temperatures.

5 ● Under conditions prevalent in a large room, how does the intensity of sound develop at an air outlet vary?

The intensity of sound energy is substantially proportional to the rate at which sound energy is generated and inversely proportional to the number of sound absorption units in the room.

6 ● What are the essential differences between a high velocity long throw and short throw grille?

Generally, a high velocity long throw grille is used where a large compact mass of air is projected with a reduction in the periphery of the air stream whereas, with a short throw grille design the periphery of the air stream is expanded as much as possible to increase the scrubbing action between the incoming air stream and the stationary air

7 ● What type of system is generally used in a large continuously operated theatre?

Most large continuously operated theatres are provided with a complete downward system of air distribution With this system a large number of outlet openings are provided each of which discharges air in a thin horizontal stream at high velocity in order that the cool air would be mixed with the area in the theatre before it reaches the patrons In this type of system the best distribution is obtained when a sufficient number of exhaust openings are located under the seats

8 ● What means are available for balancing a system to secure the desired amount of air in each space?

Ways in which this may be accomplished are by *a* dampers on supply and return grilles, *b*. dampers in supply and return ducts, *c* reduction of the effective area of some outlets by blank-offs, and *d* combination of dampers in both supply and return air duct systems

Chapter 20

AIR DUCT DESIGN

Pressure Losses, Friction Losses, Friction Loss Chart, Proportioning the Losses, Sizes of Ducts, General Rules, Procedure for Duct Design, Air Velocities, Proportioning the Size for Friction, Main Trunk Ducts, Equal Friction Method, Duct Construction Details

THE flow of air due to large pressure differences is most accurately stated by thermodynamic formulae for air discharge under conditions of adiabatic flow, but such formulae are complicated, and the error occasioned by the assumption that the gas density remains constant throughout the flow may be considered negligible when only such pressure differences are involved as occur in ordinary heating and ventilating practice.

In the development of the formulae, diagrams, and tables for the flow of air, use is made of the following basic equation for the flow of fluids:

If H_V be the velocity head in feet of a fluid, and the velocity, V, be expressed in feet per minute, the fundamental equation is

$$V = 60\sqrt{2g\,H_V}$$

The factor g is the acceleration due to gravity, or 32 16 ft per second per second.

It is usual to express the head in inches of water for ventilating work and, since the heads are inversely proportional to the densities of the fluids,

$$\frac{H_V}{\frac{h_V}{12}} = \frac{62.4}{d}$$

or

$$H_V = 5.2\,\frac{h_V}{d}$$

therefore,

$$V = 1096.5\sqrt{\frac{h_V}{d}} \tag{1}$$

where

V = velocity in feet per minute.
h_V = velocity head or pressure in inches of water.
d = weight of air in pounds per cubic foot.

For standard air (70 F and 29 921 in. barometer) $d = 0.07492$ lb per cubic foot. Substituting this value in Equation 1:

$$V = 1096.5\sqrt{\frac{h_V}{0.07492}} = 4005\sqrt{h_V} \tag{2}$$

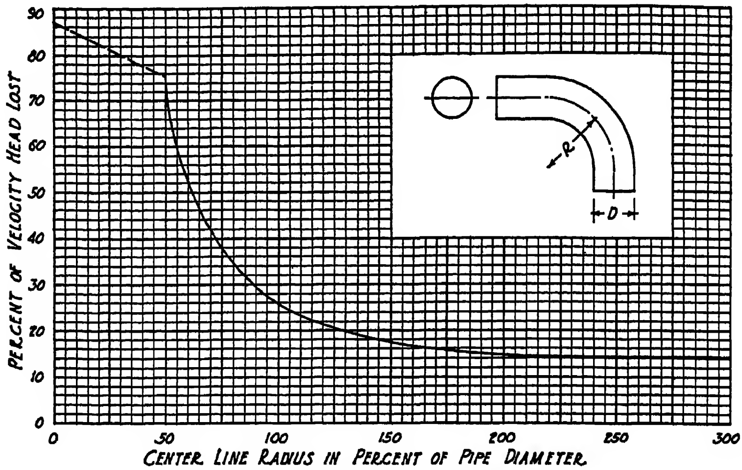


FIG. 1. CURVE SHOWING LOSS OF PRESSURE IN ROUND ELBOWS

The drop in pressure in air distributing systems is due to the *dynamic* losses and the *friction* losses. The friction losses are those due to the friction of the air against the sides of the duct. The dynamic losses are those due to the change in the direction or in the velocity of air flow.

Pressure Losses

Dynamic losses occur principally at the entrance to the piping, in the elbows, and wherever a change in velocity occurs. The entrance loss is the difference between the actual pressure required to produce flow and the pressure corresponding to the flow produced; it may vary from 0.1 to

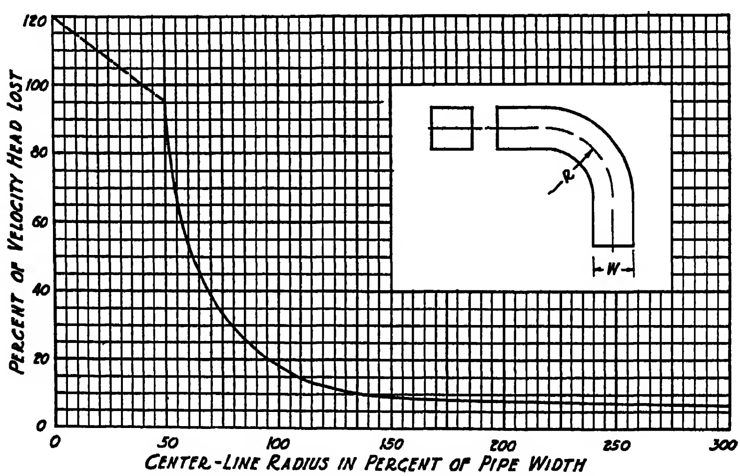


FIG. 2 CURVE SHOWING LOSS OF PRESSURE IN SQUARE ELBOWS

0.5 times the velocity head. The pressure loss in elbows must also be allowed for in the design. It is customary to express dynamic losses in terms of the percentage of the velocity head; in other words, the percentage of that pressure corresponding to the average velocity in the duct which is expressed in terms of inches of water gage. Figs. 1 and 2 show the effect of changing the radius of elbows of square and rectangular section[1]. These charts are based on tests of pipe elbows of ordinary good sheet metal construction. For example, a five-piece round pipe elbow having a centerline radius of one diameter has a loss of about 25 per cent of the velocity head. At a velocity of 2000 fpm the corresponding head is 0.25 in. water gage, and at this velocity the elbow just referred to would cause a pressure drop of 0.063 in. water gage. Experience has shown that good results may be obtained when the radius to the center of the elbow is 1½ times the pipe diameter. The pressure drop will then be approximately 17 per cent of the velocity head for round ducts, and 9 per cent for square ducts. Very little advantage is gained in making elbows with a radius of more than two diameters[2].

Friction Losses

Friction losses vary directly as the length of the duct, directly as the square of the velocity, and inversely as the diameter. Since length is a fixed quantity for any system, the factors subject to modification are the area and the velocity, which determine the relation between the first cost of the duct system and the cost of the power for overcoming friction.

The friction between the moving air and pipe surface causes a loss of head which is numerically equal to the pressure required to maintain a given velocity, and is expressed in the following modification of Fanning's formula:

For round pipe and standard air (70 F and 29.921 in. barometer)

$$h_L = f\frac{L}{D}h_v = \frac{L}{CD}\left(\frac{V}{4005}\right)^2 \qquad (3)$$

For rectangular ducts

$$h_L = fL\left(\frac{a+b}{2ab}\right)h_v = \frac{L}{C}\left(\frac{a+b}{2ab}\right)\left(\frac{V}{4005}\right)^2 \qquad (4)$$

where

h_L = loss of head, inches of water.

$h_v = \left(\frac{V}{4005}\right)^2$ = velocity head, inches of water

V = velocity of air, feet per minute.

L = length of pipe
D = diameter of pipe
a, b = sides of rectangular duct } all in feet.

f = coefficient of friction.

$C = \frac{1}{f}$ = length of pipe in diameters for one head loss

For all practical purposes C varies only with the nature of the pipe surface. C = 60 for perfectly smooth pipe; = 55 for pipe as used in planning

[1]Loss of Pressure Due to Elbows in the Transmission of Air Through Pipes or Ducts, by F. L. Busey (A.S.H.V.E. TRANSACTIONS, Vol. 19, 1913)

[2]Pressure Losses in Rectangular Elbows, by R. D. Madison and J. R. Parker (*Heating, Piping and Air Conditioning*, July, August, September, 1936)

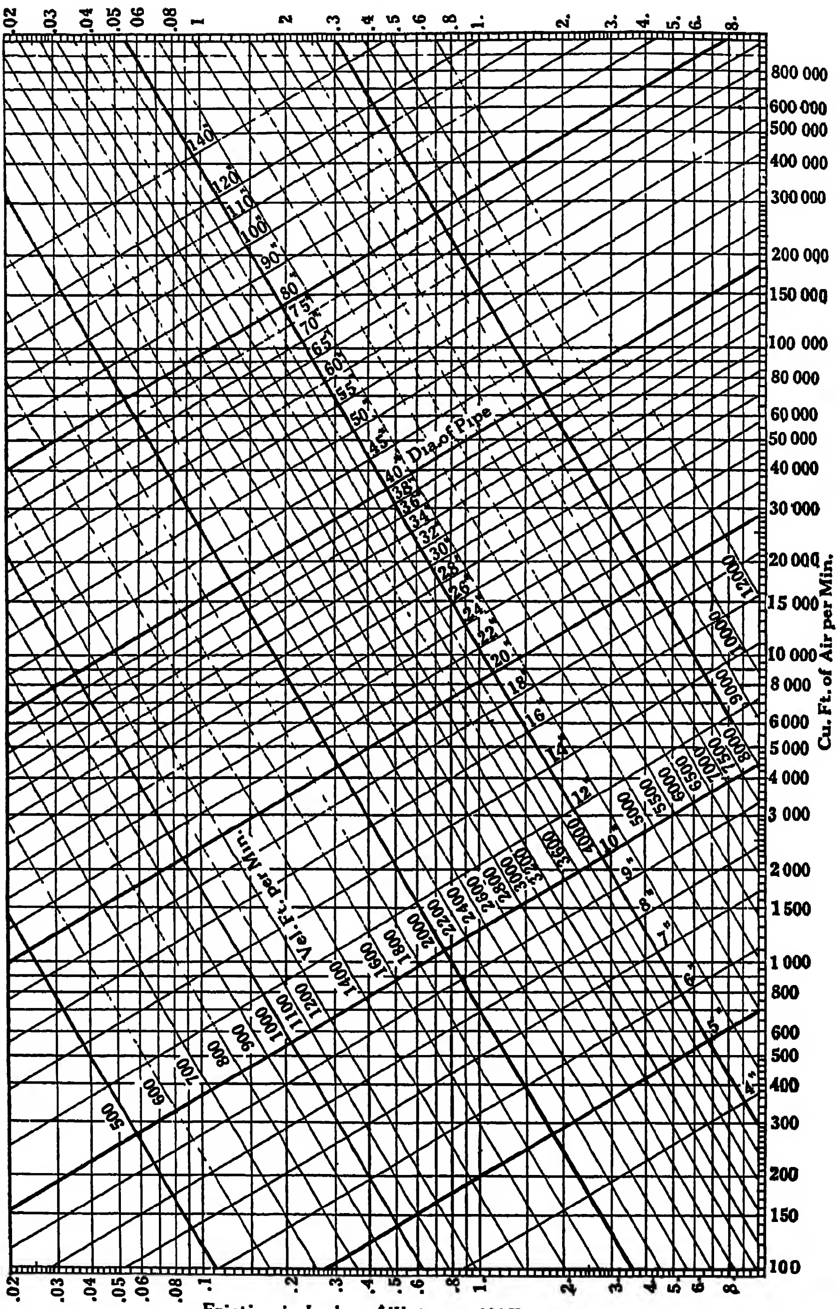


FIG. 3. FRICTION OF AIR IN PIPES

mill exhaust systems; = 50 for heating and ventilating ducts; = 45 for smooth and 40 for rough conduits of tile, brick or concrete. However, Fritzsche states (and numerous tests check very closely) that f varies inversely as the 2/7 power of the pipe diameter, and inversely as the 1/7 power of the velocity, or inversely as the 1/7 power of capacity, which is the same thing. Thus Formula 3 may be revised as follows, based upon a loss of one velocity head (at 2000 fpm) in a length equal to 50 diameters of 24-in. galvanized swedged pipe.

$$h_L = 1.1 \frac{L}{CD^{9/7}} \left(\frac{V}{4005}\right)^{13/7} \tag{5}$$

The preceding formulae are based on standard air, and for other conditions the friction varies directly as the air density and inversely (approximately) as the absolute temperature. The increase of friction due to increase of air viscosity with increased temperature is small and is generally neglected.

Friction Loss Chart

Fig. 3 is a convenient chart for determining the friction loss for various air quantities in ducts of different sizes. The general form of this chart is familiar, but it should be noted that it is corrected for changes in the coefficient of friction based on the rule that the coefficient of friction varies inversely as the 2/7 power of the diameter, and inversely as the 1/7 power of the velocity. Fig. 3 is based on a loss of one velocity head (at a velocity of 2000 fpm) in a length equal to 50 diameters of 24-in. round galvanized-iron duct of the usual construction. Although this chart is laid out for a value of C equivalent to 50, it may be used for other values of C by varying the friction inversely as this constant. For example, if a rougher pipe is used with 40 as the value of C, the friction loss as read from the chart should be multiplied by $\frac{50}{40}$.

Example 1. Assume that it is desired to pass 10,000 cfm of air through 75 ft of 24-in. diameter pipe. Find 10,000 cfm on the right scale of Fig. 3 and move horizontally left to the diagonal line marked 24-in. The other intersecting diagonal shows that the velocity in the pipe is 3200 fpm. Directly below the intersection it is found that the friction per 100 ft is 0 59 in., then for 75 ft the friction will be 0 75 × 0 59 = 0 44 in. In a like manner any two variables may be determined by the intersection of the lines representing the other two variables.

Proportioning the Losses

Other losses of pressure occur at the entrance to the duct, through the heating units, and at the air washer. In ordinary practice in ventilation work it is usual to keep the sum of the duct losses ⅓ to ½ and the loss through the heating units at less than ½ of the static pressure. The remainder is then available for producing velocity. In the design of an ideal duct system, all factors should be taken into consideration and the air velocities proportioned so that the resistance will be practically equal in all ducts regardless of length.

SIZES OF DUCTS

The sizes of ducts and flues for gravity or mechanical circulation of air are usually based on the losses due to friction, and these losses must be

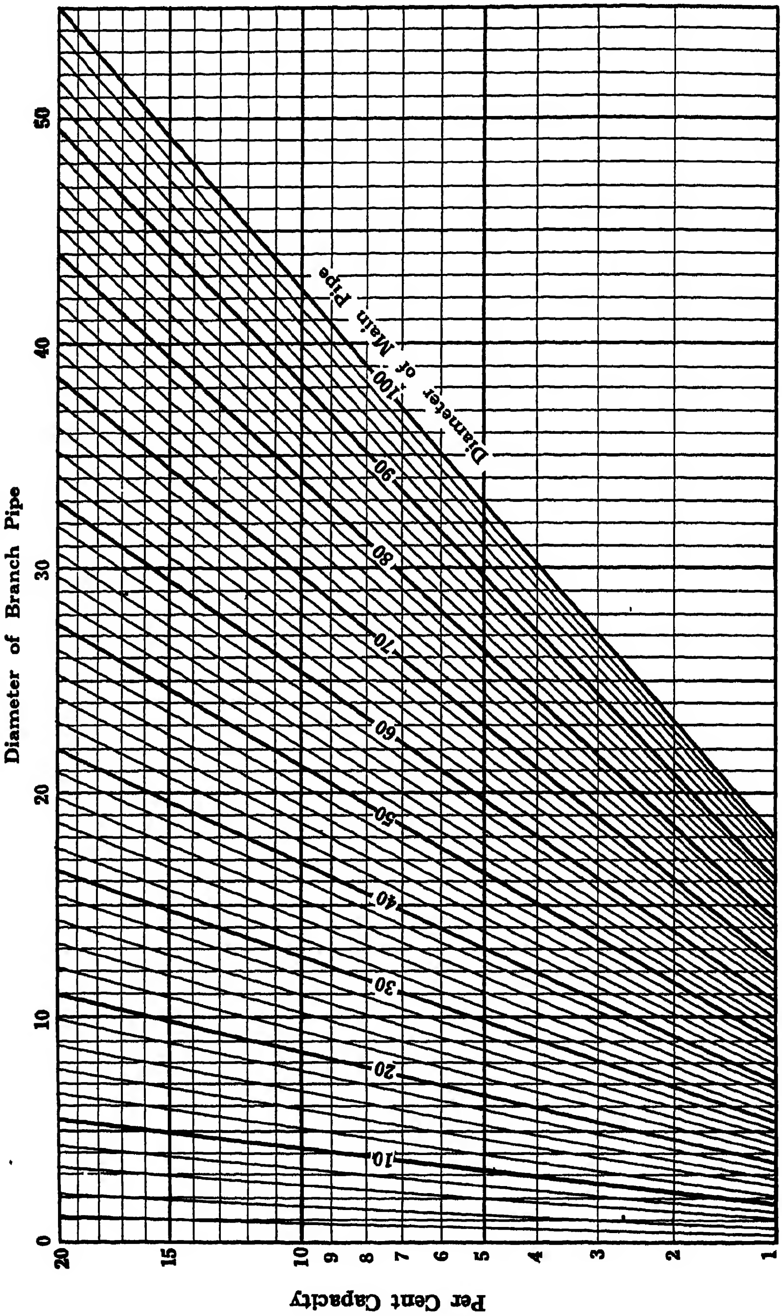


Fig. 4. Main and Branch Pipes for Equal Friction per Foot of Length (1 to 20 Per Cent Capacity)

kept within the available pressure difference. This pressure difference in mechanical ventilation is that derived from the fan, while in gravity ventilation the aspirating effect due to the temperature and height of the column of heated air causes the pressure difference.

General Rules

The general rules to be followed in the design of a duct system are:

1. The air should be conveyed as directly as possible at reasonable velocities to obtain the results desired with greatest economy of power, material and space.
2. Sharp elbows and bends should be avoided.
3. The sides of all ducts or flues should be as nearly equal as possible (In no case should the ratio between long and short sides be greater than 10 to 1.)

Procedure for Duct Design

The general procedure for designing a duct system is as follows:

1. Study the plan of the building and draw in roughly the most convenient system of ducts, taking cognizance of the building construction, avoiding all obstructions in steel work and equipment, and at the same time maintaining a simple design.
2. Arrange the positions of duct outlets to insure the proper distribution of heat.
3. Divide the building into zones and proportion the volume of air necessary to supply the heat for each zone.
4. Determine the size of each outlet, based on the volume as obtained in the preceding paragraph, for the proper outlet velocity.
5. Calculate the sizes of all main and branch ducts by either of the following two methods:

   a. *Velocity Method.* Arbitrarily fix the velocity in the various sections, reducing the velocity from the point of leaving the fan to the point of discharge to the room. In this case the pressure loss of each section of the duct is calculated separately and the total loss found by adding together the losses of the various sections.

   b. *Friction Pressure Loss Method.* Proportion the duct for equal friction pressure loss per foot of length

6. Calculate the friction for the duct offering the greatest resistance to the flow of air, which resistance represents the static pressure which must be maintained in the fan outlet or in the plenum space to insure distribution of air in the duct system. The duct having the greatest resistance will usually be that having the longest run, although not necessarily so.

Air Velocities

The following velocities of air are considered standard for public buildings:

1. Through the outside air intakes, 1000 fpm.
2. Through connections to and from heating unit, 1000 to 1200 fpm.
3. Through the main discharge duct, from 1200 to 1600 fpm.
4. In branch ducts, 600 to 1000, and in vertical flues, 400 to 800 fpm.
5. In registers or grilles, 200 to 400 fpm depending upon the size and location. If diffusers of proper design are used, 25 per cent higher air velocities are permissible.

These duct velocities may safely be increased 20 per cent if first class construction is used to prevent any breathing, buckling, or vibration. High velocities at one point in the system neutralize the effect of proper design at all other points; hence the importance of splitters in elbows and similar precautions. For industrial buildings noise is seldom considered, and main duct velocities as high as 2800 or 3000 fpm may be used where conditions will permit. For department stores and similar buildings,

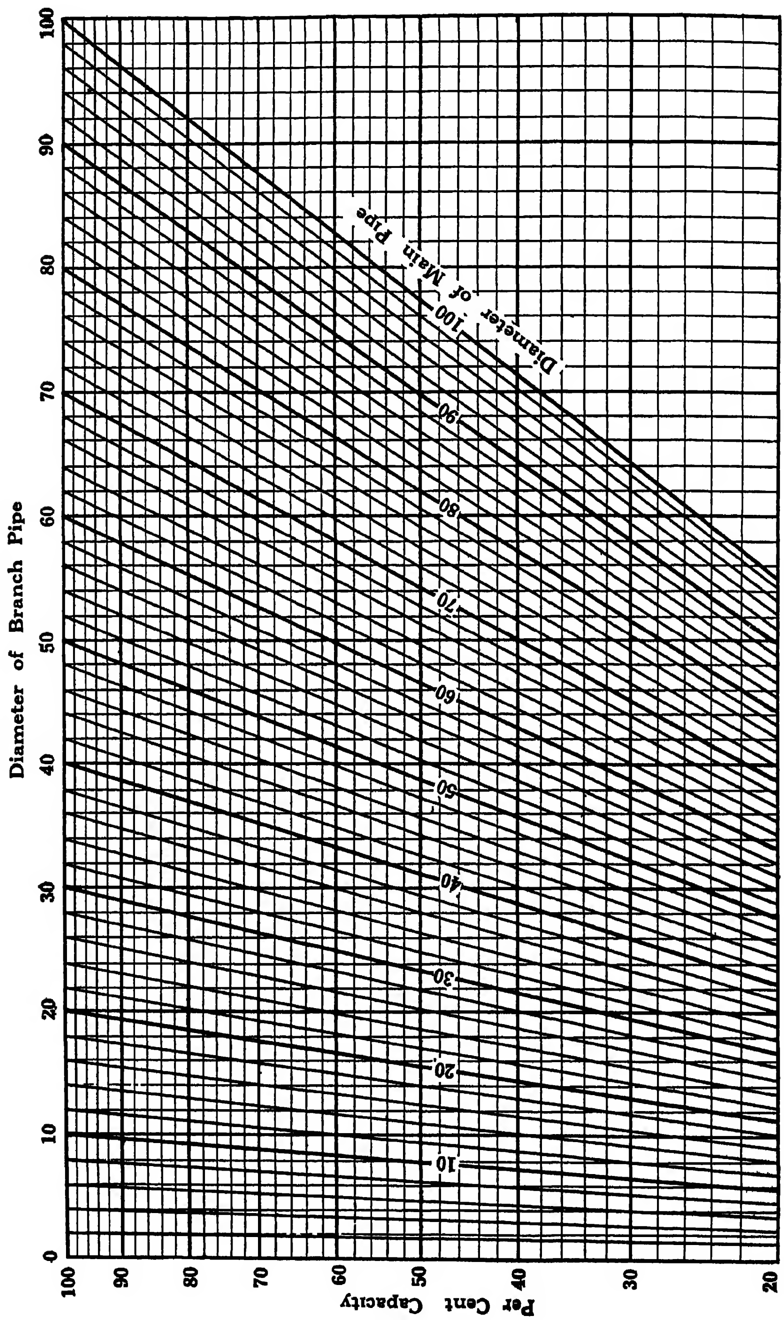


Fig. 5. Main and Branch Pipes for Equal Friction per Foot of Length (20 to 100 Per Cent Capacity)

maximum velocities with good construction and design may be as high as 2000 or 2200 fpm in main ducts, with suitable reduction in branches and outlets. With these velocities first-class duct construction is essential.

Proportioning the Size for Friction

By means of Figs. 4 and 5 the diameter of branch pipes necessary to carry a given percentage of the total air in the main pipe and to maintain equal friction per foot of the length through the entire system may be determined. These charts, as well as Fig. 3, are based on the assumption that the coefficient of friction varies inversely as the 1/7 power of the capacity.

Example 2. Suppose a 60-in. main pipe is to be used, and it is desired to know the size of branch pipe required to carry 50 per cent of the total air in the main Find 50 per cent at the left of the chart, move right to the 60-in. diagonal line and note directly above at the top of the chart that the branch pipe will be 46.5 in. in diameter.

Where rectangular ducts are used it is frequently desirable to know the equivalent diameter of round pipe to carry the same capacity and have the same friction per foot of length. Table 1 gives directly the circular equivalent of rectangular ducts for equal friction and capacity. To obtain the size of rectangular ducts for different capacities, but of the same friction per foot of length, first obtain the equivalent round pipe for equal friction. Thus, if a branch of sufficient size to carry 30 per cent of a 12 x 36-in. pipe is desired, it is found from Table 1 that the main is equivalent to a 22.2-in. diameter round pipe. From Fig. 5, 30 per cent of this is a pipe 14.3 in. in diameter, and referring again to Table 1, the rectangular equivalent branch is a 12 x 14-in., 10 x 17¼-in., or any other desirable combination.

Multiplying or dividing the length of each side of a pipe by a constant is the same as multiplying or dividing the equivalent round size by the same constant. Thus, if the circular equivalent of an 80 x 24-in. duct is required, it will be just twice that of a 40 x 12-in. duct, or $2 \times 23.3 = 46.6$ in.

MAIN TRUNK DUCTS

A main duct with branches is generally used to convey tempered air for ventilation purposes only. In place of individual ducts, a comparatively large main duct supplies air by branches to the room or rooms. The velocities vary according to the nature of the installation and the degree of quietness required. At the start of the run a velocity as high as 2000 fpm may be used, but this is considered the maximum for public building work, and is reduced to from 400 to 800 fpm in the risers. This duct system may be designed so that the loss of pressure in the branches is equalized in a manner similar to that previously described.

Equal Friction Method

Example 3. Fig. 6 shows a typical layout of an air distribution system which is applicable for ventilation of hotel dining rooms and offices.

The volume of air in cubic feet per minute for the room is determined on the basis of the number of air changes per hour required. In the example shown, the room ventilated is a hotel dining room 135 ft x 85 ft x 15 ft A 7½-minute air change (8 air changes per hour) is assumed for proper ventilation, giving 22,935 cfm as the air required.

The clear area of the fresh air inlet is based on a velocity of 1000 fpm or $\frac{22,935}{1000} =$

Table 1. Circular Equivalents of Rectangular Ducts for Equal Friction[a]

Side Rectangular Duct	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	24
8	6.1	6.9	7.6	8.2	8.8															
9	6.5	7.3	8.0	8.7	9.3	9.9														
10	6.8	7.7	8.4	9.2	9.8	10.4	11.0													
11	7.1	8.0	8.8	9.6	10.2	10.9	11.5	12.1												
12	7.4	8.3	9.2	10.0	10.7	11.4	12.0	12.6	13.2											
13	7.6	8.7	9.6	10.4	11.1	11.8	12.5	13.1	13.7	14.3										
14	7.9	8.9	9.9	10.8	11.5	12.3	12.9	13.6	14.3	14.9	15.4									
15	8.2	9.2	10.2	11.1	11.9	12.7	13.4	14.1	14.7	15.3	16.0	16.5								
16	8.4	9.5	10.5	11.4	12.3	13.1	13.8	14.5	15.2	15.8	16.5	17.1	17.6							
17	8.6	9.8	10.8	11.8	12.6	13.5	14.2	15.0	15.7	16.3	17.0	17.6	18.2	18.7						
18	8.9	10.0	11.1	12.1	13.0	13.8	14.6	15.4	16.1	16.8	17.4	18.1	18.7	19.2	19.8					
19	9.1	10.3	11.4	12.4	13.3	14.2	15.0	15.8	16.5	17.2	17.9	18.6	19.2	19.8	20.4	20.9				
20	9.3	10.5	11.6	12.7	13.6	14.5	15.4	16.2	17.0	17.6	18.4	19.0	19.7	20.3	20.9	21.5	22.0			
22	9.7	11.0	12.1	13.2	14.2	15.2	16.1	16.9	17.8	18.5	19.2	19.9	20.6	21.3	21.9	22.5	23.1	23.6	24.2	
24	10.0	11.4	12.6	13.8	14.8	15.8	16.8	17.6	18.5	19.3	20.0	20.8	21.5	22.2	22.8	23.5	24.0	24.7	25.2	26.4
26	10.4	11.8	13.1	14.3	15.4	16.4	17.3	18.3	19.2	20.0	20.8	21.6	22.3	23.0	23.8	24.4	25.1	25.7	26.3	27.5
28	10.8	12.2	13.5	14.8	15.9	17.0	18.0	19.0	19.8	20.7	21.5	22.4	23.1	23.9	24.6	25.3	26.0	26.6	27.3	28.5
30	11.0	12.6	13.9	15.2	16.4	17.5	18.5	19.5	20.5	21.4	22.2	23.1	23.9	24.7	25.4	26.2	26.8	27.5	28.2	29.5
32	11.3	12.9	14.3	15.6	16.9	18.0	19.1	20.1	21.1	22.0	22.9	23.8	24.6	25.4	26.2	27.0	27.7	28.4	29.1	30.5
34	11.6	13.2	14.7	16.1	17.3	18.5	19.6	20.7	21.6	22.6	23.5	24.4	25.3	26.2	26.9	27.7	28.5	29.2	30.0	31.3
36	11.9	13.6	15.1	16.4	17.7	19.0	20.1	21.2	22.2	23.2	24.2	25.1	26.0	26.8	27.7	28.5	29.3	30.0	30.8	32.2
38	12.2	13.9	15.4	16.8	18.2	19.4	20.6	21.7	22.8	23.8	24.8	25.8	26.7	27.5	28.4	29.2	30.0	30.8	31.5	33.1
40	12.5	14.3	15.7	17.2	18.6	19.8	21.1	22.2	23.3	24.4	25.4	26.4	27.3	28.2	29.1	29.9	30.8	31.6	32.4	33.9
42	12.7	14.5	16.1	17.6	19.0	20.3	21.6	22.7	23.8	24.9	25.9	26.9	27.9	28.8	29.8	30.7	31.4	32.2	33.0	34.5
44	13.0	14.8	16.4	18.0	19.4	20.7	22.0	23.1	24.3	25.4	26.5	27.5	28.5	29.5	30.3	31.2	32.1	32.9	33.7	35.3
46	13.3	15.1	16.7	18.4	19.8	21.1	22.4	23.6	24.8	25.9	27.0	28.1	29.1	30.1	31.0	31.9	32.8	33.8	34.6	36.2
48	13.5	15.4	17.0	18.7	20.1	21.5	22.8	24.1	25.2	26.4	27.5	28.6	29.6	30.5	31.6	32.5	33.4	34.3	35.2	37.0
50	13.7	15.7	17.3	19.0	20.4	21.9	23.2	24.5	25.7	26.9	28.0	29.2	30.3	31.3	32.2	33.1	34.1	35.0	35.9	37.6
52	13.9	15.9	17.6	19.2	20.8	22.2	23.6	24.9	26.2	27.4	28.5	29.6	30.7	31.8	32.9	33.8	34.7	35.6	36.5	38.3
54	14.1	16.1	17.9	19.6	21.1	22.6	24.0	25.3	26.6	27.8	29.0	30.1	31.2	32.3	33.4	34.4	35.3	36.3	37.2	38.9
56	14.3	16.3	18.2	19.9	21.5	22.9	24.4	25.7	27.0	28.3	29.5	30.6	31.7	32.8	33.9	34.9	35.9	36.9	37.8	39.6
58	14.6	16.6	18.4	20.2	21.8	23.3	24.7	26.1	27.4	28.7	30.0	31.1	32.2	33.3	34.4	35.4	36.4	37.4	38.4	40.3
60	14.7	16.8	18.7	20.4	22.1	23.6	25.1	26.5	27.8	29.1	30.5	31.6	32.7	33.8	34.9	36.1	37.1	38.1	39.1	40.9
62	15.0	17.0	19.0	20.7	22.4	24.0	25.5	26.9	28.2	29.5	30.9	32.1	33.2	34.3	35.4	36.6	37.7	38.7	39.6	41.6
64	15.1	17.3	19.2	21.0	22.7	24.3	25.9	27.3	28.6	29.9	31.3	32.6	33.7	34.8	35.9	37.1	38.2	39.2	40.2	42.2
66	15.3	17.5	19.5	21.2	23.0	24.6	26.2	27.7	29.0	30.3	31.7	33.0	34.2	35.3	36.4	37.6	38.7	39.8	40.8	42.8

[a]Additional sizes: 4 × 5 = 4.9; 4 × 6 = 5.4; 4 × 7 = 5.8; 5 × 5 = 5.5, 5 × 6 = 6.3, 5 × 7 = 6.5

TABLE 1. CIRCULAR EQUIVALENTS OF RECTANGULAR DUCTS FOR EQUAL FRICTION—(Continued)

Side Rectangular Duct	26	28	30	32	34	36	38	40	42	44	46	48
26	28.6											
28	29.7	30 8										
30	30.7	31.9	33.0									
32	31.7	32 9	34.1	35.2								
34	32.7	33 9	35 1	36 3	37.4							
36	33.7	34.9	36.1	37.3	38 5	39 6						
38	34.6	35.9	37 1	38.4	39 5	40.7	41 8					
40	35 3	36 7	38 0	39.3	40.5	41.7	42 9	44 0				
42	36.0	37.6	39 0	40 3	41.5	42.7	44.0	45.1	46.2			
44	36.9	38 5	39 9	41.2	42.5	43.7	44.9	46.1	47.2	48.4		
46	37.8	39.3	40.8	42 2	43 5	44 8	46 0	47.2	48 4	49 5	50 6	
48	38.5	40 0	41.5	43.0	44 4	45.6	46 9	48.1	49.3	50.5	51.6	52 8
50	39.2	40 8	42 3	43 8	45.2	46 5	47 9	49 1	50.4	51 6	52 9	54 0
52	40 0	41.6	43.1	44 7	46 1	47 5	48 9	50 1	51 3	52 5	53.8	55.0
54	40.7	42.4	44.0	45.5	47.0	48.4	49.9	51.1	52 3	53.5	54 8	56 0
56	41.3	43 0	44.6	46.2	47 7	49.1	50.6	52.0	53.3	54 6	55 9	57.0
58	42 1	43.8	45.4	47.0	48.5	50.0	51 5	52 9	54 2	55.5	56 8	58 0
60	42.7	44 5	46.1	47.8	49.3	50.9	52 3	53.8	55.0	56.4	57.7	58 9
62	43.4	45.1	46.8	48 4	50 0	51.7	53.0	54.5	55.9	57.2	58.5	59.7
64	44.0	45.8	47.5	49.2	50 9	52 4	53 9	55.4	56.8	58.1	59.4	60.6
66	44.7	46.5	48.2	50.0	51.6	53.1	54.7	56 2	57.6	59 1	60.4	61.6
68	45.3	47.2	48.9	50 7	52.2	53.8	55.5	56 9	58 4	59.9	61 3	62 6
70	46.0	47.8	49.5	51.3	52.9	54.5	56.2	57.7	59.1	60 6	62.1	63.5
72	46 5	48.4	50.1	51 9	53 7	55.4	57.0	58 7	60.0	61.3	63.0	64 5

Side Rectangular Duct	50	54	60	66	72	78	84	88
50	55 0							
52	56.1							
54	57 2	59.4						
56	58 3	60 5						
58	59 3	61.6						
60	60 3	62 7	66 0					
62	61 3	63 7	67.1					
64	62 2	64 7	68.2					
66	63 2	65.7	69.3	72 6				
68	64 1	66 6	70 3	73 7				
70	65 0	67 6	71 3	74 8				
72	65 9	68 5	72 3	75 9	79 2			
74	66 8	69 4	73.3	76 9	80 3			
76	67 6	70 3	74 2	77.9	81 4			
78	68 4	71 2	75 2	78 9	82 5	85 8		
80	69 2	72 1	76.1	79 9	83 6	86.9		
82	70 1	73 0	77 1	80 9	84.6	88.0		
84	70 9	73 8	78.0	81 9	85 6	89.1	92.4	
86	71 7	74 6	78 9	82 9	86 6	90.2	93.5	
88	72 5	75 5	79 8	83 9	87.5	91.2	94.6	96.8
90	73 3	76.3	80.6	84.7	88 5	92.2	95.7	97.9
92	74.1	77 1	81 4	85.6	89 5	93.2	96.7	99.0
94	74.8	77 8	82.2	86 5	90.4	94.2	97.8	100 1
96	75.5	78 7	83.0	87.4	91.3	95.2	98 8	101.2

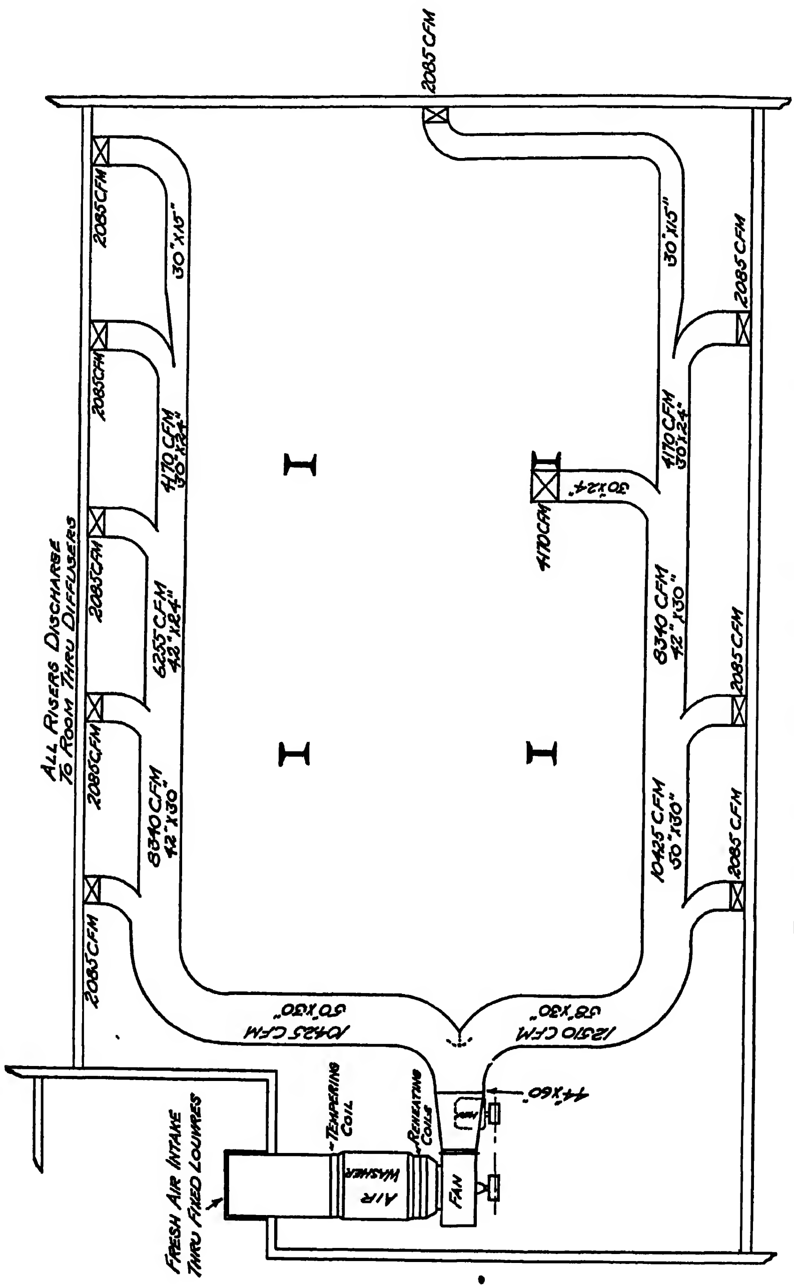


FIG. 6. TYPICAL LAYOUT OF AIR DISTRIBUTION SYSTEM

22.94 sq ft If the air washer is provided with automatic humidity control, the tempering coil should raise the temperature of the entering air to 32 F The washer with its automatic control will then raise the temperature from 32 F to 42 F If the washer is not provided with automatic humidity control, the tempering coil must raise the temperature of the entering air to at least 55 F to allow for some temperature drop in the washer due to evaporation The reheating coil is selected to raise the temperature of the air from that leaving the air washer to 70 F The air washer should have a maximum velocity of 500 fpm through the clear area, which, in this case, is 46 sq ft For more detailed information on tempering coil and air washer control, see Chapters 14 and 24.

Since the plan shows a moderately short run of main duct with no risers near the fan outlet, a fan should be selected which will have the required capacity of 22,935 cfm with a maximum velocity through the fan outlet of 1400 fpm The outlet area, therefore, should be 16½ sq ft

TABLE 2. PIPE SIZES FOR EXAMPLE 3[a]

VOLUME OF AIR (CFM)	PER CENT OF TOTAL VOLUME	DIAMETER OF PIPE (INCHES)	EQUIVALENT SIZE OF RECTANGULAR DUCT (INCHES)
22,935	100 0	56	60 x 44
12,510	54 6	45	58 x 30
10,425	45 4	42	50 x 30
8,340	36 3	39	42 x 30
6,255	27 2	35	42 x 24
4,170	18 2	29½	30 x 24
2,085	9.1	23	30 x 15

[a]Velocity through diffusers (not shown) to be approximately 300 fpm

The main pipe size should be selected to give a velocity equal to or less than the velocity at the fan outlet. Choosing a 56-in. pipe with a cross-sectional area of 17.1 sq ft, the velocity in the main pipe will be 1340 fpm. Using the friction pressure loss method this 56-in main pipe will be taken as the basis of calculation.

Fig. 6 shows the amount of air to be handled by each section of pipe. Expressing the volume handled by each section as a percentage of the total volume and using the charts, Figs. 4 and 5, the pipe sizes are as shown in Table 2.

The pressure at the outlets nearest the fan will be greater than at the pipes farther along the run so that the former will tend to deliver more than the calculated amount of air. To remedy this condition, volume regulating dampers should be located at the base of each riser and adjusted for proper distribution. At points where branches leave the main it may be advisable, depending upon the nature of the installation, to install adjustable splitters similar to that shown in Fig. 6 where the main duct divides into the 58 in. × 30 in. and 50 in. × 30 in. branches

The rectangular equivalents are selected from Table 1, the width to depth proportion will be determined by construction requirements and ease of fabrication The calculation of the friction is as follows

The longest run from the fan outlet to diffuser is 150 ft 0 in ; 150 ft of 56-in pipe is equivalent to $\frac{150 \times 12}{56}$ .... 32.2 dia.

Two 45-in., 90-deg elbows $(2 \times \frac{45}{56} \times 8\ 5)$ .... 13.7 dia.

(Assume each elbow equivalent to 8 5 diameters of duct, Fig 1)

Two 23-in , 90-deg elbows $(2 \times \frac{23}{56} \times 8\ 5)$ .... 7.0 dia.

Two 23-in , 90-deg elbows in riser $(2 \times \frac{23}{56} \times 30)$ .... 24 7 dia

(Two bad elbows in riser, each equivalent to 30 diameters of duct)

Total diameter of 56-in. pipe 77 6

The velocity head corresponding to a velocity of 1340 fpm is $\left(\frac{1340}{4005}\right)^2 = 0.112$ in.

Taking 50 diameters as one head loss, then $\frac{77.6}{50} \times 0\ 112 = 0\ 174$ in. static loss in duct.

Where the connection pieces are made with long easy slopes and the general workmanship is good, a regain in static pressure may be deducted from the foregoing pressure loss This can be taken as approximately two-thirds the difference in velocity pressures at the fan outlet and the last run of pipe. The velocity in the riser is 667 fpm with a corresponding velocity pressure of 0 027 in. The fan outlet velocity is 1400 fpm with a corresponding velocity pressure of 0.122 in The regain equals ⅔ (0 122 − 0 027) = 0.063 in.

The net static pressure loss in the duct is:

0 174 in. − 0.063 in. .... 0 111 in.

Other friction losses are as follows:

(1) Fresh air intake 1000-fpm velocity (1½ heads × 0 0625) .... 0 094 in.
(2) Tempering coil loss (from manufacturer's tables) .... 0 100 in.
(3) Air washer loss (from manufacturer's tables) .... 0.250 in.
(4) Reheating coil loss (from manufacturer's tables) .... 0.100 in.
(5) Allowance for regulating dampers and diffusers .... 0.100 in.

Static pressure loss of system .... 0 755 in.

The fan should be selected from the manufacturer's ratings which, according to the Standard Test Code for Disc and Propeller Fans, Centrifugal Fans and Blowers[1], will deliver 22,935 cfm at a static pressure of 0 755 in. and which has an outlet area of 16½ sq ft.

The method of design used in Example 3 is the *equal friction method* described under the heading Procedure for Duct Design. This involves the arbitrary reduction of velocity from the fan outlet to the point of discharge to the room, and the friction is calculated by adding the pressure losses of each section of duct. This method requires dampering in the risers.

Example 4. Fig. 7 shows an exhaust system layout for exhausting from buildings of the same type as in Example 3. Assume the air requirements based on the number of air changes per hour to be 16,800 cfm. Using a velocity of 1400 fpm in the main duct at

TABLE 3. PIPE SIZES FOR EXAMPLE 4[a]

Volume of Air (cfm)	Per Cent of Total Volume	Diameter of Pipe (Inches)	Equivalent Size of Rectangular Duct (Inches)
16,800	100.0	47	38 x 48
11,550	68.8	41	30 x 46
9,450	56.2	38	30 x 40
5,250	31.3	31	24 x 34
4,200	25.0	28.5	24 x 28
3,150	18.8	25.3	16 x 34
2,100	12.5	21 6	16 x 24

[a]Velocity through intake grilles (not shown) to be approximately 400 fpm

[1]See Chapters 17 and 44

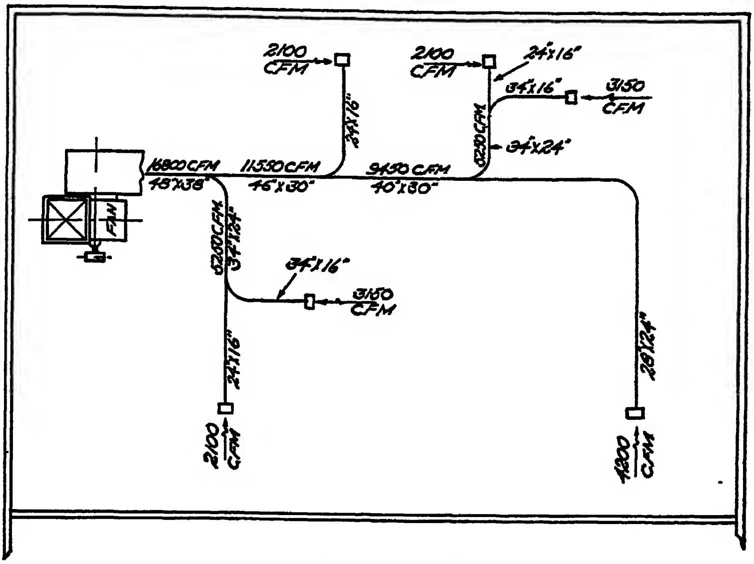


FIG. 7. EXHAUST SYSTEM LAYOUT

the fan inlet, which is an average velocity for this type of system, the area of the main is 12 sq ft, which corresponds to a 47-in. pipe. Referring to Example 3, and using the charts, Figs. 4 and 5, the pipe sizes are as indicated in Table 3.

All risers will require dampering as in Example 3. The calculation of the friction is as follows:

The longest run from the intake grille to fan inlet is 100 ft

(1) Duct friction 100 ft of 47-in. pipe $\left(\frac{100 \times 12}{47}\right)$ .... 25.6 dia.

Two 28½-in., 90-deg elbows in riser $\left(\frac{2 \times 28.5 \times 30}{47}\right)$ .... 36.4 dia.

(Two bad elbows in riser each equivalent to 30 diameters of duct).

One 28½-in., 90-deg elbow in horizontal run $\left(\frac{28.5 \times 8.5}{47}\right)$ .... 5 2 dia.

Total diameter of 47-in. pipe .... 67.2 dia.

Velocity head corresponding to 1400 fpm is $\left(\frac{1400}{4005}\right)^2 = 0.122$ in

Taking 50 diameters as one head loss, then $\frac{67.2 \times 0.122}{50}$ .... 0.164 in.

(2) Intake loss from grille (1½ heads at a 400 fpm velocity 1½ × 0.01) .... 0.015 in.
(3) Static pressure required to produce one velocity head at 1400 fpm .... 0.122 in.
(4) Loss occasioned by step-up of velocity (0.20 × 0.122) .... 0.024 in.

(This loss varies from 0 05 to 0 40 velocity head depending upon the nature of the change. For average systems 0 20 velocity head is a close approximation.)

Static pressure loss on inlet side .... 0.325 in.

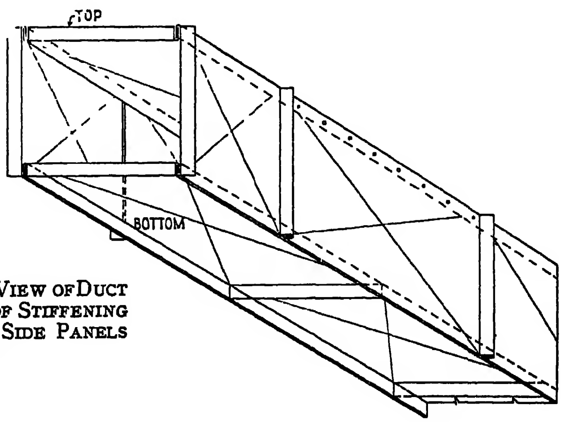


Fig 8 Isometric View of Duct Showing Location of Stiffening Seams on Top and Side Panels of Duct

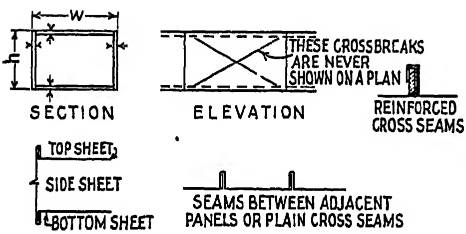


Fig. 9. Details of Seams

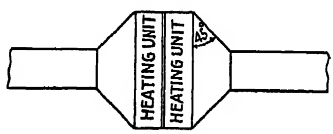


Fig. 10. Method of Installing Heating Unit

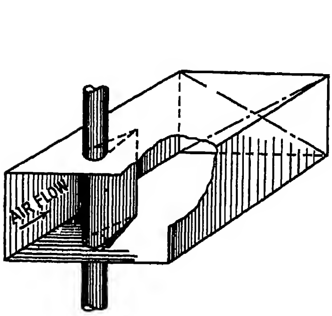


Fig. 11. Installation of Easement in Duct Around Obstruction

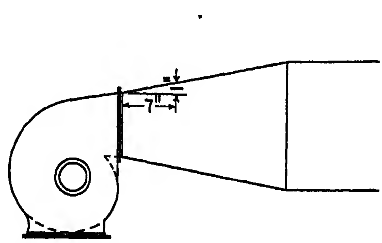


Fig. 12. Fan Discharge Connection

To this must be added the resistance on the discharge side of the fan A fan outlet velocity of approximately 1500 to 1600 fpm may be used Assuming the fan outlet to be equivalent in area to a 45-in pipe, the velocity is 1525 fpm

Loss on discharge (15 ft from fan outlet to discharge)

$$\frac{15 \times 12}{45} = 4 \text{ diameters of 45-in pipe.}$$

The velocity head corresponding to a velocity of 1525 fpm is 0.145 and the discharge-side loss is $\frac{0\,145 \times 4}{50} = 0\,012$ in The total static pressure loss of the system is then:

$$0\,012 + 0\,325 = 0\,337 \text{ in}$$

The fan will be selected to handle 16,800 cfm at a static pressure of 0 337 in and to have an outlet velocity of 1525 fpm Outlet area 11 sq ft.

Where there are one or more ducts with branches, the velocity of air in the ducts may be either chosen arbitrarily or calculated for friction losses. When arbitrary values are assigned, a certain amount of dampering should be provided for; this will be small when the method chosen permits a drop in velocity as the quantity of air is reduced.

After the total air quantity and the size of fan are ascertained, the main duct is usually fixed as being at least equal in area to the fan outlet, or perhaps 10 per cent greater. From this main pipe all others are proportioned. For example, if the main duct is 30 in. in diameter, a branch to carry 10 per cent of the total capacity should be 12.7 in. in diameter (see Fig. 4) in order to have the same friction per foot of length, while one carrying one-half the total capacity of a 30-in. main with the same friction loss per foot would be 23.4 in. in diameter. By this method of equalizing friction it is unnecessary to consider the resistance of each section of pipe independently, but only to know the distance from the fan outlet to the end of the longest run of pipe, the number and size of elbows, and the diameter and velocity in the largest pipe.

Example 5. If the greatest length of piping in a system is 130 ft with a 26-in diameter main pipe and one 20-in elbow, the piping having been designed for equal friction per foot of length, the friction would be the same as for 130 linear feet of 26-in. pipe, or 60 diameters. To this should be added the friction loss in elbows, in this case one 20-in elbow, which has a loss equivalent to 8.5 diameters of 20-in pipe. This in turn is $\frac{20}{26} \times 8.5 = 6.6$ diameters of 26-in. pipe. The total equivalent length of the system will then be $60 + 6.6$, or 66 6 diameters Since 50 diameters is equivalent to one velocity head, the loss is $\frac{66\,6}{50} = 1.33$ times the velocity head. If the velocity is, for example, 2200 fpm, corresponding to 0.3-in. pressure, the friction loss of the system will be $1.33 \times 0.3 = 0.399$ in.

Frequently the prevention of sound in a heating or ventilating system imposes more severe restrictions than the prevention of excessive pressure drop. This question is highly involved and requires consideration of many factors. The air velocities to be used will vary with the standard of construction used in the ducts themselves as well as with the nature of the occupancy and the construction of the building. In general, architects and engineers who leave the details of duct construction to the contractor must, of necessity, design for lower velocities than might be required for quiet operation if proper construction details were always followed. The

contractor may be expected to build the ducts by the least expensive methods, and the engineer must anticipate this. For further information on noise reduction, see Chapter 18.

Duct Construction Details

If panel construction is used with standing seams or similar reinforcement, and the panels are cross-broken to give rigidity, there is less likelihood of vibration due to air flow, or deflection due to air pressure. Elbows made without splitters, and improperly shaped transformation sections produce high local velocities which are the cause of noise in duct work. The use of first-class duct construction with well-designed transformation sections and splitters in elbows tends to maintain relatively uniform velocities with decrease in turbulence and in the noise produced.

TABLE 4. SHEET METAL GAGES FOR RECTANGULAR DUCT CONSTRUCTION[a]

GAGE	WIDTH OF DUCT	SEAM	REINFORCED SEAM
26	Up to 12 in.		
24	13 in. to 30 in	1	
22	31 in. to 48 in.	1	
22	49 in to 60 in	1½	⅛ in x 1⅜ in
20	61 in. to 90 in	1½	⅛ in x 1⅜ in

[a]If panels are not cross-broken two gages heavier material should be used

Figs. 8 to 12 show acceptable construction details for rectangular ducts, elbows, and transformation pieces or connections. Other methods are also acceptable, such as the use of angle iron stiffeners for large ducts. Good construction is essential to the elimination of duct noises and for the prevention of a flimsy installation.

Fig. 8 is an isometric view of a duct showing the location of the stiffening seams on the top and side panels. The cross seams should not occur at the same place but should be staggered as indicated. Heating units should be installed as shown in Fig. 10 with the duct connections making an angle of not less than 45 deg, but preferably 60 deg. Fan discharge connections should have a maximum slope of 1 in 7, as indicated in Fig. 12. Whenever a pipe or other obstruction passes through a duct an easement should be placed around the pipe as indicated in Fig. 11. The recommended gages for rectangular sheet metal duct construction are given in Table 4.

REFERENCES

Fan Engineering, Buffalo Forge Co.

Heat Power Engineering, by Barnard, Ellenwood, and Hirshfeld, Part III

Mechanical Engineers' Handbook, by Lionel S. Marks, McGraw-Hill Book Co

The Flow of Liquids, by W H McAdams, *Refrigerating Engineering*, February, 1925, p 279.

A Study of the Data on the Flow of Fluids in Pipes, by Emory Kemler, *A S M E. Transactions*, Hydraulics Section, August 31, 1933, p 7

PROBLEMS IN PRACTICE

1 ● Determine the equivalent number of diameters of straight pipe equivalent to a 90 deg elbow having center line radii of (a) 100 per cent, (b) 150 per cent, abd (c) 200 per cent of the pipe diameter.

Assume 1 velocity head lost in 50 diameters.

From Fig. 1 the per cent of velocity head lost

a For 100 per cent radius is 25 5 per cent × 50 = 12 8 diameters straight pipe
b For 150 per cent radius is 17 0 per cent × 50 = 8.5 diameters straight pipe
c For 200 per cent radius is 14.5 per cent × 50 = 7 3 diameters straight pipe

2 ● Why is it desirable to make elbows with a radius equal to one and one-half times the pipe diameter?

Reference to Figs 1 and 2 will show that while the loss of velocity head, as indicated by the curves, shows considerable variation for elbows between the range of 50 and 150 per cent radius, the line is practically straight after 150 per cent, indicating very little variation in loss of head for elbows of larger radius

3 ● What is the best shape to use for ducts?

The shapes to be used in designing ducts, in the order of their preference, are round, square, and rectangular

4 ● What determines which shape to use?

Structural and space conditions Because ducts are as a rule part of the building or structure, it is necessary to proportion their sizes to fit the spaces available.

5 ● What is meant by "arbitrarily fix the velocity in the various sections?"

When using the velocity method as a basis for design, the maximum allowable velocity is fixed for the main supply duct at the fan, and this velocity is gradually decreased as each branch or outlet is taken off the main supply duct.

6 ● Which system of duct design is to be preferred, the velocity method or the friction pressure loss method?

The friction pressure loss method can be used to advantage where no structural or building conditions limit the shape of the ducts. Where these limiting conditions exist the velocity method is to be preferred.

7 ● Are the grille sizes figured on the same basis as the outlets?

The free area through the grilles is figured the same as the outlets, and this area is increased from 20 to 50 per cent, depending on the design of the grille, to allow for the loss of area caused by the construction of the face of the grille.

8 ● Where it is necessary to provide steel angle braces, how far apart should they be spaced?

Angle braces for large ducts should be placed on 3-ft 0-in. centers

9 ● How much air will a 10-in. by 24-in. duct handle if it is part of a system designed on a pressure drop of 0.1 in. per 100 feet of run?

1450 cfm (Table 1 and Fig. 3).

10 ● How does a splitter at a duct junction influence the volume of the air going through each branch?

A splitter facing the direction of air flow cuts off the air and delivers the desired amount to the branch

11 ● Why does a wide, shallow duct offer more resistance to the flow of air than does a square duct of equal cross-sectional area?

The perimeter of the wide, flat duct is greater than that of the square-section duct, so the former has the greater frictional area which increases the resistance and thus reduces the volume at any given pressure

12 ● What methods are used to keep large ducts from vibrating because of air pulsations, and from sagging because of their own weight?

External bracing, such as standing seams, or structural shapes, like tees or angles, should be placed across the top and bottom Exterior braces or cross buckling of metal sheets in diagonal panels may be used for the sides of large ducts

13 ● What velocities of air flow should be used in the trunk ducts of a ventilating system in a public building?

From 1200 to 1600 fpm

14 ● In a ventilating system in a residence, what is the recommended air velocity through supply registers and grilles?

400 fpm.

Chapter 21

INDUSTRIAL EXHAUST SYSTEMS

System Classification, Design Procedure, Requirements for Suction and Velocity, Hoods, Design of Duct Systems, Collectors, Resistance of Systems, Efficiency of Exhaust Systems, Selection of Fans and Motors, Corrosion

SOME type of exhaust and collecting system is necessary in almost every industry and the present chapter attempts to give general information relating to the design of factory exhaust systems in order that efficient and economical control of dusts and fumes may be achieved.

SYSTEM CLASSIFICATIONS

There are two general arrangements, the central and the group systems. In the central system a single or double fan is located near the center of the shop with a piping system radiating to the various machines to be served. In the group system, which is sometimes employed where the machines to be served are widely scattered, small individual exhaust fans are located at the center of the machine groups. The group arrangement has the advantage of flexibility.

Exhaust systems are also classified by the means employed to collect dust or other material handled. The dust or refuse may be collected and controlled by enclosing hoods, open hoods, inward air leakage, or by exhausting the general air of the room.

With some classes of machinery it is not feasible to closely hood the machines and in these cases open hoods over or adjacent to the machines are provided to collect as much as possible of the dust and fumes. This class includes such machines as rubber mills, package filling machinery, sand blast, crushers, forges, pickling tanks, melting furnaces, and the unloading points of various types of conveyors.

The open hoods should be placed as close to the source of dust or fumes as possible, with due regard to the movements of the operator. When the hood must be placed at some distance above the machine it should be large enough to encompass an area of considerable extent as diffusion is usually quite rapid.

Consideration must also be given to the natural movement of the fumes. For those that are lighter than air the hood should be over or above the machine and where a heavy vapor or dust-laden air at ordinary temperature is to be removed, horizontal or floor connections are required. If it is attempted to remove heavy dust such as lead oxides by an overhead hood the conditions may be worse than if no exhaust were used at all, owing to the rising air current carrying the dust up through the

breathing zones. The objective to keep in mind in all cases is to take advantage of the natural tendency of the material to move upward or downward.

In another class of operation the main objective is to prevent the escape of dust into the surrounding atmosphere, the removal of some dust from the machine or enclosure being merely incidental. The dust-creating apparatus is enclosed within a housing which is made as tight as practicable, and sufficient suction is applied to the enclosure to maintain an inward air leakage, thus preventing escape of the dust. While the exhaust system is required to handle only the air which leaks in through the crevices and openings in the enclosure, yet in many installations leakages are very high and great care is required to obtain satisfactory results with a system of this kind. The inward-leakage principle is utilized for controlling dust in the operating of tumbling barrels, grinding, screening, elevating, and similar processes.

Certain dust and fume producing operations are best carried on by isolating the process in a separate compartment or room and then applying general ventilation to this space. The compartment or room in which the work is performed should be as small as is consistent with convenience in handling the work. The ventilating system should be designed so that a strong current of clean air is drawn across the operator, and away from him toward the work, where the dust is picked up and carried from the room.

DESIGN PROCEDURE

The first step in the design of an exhaust system is to determine the number and size of the hoods and their connections. No general rules, however, can be given since hood and duct dimensions are determined by the characteristics of the operations to which they are applied. When a tentative decision regarding the set-up has been made, it is then necessary to obtain the suction and air velocities required to effect control. At this point the designer must rely upon the prevailing practice and on such physical data relating to hoods, duct systems and collectors as are available. Finally, in choosing the fan, the area of the intake should be equal to or greater than the sum of the areas of the branch ducts. The speed, of course, must be sufficient to maintain the estimated suction and air velocities in the system. In general, the most important requirements of an efficient exhaust and collecting system are as follows[1]:

1. Hoods, ducts, fans and collectors should be of adequate size.
2. The air velocities should be sufficient to control and convey the materials collected.
3. The hoods and ducts should not interfere with the operation of a machine or any working part.
4. The system should do the required work with a minimum power consumption.
5. When inflammable dusts and fumes are conveyed, the piping should be provided with an automatic damper in passing through a fire-wall.
6. Ducts and all metal parts should be grounded to reduce the danger of dust explosions by static electricity.
7. The design of an exhaust system should afford easy access to parts for inspection and care.

---

[1]For more detailed requirements see Safe Practice Pamphlets Nos 32 and 37, published by the *National Safety Council*, Chicago

REQUIREMENTS FOR SUCTION AND VELOCITY

The removal of dust or waste by means of an exhaust hood requires a movement of air at the point of origin sufficient to carry it to a collecting system. The air velocities necessary to accomplish this depend upon the physical properties of the material to be eliminated and the

Table 1. Size of Connections for Wood-Working Machinery

Type of Machine	Diameter of Connections in Inches
Circular saws, 12-in. diam.	4
Circular saws, 12-24-in. diam.	5
Circular saws, 24-40-in. diam.	6
Band saws, blade under 2 in. wide	4
Band saws, blade 2-3 in. wide	5
Band saws, blade 3-4 in. wide	6
Band saws, blade 4-5 in. wide	7
Band saws, blade 5-6 in. wide	8
Small mortisers	6
Single end tenoners	6
Double end tenoners	7
Double end, double head tenoners	10
Planers, matchers, moulders, stickers, jointers, etc.—	
With knives, 6-10 in.	5–6
With knives, 10-20 in.	6–8
With knives, 20-30 in.	6–10
Shapers, light work	4–5
Shapers, heavy work	8
Belt sander, belt less than 6 in wide	5
Belt sander, belt 6-10 in. wide	6
Belt sander, belt 10-14 in. wide	7
Drum sander, 24 in	5
Drum sander, 30 in.	6
Drum sander, 36 in.	7
Drum sander, 48 in.	8
Drum sander, over 48 in.	10
Disc sander, 24 in. diam.	5
Disc sander, 26-36 in. diam.	6
Disc sander, 36-48 in diam	7
Arm sander	4

direction and speed with which it is thrown off. If the dust to be removed is already in motion, as is the case with high-speed grinding wheels, the hood should be installed in the path of the particles so that a minimum air volume may be used effectively. It is always desirable to design and locate a hood so that the volume of air necessary to produce results is as small as possible.

The static suction at the throat of a hood is frequently used in practice as a measure of the effectiveness of control. This is of considerable value where exhaust systems adapted to particular operations have been standardized by practice. Tables 1 and 2 present the duct sizes usually employed for standard wood-working machinery and for grinding and buffing wheels. Static pressures which in practice have been found necessary to control and convey various materials, are given in Table 3. It must be remembered, however, that the *suction* is merely a rough

Table 2. Size of Connections for Grinding and Buffing Wheels

Diameter of Wheels	Max Grinding Surface Sq In	Min Diam. of Branch Pipes in Inches
Grinding—		
6 in or less, not over 1 in thick	19	3
7 in. to 9 in., inclusive, not over 1½ in thick	43	3½
10 in. to 16 in., " " " 2 in "	101	4
17 in. to 19 in., " " " 3 in "	180	4½
20 in. to 24 in., " " " 4 in. "	302	5
25 in. to 30 in , " " " 5 in. "	472	6
Buffing—		
6 in or less, not over 1 in thick	19	3½
7 in. to 12 in., inclusive, not over 1½ in thick	57	4
13 in. to 16 in., " " " 2 in. "	101	4½
17 in. to 20 in., " " " 3 in "	189	5
21 in to 27 in., " " " 4 in "	338	6
27 in. to 33 in., " " " 5 in. "	518	7

measure of the air volume handled and consequently of the air velocity at the opening of the hood. The elimination of any dusty condition requires added information concerning the shape, size and location of the hood used with regard to the operation in question.

In some states grinding, polishing and buffing wheels are subject to regulation by codes. The static suction requirements, which range from 1½ to 5 in. water displacement in a *U*-tube, should be followed although in several instances they may appear to be excessive. Frequently, in these operations, a large part of the wheel must be exposed and the dust-laden air within the hood is thrown outward by the centrifugal action of the wheel, thus counteracting useful inward draft. This tendency may be diminished by locating the connecting duct so as to create an air flow of not less than 200 fpm about the lower rim of the wheel.

Exact determinations of hood control velocities are not available, but it is safe to assume that for most dusty operations they should not be less

Table 3. Suction Pressures Required at Hoods

Type of Installation	Static Suction in Inches of Water
Exhausting from grinding and buffing wheels	1½–5
Exhausting from tumbling barrels	2
Exhausting from wood-working machinery—light duty	2
Exhausting from wood-working machinery—heavy duty	2–4
Shoe machinery exhaust	2–3
Exhausting from rubber manufacturing processes	2
Flint grinding exhaust	2
Exhausting from pottery processes	2
Lead dust and fume exhaust	2–4
Fur and felt machinery exhaust	2–3
Exhausting from textile machinery	2–3
Exhausting from elevating and crushing machinery	2
Conveying bulky and heavy materials	3–5

than 200 fpm at the point of origin. For granite dust generated by pneumatic devices, Hatch et al[2] give velocities from 150 to 200 fpm, depending on the type of hood used, as sufficient for safe control. Considering the character of the industry, air velocities of this order may be extended to similar dusty operations. The method for approximately determining these velocities in terms of the velocity at the hood opening is given below.

HOODS

No set rule can be given regarding the shape of a hood for a particular operation, but it is well to remember that its essential function is to create an adequate velocity distribution. The fact that the zone of greatest effectiveness does not extend laterally from the edges of the opening may frequently be utilized in estimating the size of hood required. Where complete enclosure of a dusty operation is contemplated, it is desirable to leave enough free space to equal the area of the connecting duct. Hoods for grinding, polishing and buffing should fit closely, but at the same time should provide an easy means for changing the wheels. It is advisable to design these hoods with a removable hopper at the base to capture the heavy dusts and articles dropped by the operator. Such provisions are of assistance in keeping the ducts clear. Air volumes used to control many dust discharges may often be reduced by effective baffling or partial enclosure of an operation. This procedure is strongly urged where dusts are directed beyond the zone of influence of the hood.

Axial Velocity Formula for Hoods

When the normal flow of air into a hood is unobstructed, the following formula may be used to determine the air velocity at any point along the axis[3]:

$$V = \frac{0.1\,Q}{x^2 + 0.1\,A} \qquad (1)$$

where

V = velocity at point, feet per minute.
A = area of opening, square feet.
x = distance along axis, feet.
Q = volume of air handled, cubic feet per minute

Velocity Contours

It is possible by use of a specially constructed pitot-tube[4] to map contours of equal velocity in any axial plane located in the field of influence. It has been found that the positions of these contours for any hood can be expressed as percentages of the velocity at the hood opening and are purely functions of the shape of the hood[5].

[2]Control of the Silicosis Hazard in the Hard Rock Industries I A Laboratory Study of the Design of Dust Control Systems for Use with Pneumatic Granite Cutting Tools, by Theodore Hatch, Philip Drinker and Sarah P Choate (*Journal of Industrial Hygiene*, Vol XII, No. 3, March, 1930).

[3]The Control of Industrial Dust, by J M DallaValle (*Mechanical Engineering*, Vol 55, No 10, October 1933)

[4]Studies in the Design of Local Exhaust Hoods, by J M DallaValle and Theodore Hatch (TRANSACTIONS A S M E, Vol 54, 1932)

[5]Velocity Characteristics of Hoods under Suction, by J M DallaValle (A S.H V E TRANSACTIONS, Vol 38, 1932)

Further, the velocity contours are identical for similar hood shapes when the hoods are reduced to the same basis of comparison. These facts are applicable to all hood problems so that when the velocity contour distribution is known, the air flow required can be determined. Fig. 1 shows the contour distribution in two axial planes perpendicular to the sides of a rectangular hood with a side ratio of one-half. The distribution shown is identical for all openings with a similar side ratio provided the mapping is as shown in the figure. The contours, of course, are expressed as percentages of the velocity at the opening.

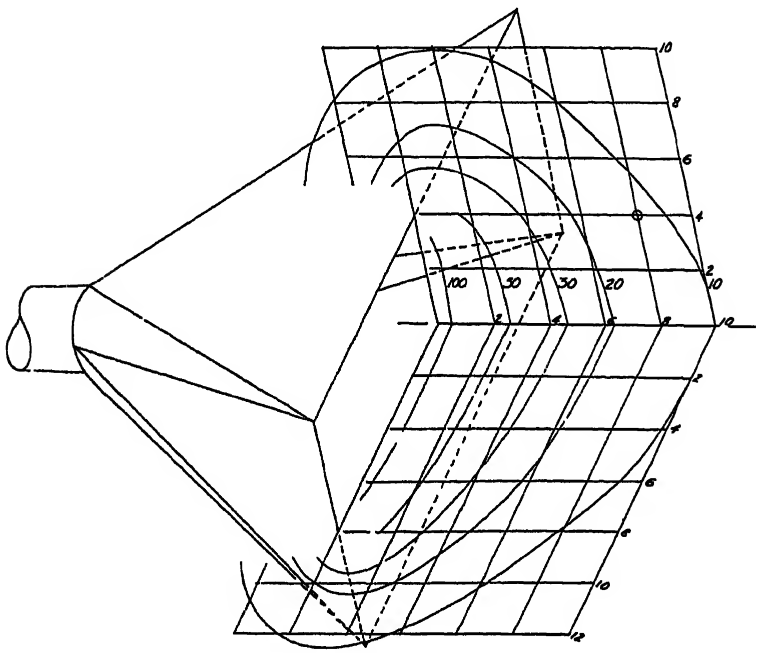


FIG. 1. VELOCITY CONTOURS FOR A RECTANGULAR OPENING WITH A SIDE RATIO OF ONE-HALF. CONTOURS ARE EXPRESSED AS PERCENTAGES OF THE VELOCITY AT THE OPENING

Air Flow from Static Readings

The volume of air flow through any hood may be determined from the following equation:

$$Q = 4005 f A \sqrt{h_t} \tag{2}$$

where

Q = volume of air flow, cubic feet per minute

A = area of connecting duct, square feet.

h_t = static suction at throat of hood, inches of water.

f = orifice or restriction coefficient which varies from 0.6 to 0.9 depending on the shape of the hood.

An average value of f is 0.71, although for a well-shaped opening a value of 0.8 may be used. The factor f is determined from the equation:

$$f = \sqrt{\frac{h_v}{h_t}} \qquad (3)$$

where h_v is the velocity head in the connecting duct.

The term *static suction* is not a good measure of the effectiveness of a hood unless the area of the opening and the location of the operation with respect to the hood are known. This is clearly indicated by Equation 1 which shows that the velocity at any point along the axis varies inversely as the area of the opening and the square of the distance. However, this formula coupled with Equation 2 should serve to indicate the velocity conditions to be expected when operations are conducted external to the hood opening.

Large Open Hoods

Large hoods, such as are used for electroplating and pickling tanks, should be subdivided so the area of the connecting duct is not less than one-fifteenth of the open area of the hood. Frequently, it will be found necessary to branch the main duct in order to obtain a uniform distribution of flow. *Canopy hoods* should extend 6 in. laterally from the tank for every 12-in. elevation, and wherever possible they should have side and rear aprons so as to prevent short circuiting of air from spaces not directly over the vats or tanks. In most cases, hoods of this type take advantage of the natural tendency of the vapors to rise, and air velocities may be kept low. Cross drafts from open doors or windows disturb the rise of the vapors and therefore provision must be made for them. The air velocities required also depend upon the character of the vapors given off, cyanide fumes, for example, requiring an air velocity of approximately 75 fpm on the surface of the tank and acid and steam vapors requiring velocities as low as 25 to 50 fpm. The total volume of air flow necessary to obtain these velocities may be approximately determined from the following simple formula:

$$Q = 1.4PDV \qquad (4)$$

where

Q = total volume of air handled by hood, cubic feet per minute.
P = perimeter of the tank, feet
D = distance between tank and hood opening, feet.
V = air velocity desired along edges and surface of tank, feet per minute.

Lateral Exhaust Systems

The lateral exhaust method, as developed for chromium plating[6], is applicable in many instances in preference to the canopy type hoods. The method makes use of drawing air and fumes laterally across the top of vats or tanks into slotted ducts at the top and extending fully along one or more sides of the tanks. The slots are 2 in. wide and for effective

[6]Health Hazards in Chromium Plating, by J. J. Bloomfield and Wm. Blum. (U S. Public Health Report, Vol 43, No 26, September 7, 1928).

ventilation a 2,000 fpm exhaust air velocity at the slot face is advisable. In addition, the duct should not be required to draw the air laterally for a distance of more than 18 in. and the level of the solution should be kept 6 to 8 in. below the top of the tanks.

Flexible Exhaust Systems

The flexible exhaust tube method may be advantageously used for removing dust or fumes. Flexible tubes having one end connected to an exhaust system and a slotted hood attached to the other end may be shaped at will to fit in with industrial processes without affecting the ease of operation. Efficient dust or fume removal may be had with use of relatively small exhaust volumes. This type of system may be used on swing grinders, portable grinding wheels, soldering operations, stone cutting, rock drilling, etc.

Spray Booths

In the design of an efficient spray booth, it is essential to maintain an even distribution of air flow through the opening and about the object being sprayed. While in many instances spraying operations can be performed mechanically in wholly enclosed booths, the volatile vapors may reach injurious or explosive concentrations. At all times the concentrations of these vapors, and particularly those containing benzol, should be kept below 100 parts per million. Spray booth vapors are dangerous to the health of the worker and care should be taken to minimize exposure to them.

It is recommended in the design of spray booths that the exhaust duct be located in a horizontal position slightly below the object sprayed. Stagnant regions within the booth should be carefully avoided or should be provided with exhaust. The air volume should be sufficient to maintain a velocity of 150 to 200 fpm over the open area of the booth, and the vapors may be discharged through a suitable stack to permit dilution, but it is better practice to pass the fumes or vapors through baffle type washers or scrubbers designed for efficient spray fume removal[7].

Hoods for Chemical Laboratories

Hoods used in chemical laboratories are generally provided with sliding windows which permit positive control of the fumes and vapors evolved by the apparatus. Their design should offer easy access for the installation of chemical equipment and should be well lighted. Air velocities should exceed 50 fpm when the window is opened to its maximum height.

DUCT SYSTEM DESIGN

The duct system should be large enough to transport the fumes or material without causing serious obstruction to the air flow. It is good practice to proportion the ducts to obtain the desired velocities and suction pressures at the hoods, although in many cases only an approximation to an ideal design is possible. Many exhaust hoods, and par-

[7]For a discussion of spray booths, see Special Bulletin No. 16, Spray Painting in Pennsylvania, Department of Labor and Industry, 1926, Harrisburg, Pa

ticularly those used in buffing and polishing, are connected by short branch pipes to the main duct which renders proportioning impractical.

Construction

The ducts leading from the hoods to the exhaust fan should be constructed of sheet metal not lighter than is shown in Table 4. The piping should be free from dents, fins and projections on which refuse might catch.

All permanent circular joints should be lap-jointed, riveted and soldered, and all longitudinal joints either grooved and locked or riveted and soldered. Circular laps should be in the direction of the flow, and piping installed out-of-doors should not have the longitudinal laps at the bottom. Every change in pipe size should be made with an eccentric taper flat on the bottom, the taper to be at least 5 in. long for each inch change in diameter. All pipes passing through roofs should be equipped with collars so arranged as to prevent water leaking into the building.

The main trunks and branch pipes should be as short and straight as possible, strongly supported, and with the dead ends capped to permit inspection and cleaning. All branch pipes should join the main at an

TABLE 4. GAGE OF SHEET METAL TO BE USED FOR VARIOUS DUCT DIAMETERS

DIAMETER OF DUCT	GAGE OF METAL
8 in or less	24
9 to 18 in.	22
19 to 25 in.	20
26 in. or more	18

acute angle, the junction being at the side or top and never at the bottom of the main. Branch pipes should not join the main pipes at points where the material from one branch would tend to enter the branch on the opposite side of the main.

Cleanout openings having suitable covers should be placed in the main and branch pipes so that every part of the system can be easily reached in case the system clogs. Either a large cleanout door should be placed in the main suction pipe near the fan inlet, or a detachable section of pipe, held in place by lug bands, may be provided.

Elbows should be made at least two gages heavier than straight pipe of the same diameter, the better to enable them to withstand the additional wear caused by changing the direction of flow. They should preferably have a throat radius of at least one and one-half times the diameter of the pipe.

Every pipe should be kept open and unobstructed throughout its entire length, and no fixed screen should be placed in it, although the use of a trap at the junction of the hood and branch pipe is permissible, provided it is not allowed to fill up completely.

The passing of pipes through fire-walls should be avoided wherever possible, and sweep-up connections should be so arranged that foreign material cannot be easily introduced into them.

Table 5. Air Speeds in Ducts Necessary to Convey Various Materials

Material	Air Velocities (fpm)
Grain dust	2000
Wood chips and shavings	3000
Sawdust	2000
Jute dust	2000
Rubber dust	2000
Lint	1500
Metal dust (grindings)	2200
Lead dusts	5000
Brass turnings (fine)	4000
Fine coal	4000

At the point of entrance of a branch pipe with the main duct, there should be an increase in the latter equal to their sum. Some state codes specify that the combined area be increased by 25 per cent. While this is not always necessary and is frequently done at the expense of a reduced air velocity, it is none the less advisable where future expansion of the exhaust system is contemplated.

Air Velocities in Ducts

When the static suction has been fixed for a given hood, the air velocity in the duct may be determined from Equation 2. Air velocities for conveying a material should be moderate. Table 5 gives the velocities generally employed for conveying various substances. Equations 5a and 5b may be used as tests to determine the conveying efficiency of a system[8]. Velocities determined from these formulae should be increased by at least 25 per cent since they represent the minimum at which a stated size and density of material can be transported.

For vertical ducts:

$$V = 13{,}300 \frac{s}{s+1} d^{0.57} \tag{5a}$$

For horizontal ducts:

$$V = 6000 \frac{s}{s+1} d^{0.40} \tag{5b}$$

where

V = air velocity in duct, feet per minute
s = specific gravity of particles
d = average diameter of largest particles conveyed, inches.

Example 2 Granular material, the largest size of which is approximately 0.37 in. in diameter, with a specific gravity of 1.40 is to be conveyed in a vertical pipe the velocity of the air in which is 4100 fpm; find whether the material can be transported at this velocity

Substitute data in Equation 5a and multiply by 1.25:

$$V = 1.25 \times 13{,}300 \times \frac{1.4}{2.4} \times 0.37^{0.57}$$

Antilog (0.57 × log 0.37) = 0.568; the required velocity is, therefore, 5500 fpm

[8]Determining Minimum Air Velocities for Exhaust Systems, by J M DallaValle (A S H V E Journa Section, *Heating, Piping and Air Conditioning*, September, 1932)

TABLE 6. LOSS THROUGH 90-DEG ELBOWS

ELBOW CENTER LINE RADIUS IN PER CENT OF PIPE DIAMETER	LOSS IN PER CENT OF VELOCITY HEAD
50	75
100	26
150	17
200 to 300	14

Hence, the duct velocity must be increased either by speeding up the fan or decreasing the diameter of the duct, or both.

Duct Resistance

The resistance to flow in any galvanized duct riveted and soldered at the joints may be obtained from Fig. 3, Chapter 20. The pressure drop through elbows depends upon the radius of the bend. For elbows whose centerline radii vary from 50 to 300 per cent of pipe diameter, the loss may be estimated from Table 6. It is sometimes convenient to express the resistance of an elbow in terms of an equivalent length of duct of the same diameter. Thus with a throat radius equal to the pipe diameter the resistance is equivalent to a section of straight pipe approximately 10 diameters long, while with a throat diameter radius 1½ times the diameter, the resistance is about the same as that of seven diameters of straight pipe.

COLLECTORS

The most common method of separating the dust and other materials from the air is to pass the mixture through a centrifugal or *cyclone* collector. In this type of collector the mixture of the air and material is introduced on a tangent, near the cylindrical top of the collector, and the whirling motion sets up a centrifugal action causing the comparatively heavy materials suspended in the air to be thrown against the side of the separator, from which position they spiral down to the tail piece, while the air escapes through the stack at the center of the collector.

The diameter of the cyclone should be at least 3½ times the diameter of the fan discharge duct. When two or more separate ducts enter a cyclone, gates should be provided to prevent any back draft through a system which may not be operating. Cyclones working in conjunction with two or more fans should be designed to operate efficiently at two-thirds capacity rating. The following formula is useful in computing the loss through a cyclone when the velocity of the air in the fan discharge duct is known:

$$h_c = 0.13 \left(\frac{V}{1000}\right)^2 \qquad (6)$$

where

h_c = the pressure drop through the cyclone, inches of water.
V = the air velocity in the fan discharge duct, feet per minute.

If a cyclone is used to collect light dusts such as buffing wheel dusts,

feathers and lint, the exhaust vent should be large enough to permit an air velocity of 200 to 500 fpm. This will, of course, require a cyclone of larger dimensions than given for the foregoing general case.

When a high collection efficiency is desired, or the material is very fine, multicyclones may be used. These are merely small cyclones arranged in parallel which utilize the principle of high centrifugal velocity to attain separation. The capacities and characteristics of this type of separator should be obtained from the manufacturers.

Cloth Filters

Filters are used when the material collected by an exhaust system is valuable or cannot be separated efficiently from the air with an ordinary cyclone. They are also employed when it is desirable to recirculate the air drawn from a room by the exhaust system, which otherwise might entail considerable loss in heat. Bag filters which are properly housed may be operated under suction. *Bag houses* used in the manufacture of zinc oxide and other chemical products are operated on the positive side of the fan.

Wool, cotton and asbestos cloths are commonly used as filtering mediums. When woolen cloths are employed, the filtering capacities vary from ½ to 10 cfm per square foot of filtering surface, depending on the character of the material collected. The rates for cotton and asbestos cloths are lower. The type of filter cloth and the rates of filtration depend, of course, on the material to be collected and the fan capacity. The time increase of resistance varies with the amount of material permitted to build up on the surface of the filter and can be determined only by experiment. The limits of the increase may be regulated by adjustment of the shaking or cleaning mechanism. These limits may be regulated further according to the capacity of the fan and the effective performance of the hoods and the duct system.

For additional information on Dust and Cinders, see Chapter 15, Air Pollution, p. 301.

RESISTANCE OF SYSTEM

The maintained resistance of the exhaust system is composed of three factors: (1) loss through the hoods, (2) collector drop, and (3) friction drop in the pipes.

The loss through the hoods is usually assumed to be equal to the suction maintained at the hoods. The collector drop in inches of water is given approximately by Equation 6, but where possible the resistance of the particular collector to be used should be ascertained from the manufacturer.

Friction drop in the pipes must be computed for each section where there is a change in area or in velocity. Find the velocities in each section of pipe starting with the branch most remote from the fan. The friction drop for these sections can be determined by reference to Table 6. Total friction loss in the piping system is the friction drop in the most remote branch plus the drop in the various sections of the main, plus the drop in the discharge pipe.

EFFICIENCY OF EXHAUST SYSTEMS

The efficiency of an exhaust system depends upon its effectiveness in reducing the concentration of dusts, fumes, vapors and gases below the safe or threshold limits[9].

Too much emphasis cannot be placed on the necessity of testing exhaust systems frequently by determining the concentration of atmospheric contamination at the worker's breathing level. Commonly accepted values of threshold limits for the usual gases and vapors are given in Table 7.

SELECTION OF FANS AND MOTORS

Manufacturers generally provide special fans for the collection of various industrial wastes. These are available for the collection of coal dust, wood shavings, wool, cotton and many other substances. For

TABLE 7. THRESHOLD LIMITS OF COMMON VAPORS AND GASES[a]

SUBSTANCE	SPEC GRAV OF GAS OR VAPOR (AIR 1)	INFLAMMABLE LIMITS (%)	PHYSIOLOGICAL ACTION	MAXIMUM ALLOWABLE CONCENTRATION (PPM)
Chlorine	2.486	non-inflamm	irritant	0.35
Ozone	5.5	do	do	0.80
Hydrogen chloride	1.2678	do	do	10.0
Sulphur dioxide	2.2638	do	do	10.0
Carbon monoxide	0.9671	12.5–74	asphyxiant	100.0
Hydrogen sulphide	1.190	4.3–46	do	85–130
Benzene	2.73	1.4–7.0	anesthetic	100.0
Methanol	1.1	7.5–26.5	do	100.0
Carbon tetrachloride	5.3	non-inflamm.	do	100.0

[a]The Prevention of Occupational Diseases, by R R Sayers and J M DallaValle (*Mechanical Engineering*, Vol 57, No 4, April, 1935)

particular features concerning special fans, consult the *Catalog Data Section* of THE GUIDE and manufacturers' data. When substances having an abrasive character are conveyed, the fan blades and housing should be protected from wear. This may be accomplished by placing a collector on the negative side of the fan or by lining the housing and blades with rubber.

If no future expansion of an exhaust system is contemplated, the fan motor should be chosen to provide the calculated air volume. Should, however, the exhaust system be required to handle more air in the future, the motor should be adequate for the maximum load anticipated. Further information regarding the choice of fans and motors is given in Chapters 17 and 42.

PROTECTION AGAINST CORROSION

The removal of gases and fumes in many chemical plants requires that metals used in the construction of the exhaust system be resistant to

[9]Criteria for Industrial Exhaust Systems, by J. J. Bloomfield (A.S H.V.E. TRANSACTIONS, Vol. 40 1934)

TABLE 8 MATERIALS TO BE USED FOR THE PROTECTION OF EXHAUST SYSTEMS AGAINST CORROSION[a]

TYPE OF FUME CONVEYED	PROTECTIVE MATERIAL TO BE USED
Chlorine	Rubber lining or chrome-nickel alloys
Hydrogen sulphide	Aluminum coated iron, aluminum, high chrome-nickel alloys
Ammonia	Iron or steel
Sulphurous gases	High chrome-nickel alloys
Hydrochloric acid	Rubber lining, chrome-nickel alloys
Nitrous gases	Nickel-chrome alloys

[a]Condensed from data given by Chilton and Huey (*Industrial and Engineering Chemistry*, Vol 24, 1932)

chemical corrosion. A list of the materials which may be used to resist the action of certain fumes is given in Table 8. Hoods and ducts when short, may frequently be constructed of wood and be quite effective. Rubberized paints are available and may be applied as protective coatings in handling such gases and fumes as chlorine and hydrochloric acid.

PROBLEMS IN PRACTICE

1 ● What determines the efficiency of an exhaust system?

It is dependent upon its effectiveness in reducing the concentration of dust, fumes, vapors and gases below the safe or threshold limit

2 ● Are state regulatory requirements as to *suction* applicable to all sorts of dust collecting installations?

As a rule the regulations refer only to grinding wheel and buffing wheel systems

3 ● What is the most common method of reducing total air volumes handled in cases employing large hoods over apparatus covering a large area?

The use of the petticoats on large hoods which permits a comparatively high air velocity at the rim of the hood and controllably small velocities in the center.

4 ● What other types of collectors are available for use in the place of cyclones and filters when chemical and physical conditions obviate the possibility of the use of them?

Devices such as scrubbers and contactors, using water or other contacting liquids, and electrical precipitators.

5 ● What is the most frequent error made in dust collector system design?

The omission of some means of putting into the workroom air having the proper characteristics to replace that which has been exhausted.

6 ● Are there available means for testing the performance of dust collecting systems when they are required to meet high industrial hygienic standards?

Yes. Such means are set up by the United States Public Health Service and by the Standard Code for Testing and Rating Air Cleaning Devices Used in General Ventilation Work. (Chapter 44).

7 ● Why is it not permissible to connect up emery wheels and buffing wheels to the same exhaust system?

Emery wheels and buffing wheels should be handled by separate systems because of the fire hazard, as it is possible for sparks from the emery wheels to ignite the lint and dust from the buffing wheels when both are carried through the same system

8 ● Give an important characteristic of centrifugal type dust collectors which should be given consideration when applying this type of collector to installations requiring high separating efficiencies.

The separating action of a cyclone or centrifugal type collector depends largely on centrifugal force Reducing the radius of air flow increases the centrifugal force for a given velocity of flow. Accordingly, the smaller size units usually give higher separating factors, and better results can sometimes be obtained by using a number of small collectors instead of one large unit

9 ● Mention some general suggestions relating to the design of efficient industrial exhaust systems.

a. Endeavor to obtain a maximum degree of effectiveness with a minimum volume of air, by the use of well designed hoods closing in the sources of fumes or material to be removed so located as to take advantage of the natural direction taken by the fumes or materials when leaving their source

b. Give particular care to the velocity of flow The duct velocities for material conveying systems must be high enough to properly carry the material, but they should not be higher than necessary because excessive velocities increase the pressure requirements and result in a waste of power

c. Select the type of fan best suited to the job For installations where stringy material is handled do not use a fan wheel which has a shroud.

d. When handling the refuse from various machines, study the grouping and operating cycles of the machines. Connecting a large number of machines into one system is frequently very uneconomical.

e Avoid unnecessary distances and bends in laying out the piping system.

10 ● The static pressure measured at the throat of a buffing wheel hood is 2 in. and the velocity head measured with a Pitot tube is 1.6 in. Calculate the restriction coefficient f.

From Equation 2, $V = 4005 f \sqrt{h_t}$.

From the theory of air flow, $V = 4005 \sqrt{h_v}$.

Hence, $\sqrt{h_v} = f \sqrt{h_t}$.

or

$$f = \sqrt{\frac{h_v}{h_t}} = \sqrt{\frac{1\ 6}{2.0}} = 0.89$$

11 ● A tank, 4 ft by 8 ft, contains a fluid which gives off injurious vapors. A large hood is located 30 in. above the top of the tank and extends slightly over its edges. Assuming that a velocity of 60 fpm is required to adequately control the vapors near the edges of the tank, calculate the air flow required.

Using Equation 4, $P = 2 \times 4 + 2 \times 8 = 24$ ft, $D = 30$ in. $= 2\ 5$ ft; $V = 60$ fpm.

$$\text{Hence, } Q = 1.4 \times 24 \times 2\ 5 \times 60 = 5040 \text{ cfm}$$

12 ● Silica dust with a specific gravity of 2.65 is being conveyed in a duct system. The velocity measured in a vertical portion of the system is found to be 2700 fpm. What is the maximum diameter particle transported at this velocity?

$$\text{Using Equation 5a, } 2700 = 13{,}300 \times \frac{2.65}{3.65} \times d^{0.570}$$

from which

$$d = (0.28)^{1.75} = 0.11 \text{ in.}$$

13 ● What important factors must be considered in the determination of resistance for an exhaust system?

The maintained resistance of an exhaust system is composed of three factors, namely, (1) loss through the hoods, (2) friction drop through the pipe system, and (3) collector drop.

14 ● What special materials may be used to resist chemical corrosion in a system exhausting gases and fumes?

Various protective materials are available for exhaust systems depending largely upon the type of fumes conveyed. Nickel-chrome alloys, aluminum coated metals and rubber linings are extensively used Also protective rubberized paints are available which may be applied for conveying chlorine and hydrochloric acid fumes.

Chapter 22

RAILWAY AIR CONDITIONING

Passenger Car Ventilation, Steam or Vapor Heating, Cooling Equipment, Calculation of Car Cooling Load, Humidity Control, Temperature Control, Electric Power Supply, Comparative System Costs

THE general principles of air conditioning as applied to buildings also apply to railway cars, but due to space and weight limitations and the severity of the service, equipment designed for stationary work in buildings is seldom suitable for car installations. Equipment for railway use must be safe, reliable, compact, light in weight, accessible for inspection and repairs, as nearly automatic in operation as possible, and have low initial, operating, and maintenance costs. To properly air condition a car, ventilating, filtering, heating, cooling, humidifying, and control equipment must be provided together with an adequate power supply. Air from the interior of the car is mixed with air from the outside and passed through the air conditioning unit where it is heated or cooled, humidified or dehumidified and delivered to the interior of the car through suitable ducts and grilles.

PASSENGER CAR VENTILATION

One of the important problems in connection with air conditioning of cars is that of ventilation. In non-air conditioned cars, ventilation is accomplished by exhaust fans, roof ventilators and open doors and windows. This provides an ample supply of outside air and in addition a large amount of smoke and dirt which may be excluded in an air conditioned car.

An average car contains approximately 5000 cu ft of air which is being contaminated by the occupants who are continually liberating heat, carbon dioxide, moisture, odors, and some organic matter from the breath, skin and clothing. The heat and moisture can be removed by cooling and dehumidifying, but the others can be handled only by proper ventilation. In the average car from 2000 to 2500 cfm of air should be delivered by the air conditioning unit. Some of this air may be recirculated, but a portion of it should be brought in from the outside. The amount of outside air required depends upon the type of car (dining, club, cafe), number of passengers, air temperature, humidity, smoke, and odors, and will vary from 15 to 90 per cent. This percentage of outside air should be kept as low as possible to still maintain the air in the proper condition in order to minimize the heating or cooling load.

In equipping old cars, the air is often distributed to the car from the conditioning unit through one or two side ducts, built on the outside of

monitor-roofed cars and on the inside of turtle-back or arched-roof cars. In some instances, a center duct is used and in others the air is discharged directly from the conditioning units into the upper part of the car either through the end bulkheads or from a unit placed overhead in the center of the car. For details of air distribution and duct design see Chapters 19 and 20. Care should be taken to keep the air velocity in the ducts below the point where the noise would be objectionable.

Suitable grilles should be used at duct outlets to reduce the velocity and to direct the air so as to insure thorough mixing without objectionable drafts. The incoming air should be introduced as near room temperature as possible and should have a velocity of not over 120 fpm when it reaches the passengers. The outlet grille nearest the recirculating air grille should deflect 45 deg in one direction towards the center of the car to prevent the incoming air from passing directly to the recirculating grilles. The other outlet grilles installed along the ducts should deflect a portion of the air 45 deg to the right and 45 deg to the left and to all points between.

Smoking rooms present a special problem The cloud of smoke that usually hangs near the ceiling can be broken up by having the incoming air directed along the ceiling in all directions at a velocity somewhat higher than that used for the rest of the car. The air should be exhausted from the room by a fan or through a grille to the wash room or lavatory, to be exhausted to the outside through a ventilator.

For compartments a supply duct outlet grille of suitable size and design should be provided and provisions made in the door or partition for removal of the recirculated air through a special grille, which will allow the air to pass from the room to the main recirculating air grille. The exhaust grille should be designed and arranged so as to obstruct the vision of passengers.

Lower berths in sleepers and office cars should be provided with an adjustable air outlet which will discharge the amount of air desired at low velocity in any direction so that the occupant can regulate the ventilation to meet his own requirements.

Recirculating air grilles are usually of the straight flow type, arranged to hold two 2 by 16 by 20 in. filters and hinged on an angle iron frame to permit easy access for cleaning and repairs. They should be located so that objectional drafts will not be created by the return air. The outside air intakes should be of ample size and provided with filters and dampers or shutters for regulating the amount of outside air.

In cars containing but one or two rooms or compartments, satisfactory results may be obtained by discharging the air directly from the conditioning unit into the upper part of the car. Care must be taken to have a proper discharge velocity. If the velocity is too low, the air will drop before reaching the end of the car and if too high it will discharge against the end bulkhead and be reflected back. Greater care must be exercised to secure proper circulation without more objectional drafts than when ducts are used.

Filters made of metal wool, spun glass, hemp, paper, cloth, and wire screen are in use. Some types may be cleaned, retreated and returned to service while other types are discarded when dirty.

STEAM OR VAPOR HEATING

The majority of cars in service are heated by circulating low pressure steam or vapor through pipes located along the side walls near the floor. When an overhead air conditioning unit, using air from the outside and discharging it into the car, is installed it is necessary to provide a heating coil to warm the air during cold weather. Most of the heat required in the car can be supplied by the overhead unit but in extreme weather some heat must be supplied at the floor to keep the lower part of the car at a comfortable temperature. It is also desirable to have sufficient floor radiation to keep the car from freezing up while standing in the yard with the air conditioning unit shut off. Usually from 30 to 40 per cent of the heat required is supplied from the overhead unit and the balance from the floor heating system. The amount of heat required depends upon the type and construction of car, especially the amount and kind of insulation, outside temperature, wind velocity and direction, train speed, number of passengers, and inside temperature desired. In severe weather, −10 to −20 F, an average of approximately 200 lb of steam per car per hour is required. Pullmans will require approximately 250 lb per hour, coaches 150 to 175 lb and baggage cars 150 lb.

COOLING EQUIPMENT

Three general types of cooling or refrigerating equipment are being used with satisfactory results. These are the ice, steam jet, and mechanical compressor systems. These systems when arranged for car use, function the same as in stationary service, but must be more compact and lighter in weight. See Chapters 2 and 11 for the general principles of the various systems.

The mechanical compressor systems are divided into two general classes, the direct drive and the electro-mechanical, differing only in the method of driving the compressor. With the direct drive system, the compressor is driven from the car axle through a combination of pulleys, belts, gears, a shaft, and a speed control device. With the electro-mechanical system the compressor is driven by an electric motor. The refrigerant most commonly used in the mechanical compressor systems is Freon.

Another type of system which has possibilities of being adapted to railway air conditioning uses methylene chloride as the refrigerant, dry ice, and a small circulating pump in place of the conventional compressor. The refrigerating unit consists of a condenser built into an insulated dry ice box. The methylene chloride is liquified by the low temperature produced by the dry ice and is circulated by the pump through the evaporator coils where part of it is vaporized by the heat from the air passing over the evaporator coils. The mixture of gas and liquid passes back to the condenser where the heat is removed and the gas liquified. This system will probably not offer many possibilities until an adequate supply of dry ice can be assured at a stable and reasonable price.

A system using a compressor driven by an internal combustion engine operating on propane has also been considered for railway air conditioning. The engine, compressor, condenser, starting motor and battery are

mounted together in a single unit and supported on a track on the car under-frame on pneumatic supports. Propane fuel sufficient for several days' operation is carried in three drums mounted in a rack under the car. Freon is used as the refrigerant. The unit may be used with a direct expansion air conditioning unit or it can be used with a heat exchanger with cold water cooling coils.

The capacity required in the refrigerating system depends upon a number of factors such as size and type of construction of car, thickness and kind of insulation used, amount of heat produced within the car by motors, lights, and other appliances, amount of outside air, outside air temperature and humidity, intensity of solar radiation, number of passengers, and inside temperature desired. A check of a number of cars with ice-activated systems indicates an average ice consumption, day and night, of approximately 275 lb per hour. This means an average capacity of 3.3 tons of refrigeration. It has also been observed on test that in the sunshine with an outside dry-bulb temperature of 85 to 90 F and a wet-bulb temperature of 73 to 75 F, a six-ton compressor unit operates approximately 50 per cent of the time. As the average temperature during the cooling season is below 90 F the observed performance of the six-ton compressor unit compares with the 3.3 tons average capacity determined for the ice systems. It was also observed on test that with temperatures above 90 F, high humidity, and in the sunshine, a six ton unit operates almost continuously. The sun load on a bright sunshiny day is about one ton. For average cars on sunshiny days, with high temperatures and humidity, from 65,000 to 80,000 Btu per hour will have to be removed from the interior of the car to maintain an inside effective temperature within the comfort zone. This means that a refrigerating capacity of from 5.5 to 7 tons will be required.

CALCULATION OF CAR COOLING LOAD

Due to the many variables involved the calculated heat gain will be more or less of an approximation, but if a careful study is made, results sufficiently accurate for all practical purposes may be obtained. The following example illustrates a typical cooling load calculation.

Example and Solution

Type of car	= Arched-roof steel coach.	
Roof area	= 70.3 × 12 5	= 880 sq ft.
Floor area	= 70.3 × 9 8	= 687 sq ft.
Window area	= 38 × 2 5 × 2	= 190 sq ft
Net side wall area	= (2 × 7.5 × 70 3) − (190)	= 865 sq ft.
End area	= 2 × 7.5 × 9 8	= 147 sq ft

Average roof section. 1/16 in. steel sheet, 1 layer tar paper, 2 in air space, 1/2 in. hairfelt, 1/8 in. rigid fiber board insulation.

Average window section: two layers of 1/8 in. glass with 1 1/2 in. air space between.

Average floor section: 1 in. composition, 1/16 in. steel, 1/2 in air space, 1 in hairfelt, 1/16 in steel

Average side and end section: 3/16 in steel, 1/2 in hairfelt, 3 in air space, 3/4 in wood.

End area is taken as total area at body and bulkheads with no allowance made for glass as car is vestibuled and glass in body ends is not subjected to direct solar radiation

Outside dry-bulb temperature	t_o	= 95 F.
Outside wet-bulb temperature	t'_o	= 75 F.

Outside effective temperature	ET_o	= 83 F.
Outside relative humidity	rh_o	= 40 per cent
Inside dry-bulb temperature	t	= 80 F.
Inside wet-bulb temperature	t'	= 66 5 F.
Inside effective temperature	ET	= 73 5 F.
Inside relative humidity	rh	= 50 per cent.
Density of air at t	d	= 0 07353
Moisture per pound of dry air at t_o and t'_o	G_o	= 99 grains.
Moisture per pound of dry air at t and t'	G	= 77 grains
Moisture per pound of dry air to be removed	M	$= \frac{G_o - G}{7000} = \frac{99 - 77}{7000} = 0\ 00314$ lb
Evaporator condensate temperature		= 53 F.
Latent heat of water at 53 F	L	= 1060 Btu per pound.
Specific heat of air	C_p	= 0.241
Number of seated passengers	P_s	= 68.
Number of attendents	P_a	= 1
Heat given off by each passenger		= 400 Btu per hour.
Heat given off by attendent		= 650 Btu per hour.
Outside air introduced	Q	= 500 cfm

Roof temperature in sun = 40 F above ambient.
Wall temperature in sun = 25 F above ambient.
Sun 15 deg from Zenith.

Solar heat = 4.75 Btu per minute per square foot 30 per cent of heat passing through glass is reflected and reradiated, 70 per cent remaining. 80 per cent of heat at surface passes through glass Then 4.75 × Sine 15 deg × 0.8 × 0 7 = 0.7 Btu per square foot per minute passes through glass. Evaporator motor = ½ hp. Evaporator motor efficiency = 65 per cent The total heat gain will include the gains from leakage through roof, floor, side walls, end walls, windows, sensible and latent heats from outside air, heat from occupants, heat from evaporator fan motors and solar radiation.

Calculation of Transmission Coefficients. (See Chapter 5.)

Roof coefficient = U_r = Btu per hour per square foot per degree Fahrenheit.

Outside air film	$f_o = 6\ 00$	$1/f$	= 0.1660
$\frac{1}{16}$ in. steel	$k = 308$	$\frac{x}{k} = \frac{0.0625}{308}$	= 0.0002
2 in. air space	$a = 1.10$	$1/a$	= 0.9090
½ in. hairfelt	$k = 0\ 25$	$\frac{x}{k} = \frac{0.5}{0.25}$	= 2.0000
⅛ in. rigid fiber board insulation	$k = 0\ 33$	$\frac{x}{k} = \frac{0.125}{0\ 33}$	= 0 3800
Inside air film	$f_i = 1.65$	$1/f$	= 0.6080
Total			= 4.0632

$$U_r = \frac{1}{4.0632} = 0.246 \text{ Btu per hour per square foot per Fahrenheit.}$$

Side and end wall coefficient = U_s = Btu per hour per square foot per degree Fahrenheit.

Outside air film			= 0.1660
$\frac{3}{16}$ in. steel	$k = 308$	$\frac{x}{k} = \frac{0.1875}{308}$	= 0.0006
½ in. hairfelt	$k = 0\ 25$	$\frac{x}{k} = \frac{0.5}{0.25}$	= 2.0000
3 in air space	$a = 1.10$	$1/a$	= 0.9090

¾ in. wood	$k = 0.8$	$\frac{x}{k} = \frac{0.75}{0.8}$	= 0.9350
Inside air film			= 0.6080
Total			= 4.6186

$$U_s = \frac{1}{4.6186} = 0.217 \text{ Btu per hour per square foot per degree Fahrenheit}$$

Window Coefficient = U_w = Btu per hour per square foot per degree Fahrenheit

Outside air film			= 0.1660
⅛ in. glass	$k = 2.03$	$\frac{x}{k} = \frac{0.125}{2.03}$	= 0.0616
1½ in. air space	$a = 1.10$	$1/a$	= 0.9090
⅛ in. glass			= 0.0616
Inside air film			= 0.6080
Total			= 1.8062

$$U_w = \frac{1}{1.8062} = 0.553 \text{ Btu per hour per square foot per degree Fahrenheit}$$

Floor Coefficient = U_f = Btu per hour per square foot per degree Fahrenheit.

Outside air film			= 0.1660
1⁄16 in. steel			= 0.0002
½ in. air space	$a = 1.10$	$1/a$	= 0.9090
1 in. hairfelt	$k = 0.25$	$1/k$	= 4.0000
1⁄16 in. steel			= 0.0002
1 in. composition	$k = 6.0$	$1/k$	= 0.1666
Inside air film			= 0.6080
Total			= 5.8500

$$U_f = \frac{1}{5.85} = 0.171 \text{ Btu per hour per square foot per degree Fahrenheit}$$

Heat Leakage:

Through roof	= 880 × 0.246 = 216
side walls	= 865 × 0.217 = 188
floor	= 687 × 0.171 = 118
windows	= 190 × 0.553 = 105
end walls	= 147 × 0.217 = 32
	659 Btu per hour per degree Fahrenheit.

Total heat gain = $659 (t_o - t) = 659 \times 15 = 9890$ Btu per hour

Sensible heat gain from make-up air = $Q \times 60 \times d \times C_p \times (t_o - t)$
= $500 \times 60 \times 0.07353 \times 0.241 \times 15 = 7950$ Btu per hour.

Latent heat gain from outside air = $L \times 60 \times Q \times d \times M$
= $1060 \times 60 \times 500 \times 0.07353 \times 0.00314 = 7350$ Btu per hour.

Heat gain from passengers = $(68 \times 400) + (1 \times 650) = 27{,}850$ Btu per hour

Heat gain from evaporator motor = $\frac{0.50}{0.65} \times 0.746 \times 3415 = 1960$ Btu per hour

Heat gain from solar radiation:

Heat gain through roof = $U_r \times A_r \times 40 = 0.246 \times 880 \times 40 =$ 8670 Btu per hour

Heat gain through side wall = $0.217 \times 433 \times 25$ = 2350 Btu per hour.

Heat gain through window $= 0.7 \times \frac{190}{2} \times 60$	= 4000 Btu per hour
Total heat gain from solar radiation	= 15,020 Btu per hour.

Summary of Total Heat Gains

Leakage	9,890
Sensible heat from outside air	7,950
Latent heat from outside air	7,350
Heat from passengers	27,850
Heat from evaporator fan motors	1,960
Solar radiation	15,020
Total	70,020 Btu per hour

This would require a refrigerating capacity of $\frac{70,020}{12,000} = 5.83$ tons

HUMIDITY CONTROL

The temperature to be maintained in a car depends upon the outside temperature and the desired humidity inside the car. With a low humidity it is necessary to maintain a higher temperature to establish a desirable comfort condition. Little humidity control has been attempted on cars up to the present time. A certain degree of automatic humidity control is secured with cooling, but the relative humidity obtained depends largely on the temperature of the evaporator, which should be below the dew point temperature of the air. With certain outside atmospheric conditions it may not be possible to operate the conventional equipment with a sufficiently low evaporator temperature to reduce the humidity without dropping the temperature too low. One method has been developed whereby the evaporator temperature is carried below the dew point a sufficient amount to insure dehumidification and then the cold air is heated to the proper temperature by passing it over coils through which part of the high temperature liquid from the condenser is by-passed.

During the heating season humidification is desirable from a comfort standpoint, but the proper amount of humidification required would doubtless cause the windows to become frosted. A steam or water spray controlled by a humidistat will provide the necessary moisture if humidification is desired.

TEMPERATURE CONTROL

The control of the air conditioning equipment should be simple but at the same time as nearly automatic as possible. The use of a centralized control panel for all control switches, fuses, relay, etc., will simplify the installation and operation. Generally, separate thermostats are used for heating and cooling control. The best location for the thermostats depend upon the car layout and method of air distribution and can best be determined for any particular type of car and equipment by careful consideration of the several factors involved. The floor heat thermostats are usually located near the floor. The overhead heat and cooling thermostats are placed in the upper part of the car, sometimes in the air ducts. All thermostats should be located so that the air can circulate freely around them. Maintenance of uniform comfort conditions for cooling,

floor and overhead heating, has been satisfactory with provisions for high, medium, and low thermostat settings and in some cases, two settings have been satisfactory for cooling. In many cars the following points have been found to be satisfactory, 71 and 76 F for cooling, 60, 71 and 74 F for floor and overhead heating.

The heating and refrigerating equipment should be interlocked so that they cannot both operate at the same time. While heating, the control should be so arranged that in case of a steam failure the blower fan will stop or the outside air intake closed to prevent cool air from being introduced into the conditioned space.

ELECTRIC POWER SUPPLY

One of the most important problems to be solved in connection with railway car air conditioning is that of power supply. The majority of cars now in service are electric lighted and equipped with fans. Power is furnished by storage batteries and axle generators of from 2 to 5 kw capacity. When air conditioning is installed the electrical load is increased approximately 1 kw for ice systems, 3.5 kw for steam systems and 10 kw for electro-mechanical compressor systems. Steam ejector systems require approximately 230 lb of steam per car per hour for a six ton unit. All of this power as well as the power required to move the extra weight must be supplied by the locomotive enroute and if a number of cars in the train are air conditioned, the effect on train performance should not be overlooked

In the case of mechanical compressor systems the power is taken from the locomotive draw-bar, unless the compressor is driven by an internal combustion engine, while with the steam system most of the power is furnished in the form of steam which is a load on the locomotive boiler but not on the cylinders. The ice system takes the smallest amount of power from the locomotive since no power is required to produce refrigeration The demand for power for cooling comes, however, at the time of year when steam for heating is not required and the demand for lighting is at a minimum.

Inasmuch as axle generators and mechanical driven compressors will not operate below a certain cut in speed and will not carry full load until a speed considerably higher than cut in speed is reached, it is quite apparent that the characteristics of the run such as schedule speed, top speed, number of stops, length of stops, percentage of time of operation at slow speed, length of run, etc., must be considered in the selection of equipment. Fig. 1 shows the tractive resistance of a 75 ton car with six wheel trucks without an axle generator, and with a 4 kw generator and for the same car with an increase in weight of five tons and with a 20 kw axle generator load.

The direct-drive compressor system has only the friction of the drive during starting and at low speeds. After the compressor cuts in, the load increases with the speed. The compressor output increases with the speed until maximum output is reached and then the drive efficiency decreases as the speed continues to increase so that the power input to the drive continues to increase and the greater power requirements come at the higher speeds.

Consideration must also be given to the power requirements for refrigeration while the car is at a standstill or running at slow speeds. The electrical energy required for the ice activated and steam systems is easily supplied from the storage battery. With the ice system a supply of ice is necessary and steam for the steam system can be supplied from the locomotive or from a stationary plant. The majority of the electro-mechanical systems are equipped with A.C, D.C. motors. While standing in the yards and stations the A.C. motor is connected to a 220 volt, 3 phase circuit. The majority of these equipments are so arranged that while operating on A C. power the D.C. motor may be used as a generator for battery charging. If an auxiliary circuit is not available the D.C. compressor motor may be operated from the storage battery.

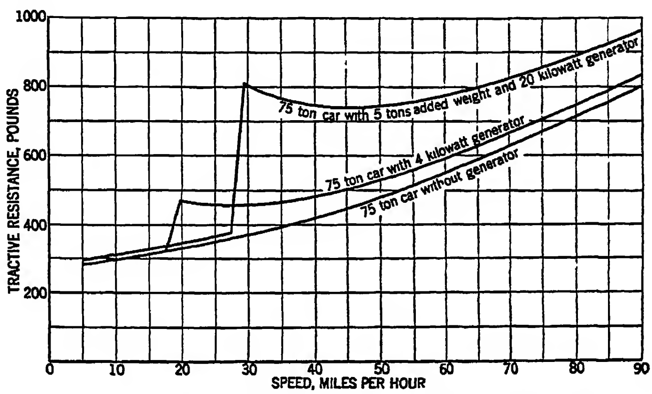


FIG. 1. TRACTIVE RESISTANCE OF 75 TON PASSENGER CAR WITH SIX WHEEL TRUCKS

The direct drive mechanical compressor systems are also equipped with A.C. motors for operation from auxiliary circuits. These equipments can only be operated when connected to the auxiliary circuit or while the train is running above the cut in speed of the drive.

COMPARATIVE SYSTEM COSTS

It is rather difficult to calculate the costs of car air conditioning due to the many variables involved. A study of reports covering a large number of cars of all kinds, operating under various conditions in all parts of the country, indicate a great variation in costs and weights for all systems. The following tabulation shows the average values for all cars considered.

	ICE	STEAM	MECH COMP ELECTRO	DIRECT DRIVE
Av. cost	$4000.00	$8200.00	$8200.00	$8200.00
Av. weight	8500 lb	11300 lb	9600 lb	8600 lb
Av. maintenance per 1000 car-miles	$2 00	$2 20	$3.85	$3 22
Av. ice capacity	4500 lb			
Av. ice consumption per hour	275 lb			

Based on an average cost of coal, water and locomotive lubrication of \$0.001133 per pound of fuel burned and the following assumptions: generator and drive efficiency 80 per cent, locomotive mechanical efficiency (cylinders to tender draw-bar) 90 per cent, 3.5 lb coal per cylinder horsepower and 6 lb of water per pound of coal, the cost of generating 1 kw of electrical energy would be \$0.00738, and 100 lb of steam \$0.01888. This is based on the assumption that the cost of coal, water and locomotive lubrication would increase in direct proportion to the amount of coal burned. As the cost of locomotive lubrication would not increase in direct proportion to the amount of coal burned, the extra cost of lubrication included in the figure should offset the increase in cost of locomotive maintenance due to the additional load. Using an average capacity of 3.3 tons, the cost per hour of operation would be approximately \$0.04 for steam and mechanical compressor systems and \$0.55 for ice, using a price of \$4.00 per ton for ice.

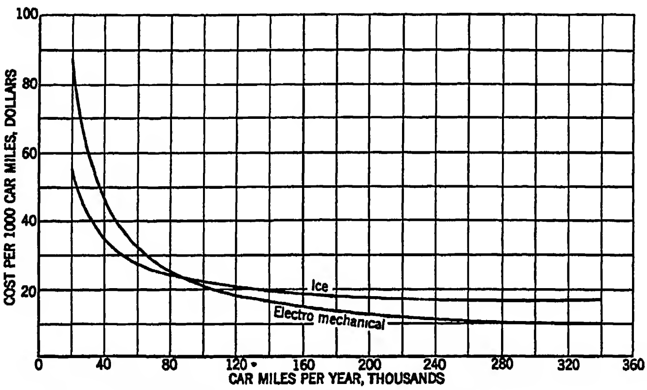


Fig. 2 Costs Railway Passenger Car Air Conditioning

These figures are based on the assumption that the motor driven compressor is receiving power direct from the generator and that the efficiency of the direct drive is the same as the combined efficiency of the axle generator, drive, and motor. Of course part of the current for the electric drive would be taken from the battery and therefore the battery efficiency would have to be taken into consideration. Likewise the lower efficiency should be considered of the direct drive at the higher speeds and it would be necessary to know the characteristics of the run under consideration to make an exact comparison.

For a given run the total kilowatt hours required for the electromechanical system would

$$= \frac{E_g \times I_g \times T_g}{1000} + \frac{E_b \times I_b \times T_b}{e_b \times 1000}$$

where

E_g = voltage with generator cut in.

E_b = voltage with generator cut out.

I_g = current with generator cut in
I_b = current with generator cut out
T_g = time generator cut in
T_b = time generator cut out
e_b = battery efficiency

$$\text{Cylinder horsepower hour} = \frac{\text{kilowatt hours} \times 1000}{e_g \times e_d \times e_l \times 746}$$

where

e_g = generator efficiency
e_d = drive efficiency.
e_l = locomotive mechanical efficiency

With these formulae and the coal and water rates of the locomotive the total amount of fuel and water required to supply the power enroute can be calculated.

Fig. 2 shows the relation between the cost per thousand car miles and total miles run per year for the ice and electro-mechanical systems. These curves are based on the cost figures previously given and the following formula:

$$\text{Cost per thousand car miles} = \frac{A}{CM \times D} + M + \frac{1000}{45}\,C$$

where

A = Annual fixed charges (dollars) calculated as follows:

a. Depreciation	12.5 per cent
b. Interest	6.0 per cent
c. Taxes and insurance	1.5 per cent
Total	20 0 per cent

CM = thousand car-miles per day.
D = days in service per year
M = maintenance cost per thousand car miles
C = operating cost per hour.
45 = average miles per hour.

No charges for precooling are included. The curves for the steam and direct drive mechanical compressor systems would be between those shown for the ice and electro-mechanical compressor systems.

PROBLEMS IN PRACTICE

1 ● What item is the greatest among the cooling loads figured in the design of a summer air conditioning system for a passenger car?

The heat from passengers

2 ● To what extent does bright sunshine increase the cooling requirements of a passenger car?

About one ton of refrigeration.

3 ● What is the total refrigerating capacity generally required in a passenger car?

5 5 to 7 tons per car.

4 ● What is the effect of train speed upon the cooling requirements of a car?

Requirements are slightly increased because of increased heat transmission

5 ● What is the capacity of the air conditioning unit in the average car?

2000 to 2500 cfm

6 ● When is it economical to take all air for car cooling from outdoors?

When the outdoor wet-bulb temperature is lower than that in the car

7 ● What various arrangements are used for distributing cooled air into cars?

Bulkhead delivery at center or ends of car, center duct, and side duct on one or both sides.

8 ● What types of cooling systems are used?

Ice, steam jet, and mechanical compressor systems

9 ● What cooling medium is used for condensing the refrigerant in a railroad air conditioning system?

Outdoor air

10 ● How may adequate cooling of condensers be provided in hot desert regions?

By evaporative cooling with water sprays.

11 ● How is the temperature controlled in railroad cooling systems?

By intermittent operation of the compressor or the steam jet or the ice-water circulating pump.

12 ● How much steam is required for car heating on the coldest days?

Pullmans 250 lb per hour, coaches 150 to 175 lb per hour, baggage cars 150 lb per hour

13 ● At present costs which is likely to be the greater, fixed charges or operating costs?

The fixed charges for steam and mechanical compression systems, and operating costs for ice systems

14 ● What would be the annual operating cost for a car cooling system, using ice, if the car travels 150,000 car miles?

$2,850

Chapter 23

GRAVITY WARM AIR FURNACE SYSTEMS

Design Procedure, Estimating Heating Requirements, Leader Pipe Sizes, Proportioning Wall Stacks, Register Selections, Recirculating Ducts and Grilles, Furnace Return Connection, Furnace Capacity, Examples, Booster Fans

WARM air heating systems of the gravity type are described in this chapter[1], and those of the mechanical type are described in Chapter 24. In the gravity type, the motive head producing flow depends upon the difference in weight between the heated air leaving the top of the casing and the cooled air entering the bottom of the casing, while in the mechanical type a fan may supply all or part of the motive head. Booster fans are often used in conjunction with gravity-designed systems to increase air circulation.

In general, a warm-air furnace heating plant consists of a fuel-burning furnace or heater, enclosed in a casing of sheet metal or brick, which is placed in the basement of the building. The heated air, taken from the top or sides near the top of the furnace casing, is distributed to the various rooms of the building through sheet metal warm-air pipes. The warm-air pipes in the basement are known as leaders, and the vertical warm-air pipes which are run in the inside partitions of the building are called stacks. The heated air is finally discharged into the rooms through registers which are set in register boxes placed either in the floor or in the side wall, usually at or near the baseboard.

The air supply to the furnace may be taken (1) entirely from inside the building through one or more recirculating ducts, (2) entirely from outside the building, in which case no air is recirculated, or (3) through a combination of the inside and the outside air supply systems.

DESIGN PROCEDURE

The design of a furnace heating system involves the determination of the following items:

1. Heat loss in Btu from each room in the building.
2. Area and diameter in inches of warm-air pipes in basement (known as leaders).
3. Area and dimensions in inches of vertical pipes (known as wall stacks).
4. Free and gross area and dimensions in inches of warm-air registers
5. Area and dimensions of recirculating or outside air ducts, in inches.
6. Free and gross area and dimensions in inches of recirculating registers.

---

[1]All figures and much of the engineering data which follow are from University of Illinois *Engineering Experiment Station Bulletins Nos* 141, 188, 189 and 246, Warm Air Furnaces and Heating Systems, by A. C. Willard, A P. Kratz, V S Day, and S. Konzo.

7. Size of furnace necessary to supply the warm air required to overcome the heat loss from the building. This size should include square inches of leader pipe area which the furnace must supply. It is also desirable to call for a minimum bottom fire-pot diameter in inches, which is the nominal grate diameter.

8. Area and dimensions in inches of chimney and smoke pipe. If an unlined chimney is to be used, that fact should be made clear.

The heat loss calculations should be made in accordance with the procedure outlined in Chapter 7, taking into consideration the transmission losses as well as the infiltration losses.

LEADER PIPE SIZES

In a gravity circulating warm-air furnace system the size of the leader to a given room depends upon the temperature of the warm air entering the room at the register. A reasonable air temperature at the registers must, therefore, be chosen before the system can be designed. The *National Warm Air Heating and Air Conditioning Association* has approved an air temperature of 175 F at the registers as satisfactory for design purposes. At this temperature, the heat-carrying capacity (heat available above 70 F) per square inch of leader pipe per hour for first, second or third floors is shown by Fig. 1 at 175 F to be 105, 170 and 208 Btu, respectively. For average calculations, the values 111, 166 and 200 will simplify the work and may be satisfactorily substituted for these heat-carrying capacities. If H represents the total heat to be supplied any room, the resulting equations are:

$$\text{Leader areas for first floor, square inches} = \frac{H}{111} = \text{approximately } 0.009H \qquad (1)$$

$$\text{Leader areas for second floor, square inches} = \frac{H}{166} = \text{approximately } 0.006H \qquad (2)$$

$$\text{Leader areas for third floor, square inches} = \frac{H}{200} = \text{approximately } 0.005H \qquad (3)$$

In designing for a lower warm-air register temperature, say 160 F, the factors 111, 166 and 200 become 80, 140 and 166 (Fig. 1 at 160 F), and the resulting equations are:

$$\text{Leader areas for first floor, square inches} = \frac{H}{80} = \text{approximately } 0.012H \qquad (4)$$

$$\text{Leader areas for second floor, square inches} = \frac{H}{140} = \text{approximately } 0.007H \qquad (5)$$

$$\text{Leader areas for third floor, square inches} = \frac{H}{166} = \text{approximately } 0.006H \qquad (6)$$

These equations are applicable to straight leaders from 6 to 8 ft in length. Longer leaders must be thoroughly covered or the vertical stacks must be increased in area as discussed under wall stacks. If some provision is not made for these longer leaders, the air temperature may be much lower than anticipated and the room will not be properly heated.

The values shown in Fig. 1 apply only to the case where the straight, leader pipe is 8 ft in length and is connected to stacks whose cross-sectional area is approximately 75 per cent of that of the leader pipe.

Any deviation from these conditions requires a modification of the constants used in Equations 1, 2, and 3. The temperature drop in leaders of various lengths at three different register temperatures is shown in Fig. 2, and should be used to obtain new register temperatures, lower than 175 F, on which to base selections from the curves of Fig. 1, and thereby new constants for Equations 1, 2 and 3.

Leader sizes should in general be not less than those obtained by Equations 1 to 3 nor should leaders less than 8 in. in diameter be used. It is not considered good commercial practice to specify diameters except

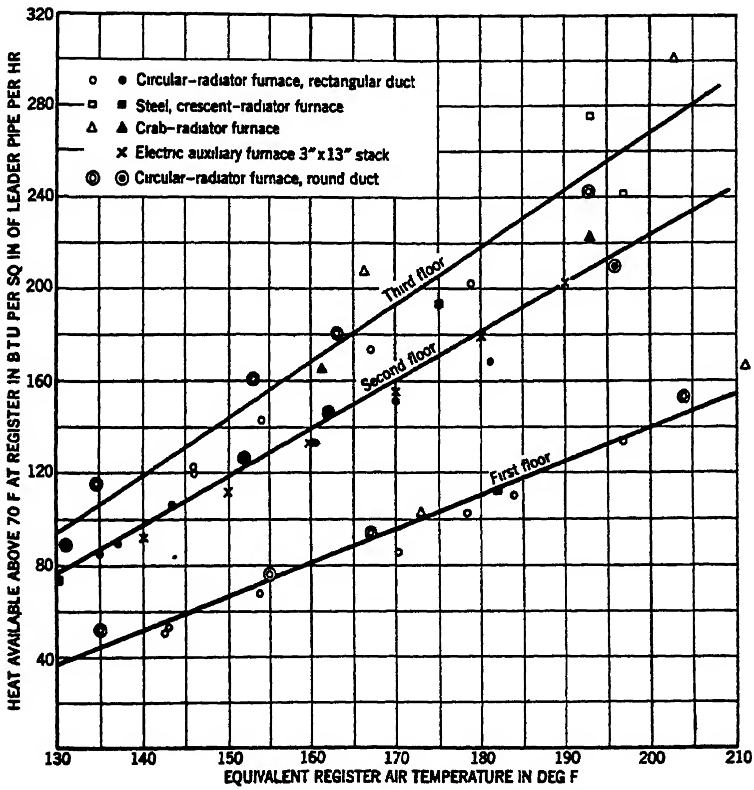


FIG. 1. VALUE OF SQUARE INCH OF LEADER PIPE AREA FOR FIRST, SECOND, AND THIRD FLOORS FOR SIMPLE SYSTEM HAVING LEADERS 8 FT IN LENGTH

in whole inches. The tops of all leaders should be at the same elevation as they leave the furnace bonnet, and from this point there should be a uniform up-grade of 1 in. per foot of run in all cases. Leaders over 12 ft in length should be avoided if possible. In cases where such leaders are required, the use of a larger size pipe, than is required by the application of the equations, smooth transition fittings, and duct insulation are recommended.

PROPORTIONING WALL STACKS

The wall stack for an upper floor should be made not less than 70 per cent of the area of the leader. In cases where the leader is short and straight as was the case for Fig. 1, such a practice is probably justified, since the loss (Fig. 3) in capacity occasioned by the smaller stack is not serious for stacks having areas in excess of 70 per cent of the leader area. For leaders over 8 ft in length or for leaders which are not straight, the ratio of stack area to leader area should be greater than 70 per cent in

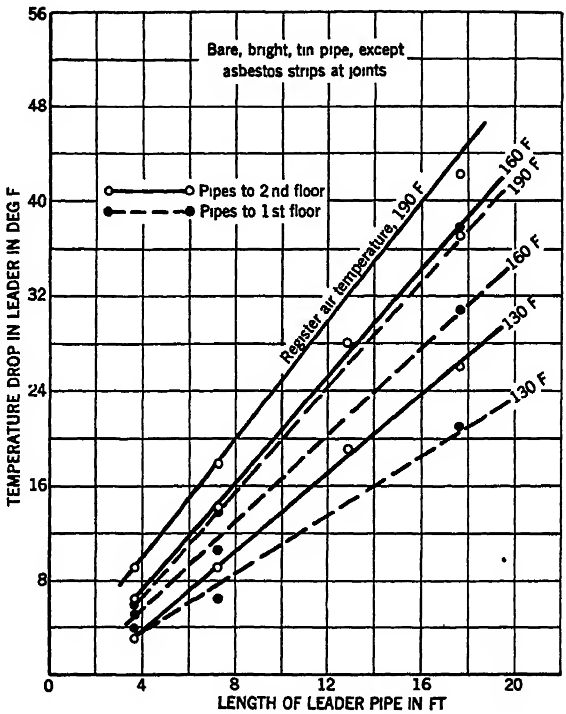


Fig. 2. Influence of Leader Pipe Length on Temperature Loss in Air Flowing through Pipe

order to offset the greater temperature losses (Fig. 2) in the longer leader. In gravity circulating systems, this ratio of stack to leader area is a very important matter.

The curves in Figs. 4 and 5 indicate that for rooms having a heat requirement exceeding approximately 9000 Btu per hr, exceedingly high register temperatures are required for stacks whose width is less than 3½ in. For such requirements either multiple stacks, or stacks having larger cross-sectional area (placed in 6 in. studding spaces) will be required.

REGISTER SELECTIONS

The registers used for discharging warm air into the rooms should have a free or net area not less than the area of the leader in the same run of piping. The free area should be at least 70 per cent of the gross area of the register. No upper-floor register should be wider horizontally than the wall stack, and it should be placed either in the baseboard or side wall, if this can be done without the use of offsets. First-floor registers may be of the baseboard or floor type, with the former location preferred. High

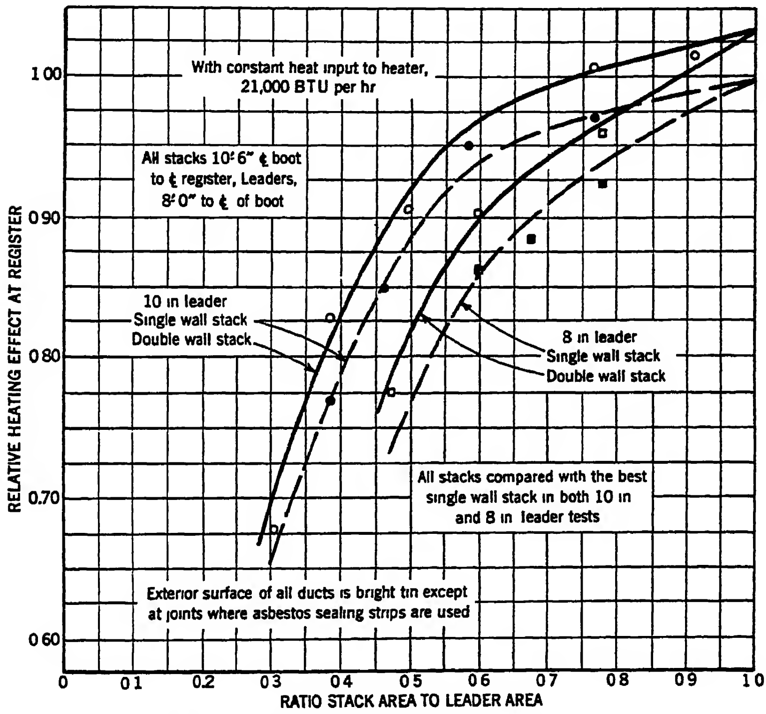


FIG. 3. RELATIVE HEATING EFFECT OF STACKS AT CONSTANT HEAT INPUT TO FURNACE

Note.—Exterior surface of all ducts is bright tin except at joints where asbestos sealing strips are used

sidewall locations for warm air registers in gravity circulating systems are not recommended on account of the tendency for stratification of the air in the room, resulting in high temperatures at the ceiling.

RECIRCULATING DUCTS AND GRILLES

The ducts through which air is returned to the furnace should be designed to minimize friction and turbulence. They should be of ample area, in excess of the total area of warm-air pipes, and at all points where

the air stream must change direction or shape, streamline fittings should be employed. Horizontal ducts should pitch at least ½ in. per foot upward from the furnace.

The recirculating grilles (or registers) should have a free area at least equal to the ducts to which they connect, and their free area should never be less than 50 per cent of their gross area.

The location and number of return grilles will depend on the size, details and exposure of the house. Small compactly built houses may frequently be adequately served by a single return effectively placed in a central hall. More often it is desirable to have two or more returns, provided, however, that in two-story residences one return is placed to effectively receive the cold air returning by way of the stairs.

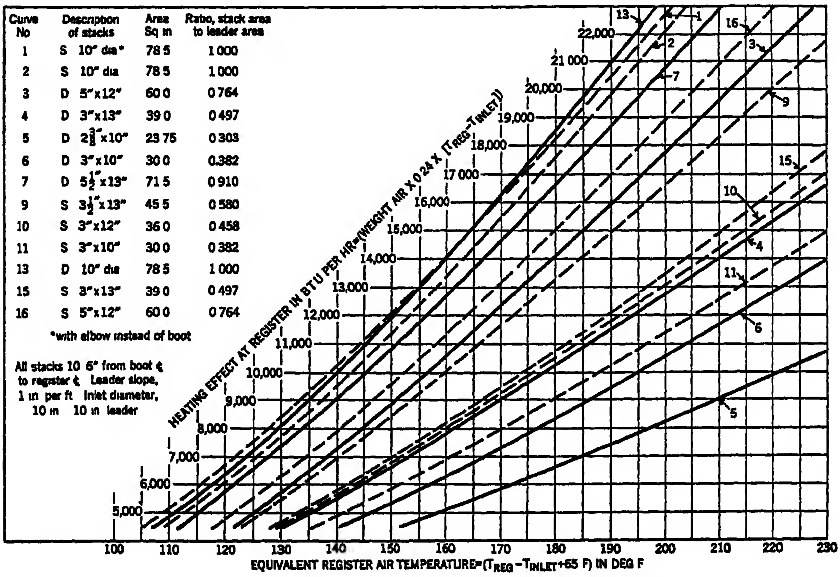


Fig 4. Heating Effect at Registers for Various Stacks with 10-in. Leader

Where a divided system of two or more returns is used, the grilles must be placed to serve the maximum area of cold wall or windows. Thus in rooms having only small windows the grille should be brought as close to the furnace as possible, but if the room has a bay window, French doors, or other large sources of cooling or leakage of cold air, the grille should be placed close by, so as to collect the cool air and prevent drafts. When long ducts of this type are employed they must be made oversize. This precaution is particularly important when long ducts and short ducts are used in the same system. The long ducts must be oversize, if they are to operate satisfactorily in parallel with short ducts.

Return ducts from upstairs rooms may be necessary in apartments or other spaces which are closed off or badly exposed. Metal linings are

advisable in such ducts. It is important that these ducts be free from unnecessary friction and turbulence, and that they be located to prevent preheating of the air before it reaches the furnace.

Furnace Return Connection

Circulation of the air is accelerated if the return connection to the furnace is through a round inclined pipe connected to two 45 deg elbows rather than through a vertical pipe connected to two 90 deg elbows. The top of the return shoe should enter the casing below the level of the grate in the furnace. In order to accomplish this the shoe must be wide as is indicated in Fig. 6, No. 1 arrangement.

Tests of six different systems of cold air returns, Fig. 6, made at the University of Illinois[2], resulted in the following conclusions:

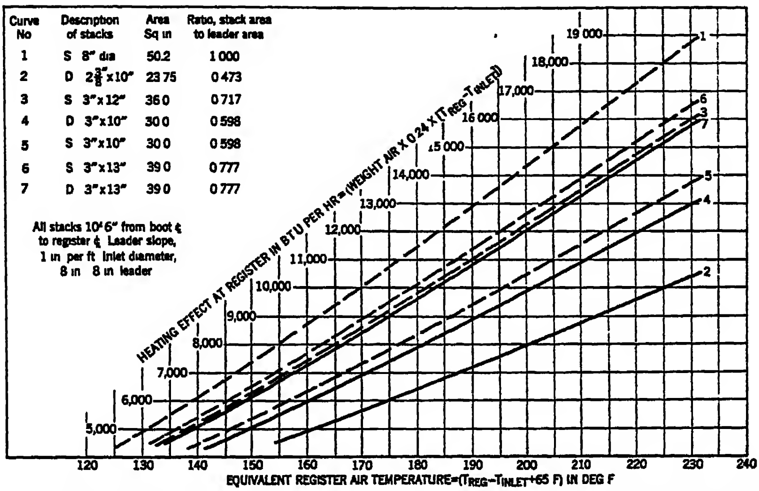


FIG. 5. HEATING EFFECT AT REGISTERS FOR VARIOUS STACKS WITH 8-IN. LEADER

1. In general, somewhat better room temperature conditions may be obtained by returning the air from positions near the cold walls.

2. Friction and turbulence in elaborate return duct systems retard the flow of air, and may seriously reduce furnace efficiency, and lessen the advantages of such a design.

3. The cross-sectional duct area is not the only measure of effectiveness. Friction and turbulence may operate to make the air flow out of all proportion to the various duct areas.

FURNACE CAPACITY

The size of furnace should, of course, be such as will provide the necessary air heating capacity, usually expressed in square inches of leader pipe area, and at the same time provide a grate of the proper area to burn the necessary fuel at a reasonable chimney draft. The total leader pipe area required is obtained by finding the sum of the leader pipe areas as already designated.

---

[2]Investigation of Warm-Air Furnaces and Heating Systems, Part IV, by A. C Willard, A. P. Kratz, and V S Day (University of Illinois *Engineering Experiment Station Bulletin No* 189).

The grate area will depend on several factors of which four are very important. First of all, the air temperature at the register for which the plant has been designed must be determined. Usually, this temperature is taken at 175 F. Second in importance is the combustion rate, *which must always correspond with the register air temperature*, as is shown by a set of typical furnace performance curves (Fig. 7) for a cast-iron, circular radiator furnace with a 23 in. diameter grate and 50 in. diameter casing. The third factor is efficiency, which is a function of the combustion rate, and varies with it as shown by the efficiency curve of Fig. 7. The fourth factor is the heat value per pound of fuel burned, which was 12,790 Btu. This is not shown on the curves since it was constant for all combustion rates.

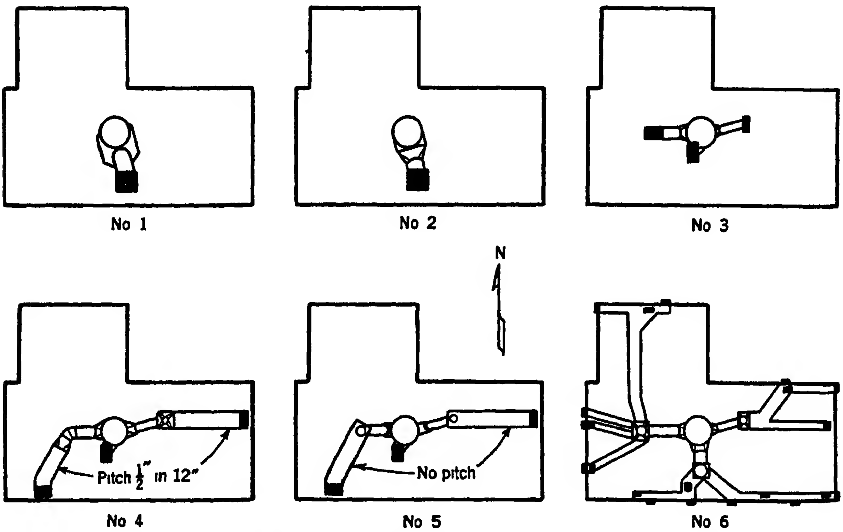


Fig. 6. Arrangement of Cold Air Returns for Six Installations

It may be noted from Fig. 7 that for this particular furnace a register temperature of 175 F was accompanied by a combustion rate of approximately 7.5 lb per sq ft per hr, a capacity at the bonnet of 152,000 Btu per hr and a furnace efficiency of 58 per cent. Under these conditions the capacity at the bonnet per square foot of grate was equivalent to a value of 52,800 Btu per hr and per square inch of grate was equivalent to 367 Btu per hr. If it is desired to use these curves to select a furnace to deliver air at 175 F register temperature in a house where the total heat loss is H Btu per hour and the loss between the furnace and the registers is 0.25 H Btu per hour, the area of the grate in square inches will be

$$\frac{1.25\,H}{367} = 0.0034\,H.$$

If, on the other hand, it is desired to select a furnace to deliver air at 160 F register temperature, the combustion rate is 5.5 lb and the efficiency

of the furnace is 62 per cent. Under this condition the capacity at the furnace bonnet per square foot of grate is 43,200 Btu per hr and per square inch of grate is 300 Btu per hr, the required area of the grate in square inches in this case will be $\frac{1.25\ H}{300} = 0.0042\ H$. It should be noted that a larger grate area is required if the furnace is to deliver air at a lower register temperature.

The typical performance curves shown in Fig. 7 are not applicable to

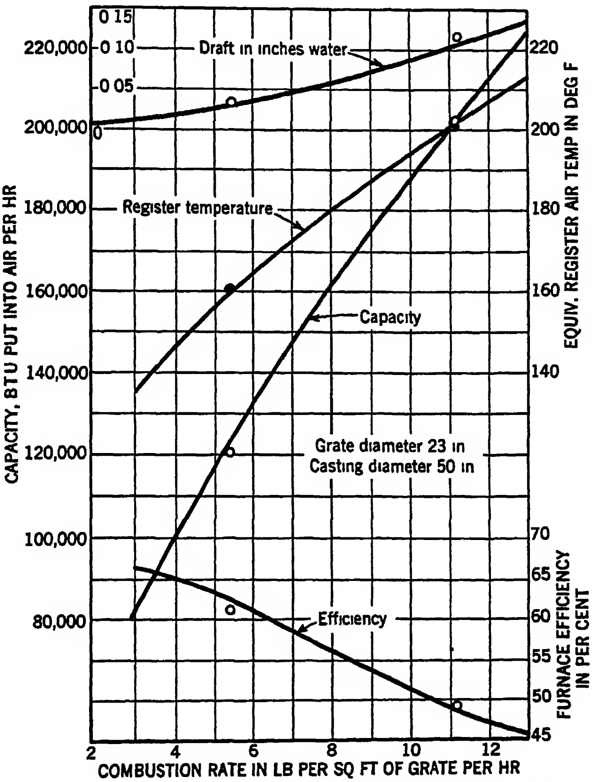


FIG. 7. TYPICAL PERFORMANCE CURVES FOR A WARM AIR FURNACE AND INSTALLATION IN A THREE STORY TEN LEADER PLANT, OPERATING ON RECIRCULATED AIR

all furnaces and hence for ordinary design purposes the values recommended in the Standard Code[3] should be used. The equation for a furnace having a ratio of heating surface to grate area of 20 to 1 is equal to:

$$H = \frac{G \times p \times f \times E_1 \times E_2 \times 0.866}{144} \qquad (7)$$

[3]Standard Code Regulating the Installation of Gravity Warm Air Heating Systems in Residences. This code has been sponsored by the *National Warm Air Heating and Air Conditioning Association*, the *National Association of Sheet Metal Contractors*, and the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS It is recommended that the installation of all gravity warm air heating systems in residences be governed by the provisions of this code, the eighth edition of which may be obtained from the *National Warm Air Heating and Air Conditioning Association*, 50 W Broad St., Columbus, Ohio

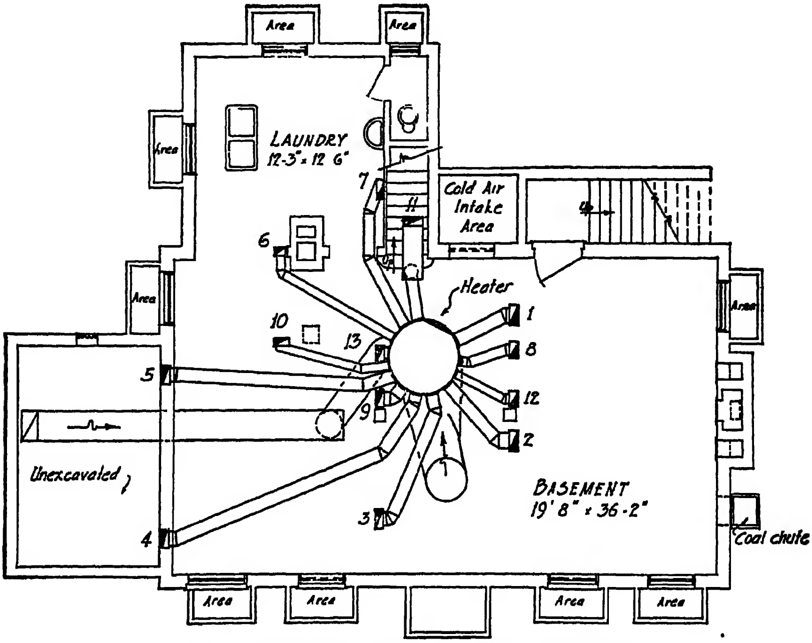


Fig. 8. Basement Plan, Research Residence

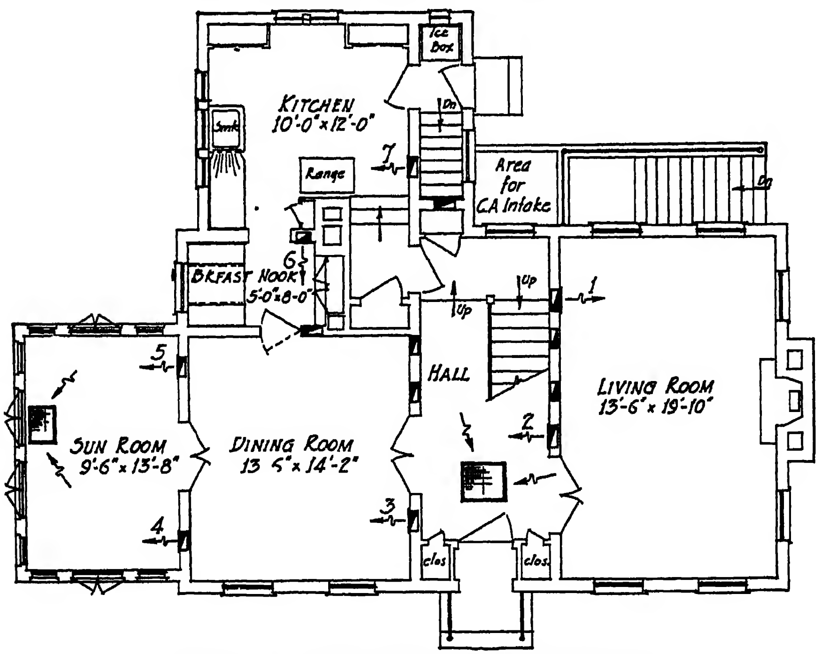


Fig. 9. First-Floor Plan, Research Residence

where

G = grate area, square inch
p = combustion rate, pound coal per square foot of grate per hour
f = heating value of the coal, Btu per pound.
E_1 = efficiency at bonnet, ratio of heat delivered at bonnet to heat developed in furnace
E_2 = efficiency of duct transmission, ratio of heat delivered at register to heat delivered at bonnet
0.866 = factor of safety to allow for contingencies under service conditions such as accumulations of soot and ashes, ineffective firing methods, etc
H = total heat loss from structure

An addition of 2 per cent of the furnace capacity is proposed for each unit that that ratio of heating surface to grate area exceeds 20. This addition is based on tests[4] conducted at the University of Illinois on seven types of furnaces having varying ratios of heating surface to grate area. This correction does not, however, apply to values of the ratio less than 15 nor greater than 30.

By transposing the terms in Equation 7 and adding the correction term for ratios of heating surface to grate area other than 20 to 1, the following equation is obtained:

$$G = \frac{144 \times H}{p \times f \times E_1 \times E_2 \times 0.866\ [1 + 0.02\ (R\text{-}20)]} \tag{8}$$

in which R = ratio of heating surface to grate area.

In the case of the Standard Code[5] the numerical values used in Equation 8 were based on those determined from the tests conducted on the different types of furnaces.

$$G = \frac{144 \times H}{7.5 \times 12{,}790 \times 0.55 \times 0.75 \times 0.866\ [1 + 0.02\ (R\text{-}20)]} \tag{9}$$

$$G = 0.004205 \frac{H}{[1 + 0.02\ (R\text{-}20)]} \tag{10}$$

As used in these calculations, H = Btu heat loss from the entire house per hour = summation of all room losses $H_1 + H_2 +$ etc. + the Btu necessary to heat the fresh air, if any, at intake. This fresh air loss in Btu per hour will be approximately 1.27 times the cubic feet of air admitted through the intake per hour on a zero day. For systems which recirculate all the air this value will be zero. For systems which have a fresh air intake, controlled by damper, this value might well be approximated, since this loss will probably be reduced to a minimum on a zero day. Assume for such cases that the building loss is increased by 25 per cent, and that there is the usual 25 per cent loss between furnace and registers.

TYPICAL DESIGN

The application of the preceding data to an actual example may be of assistance to the designer. Figs. 8, 9, 10 and 11 represent the plans of

[4]University of Illinois *Engineering Experiment Station Bulletin* No. 246, by A C Willard, A P Kratz, and S Konzo, Chapter X, pp 126-146

[5]Loc Cit Note 3.

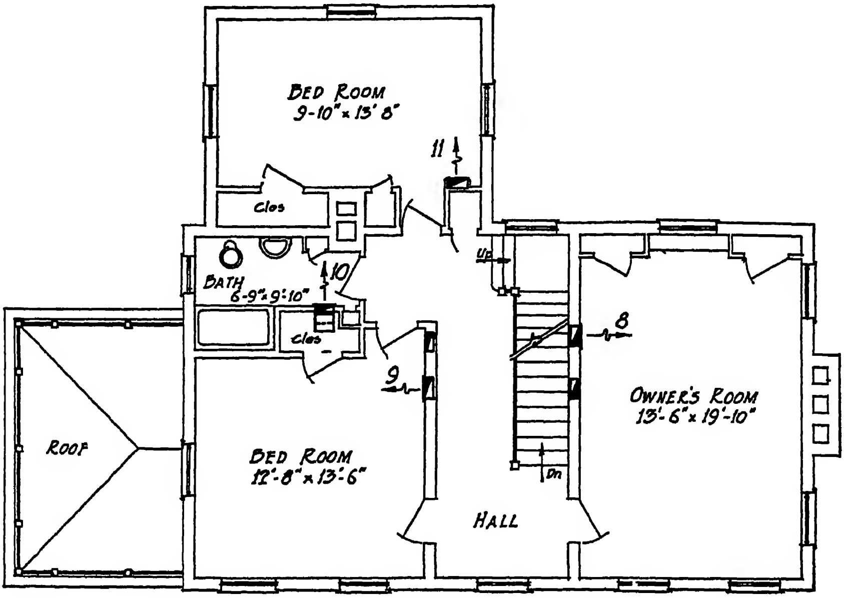


FIG 10 SECOND-FLOOR PLAN, RESEARCH RESIDENCE

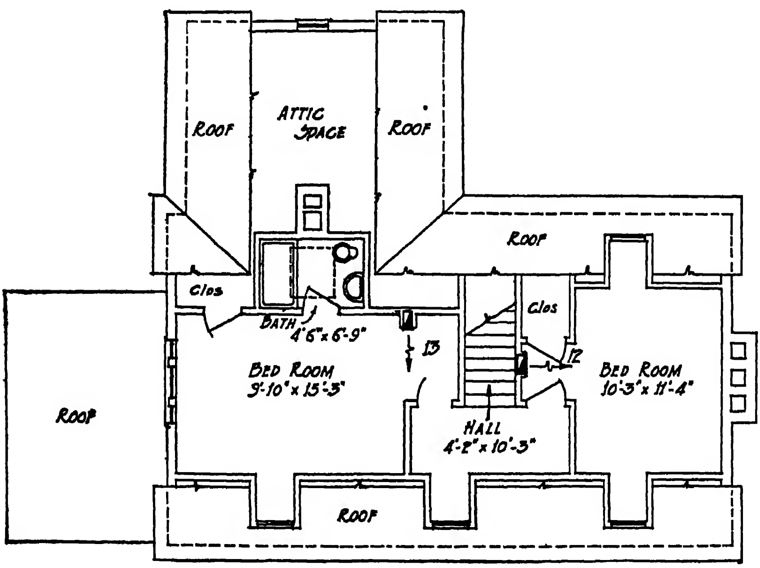


FIG. 11. THIRD-FLOOR PLAN, RESEARCH RESIDENCE

the Warm Air Research Residence of the *National Warm Air Heating and Air Conditioning Association* erected at the University of Illinois[6].

Leaders, Stacks and Registers. (Direct Method)

Living Room, 1st floor:

17,250 ÷ 111 = 155 sq in leader area. See summary, Table 1, also example under Standard Code[7], Art. 3, Basis of Working Rules for Pipes.

Leader diameter = 14 in.

Register size = 155 sq in. net area. Gross area = net area ÷ 0.7 = 14 in × 16 in.

Owner's Room, 2nd floor:

15,030 ÷ 167 = 90 sq in. leader area. See summary Table 1; also example under Standard Code[7], Art. 3, Basis of Working Rules for Pipes.

Leader diameter = 11.4, say 12 in

Stack area = 0.7 × 90 = 63 sq in. = say 5 in. × 12 in.

Register area = 90 sq in. net area. Gross area = net area ÷ 0.7 = 12 × 12 or 12 in. × 14 in.

In like manner the leaders, stacks and registers are calculated for each room in the house.

Leaders, Stacks and Registers. (Code [7] Method. See Art. 3, Sec. 1, 2, 3)

Living Room (Glass = 90, Net wall = 405, Cubic contents = 2405)

$$\text{Leader} = \left(\frac{90}{12} + \frac{405}{60} + \frac{2405}{800}\right) 9 = 155 \text{ sq in.}$$

Register, same as Direct Method.

Owner's Room (Glass = 68, Net wall = 394, Cubic contents = 2275)

$$\text{Leader} = \left(\frac{68}{12} + \frac{394}{60} + \frac{2275}{800}\right) 6 = 90 \text{ sq in.}$$

Stack and Register, same as Direct Method.

Assuming all air recirculated, the minimum furnace for the plant will be:

Grate area = 0 0042 × 132,370 = 556 sq in.
Use 27 in. diameter grate. (Equation 10).

If provision should be made for certain outside air circulation, then increase the building heat loss by, say 25 per cent and obtain by Equation 10 a 30 in. grate.

Experiments at the University of Illinois[8] have shown that the capacity of a furnace may be increased nearly three times by an adequate fan, with a constant register or delivery temperature maintained, *provided that the rate of fuel consumption can be increased to provide the necessary heat.* In other words, the capacity of a forced circulation system *is limited by the ability of the chimney to produce a sufficient draft, and the ability of the fan to deliver an adequate amount of air.*

[6]Plans used with permission Bathroom on third floor not heated.
[7]Loc Cit Note 3.
[8]University of Illinois *Eng Exp Sta. Bulletin No* 120, p 129.

Table 1 Summary of Data Applied to Warm Air Research Residence

Rooms	From Chapter 7 Estimating Heat Losses Btu Heat Losses H	Leader Area Sq In	Stack Area Sq In $0.7 \times LA$	Leader Diameter Inches	Stack Size Net	Register Size Gross
First Floor		$= 0.009H$				
Living	17250	155	—	14	....	14 × 16
Dining	6810	61	—	9	....	8 × 12
Breakfast	2300	21	—	8	....	8 × 10
Kitchen	9210	83	—	11 or 12	....	12 × 14
Sun	25710	230	—	Two 12	....	Two 12 × 14
Hall and stair	12570	113	—	12	....	12 × 14
Second Floor		$= 0.006H$				
Owner's	15030	90	63	11 or 12	5 × 12	12 × 14
S. W. Bed	9800	59	41	9	3½ × 12	8 × 12
Bath	2450	15	10	8	3 × 10	8 × 10
N. Bed	14800	89	62	11 or 12	5 × 12	12 × 14
Third Floor		$= 0.005H$				
E Bed	8220	41	29	8	3 × 10	8 × 10
W. Bed	8220	41	29	8	3 × 10	8 × 10

BOOSTER FANS

Booster fans often may be arranged to operate when gas or oil burners are running and to stop automatically when the burners shut down. The booster equipment is most effective in increasing output at low operating temperatures. According to tests, efficiencies may be advanced from 60 per cent for gravity to 70 per cent with boosters at low operating temperatures, but at high operating temperatures gravity and booster efficiencies are almost identical[9].

[9]University of Illinois *Eng Exp Sta. Bulletin No* 141, p 79, and No. 246

PROBLEMS IN PRACTICE

1 ● What may prohibit the use of a gravity warm air system in a large house having several exposed wings?

In a gravity warm air system, excessive vertical distances above the furnace cause little trouble in the design of the wall stacks, but excessive horizontal distances from the furnace should be carefully considered in the design of the leaders. To work effectively, a gravity warm air system should be balanced and leaders over 12 ft in length should be avoided if possible. Long leaders, if used, must be of ample size, well pitched, and well insulated. Large houses having exposed wings may require leaders much longer than 12 ft, infiltration may create severe back-drafts in the exposed wings, and the basement ceiling height may not be sufficient to allow the leaders to have a pitch of more than one inch per foot. These conditions may make the exposed wings very difficult to heat with a gravity system because of its low air head differentials.

2 ● A first story dining room has a calculated heat loss of 12,000 Btu per hour.

a. What size leader pipe should be used for 175 F register air temperature?

b. What size register?

a Leader area $= \frac{12{,}000}{111} = 108\ 1$ sq in Use leader with diameter of 12 in

b Register gross area $= \frac{108}{0\ 7} = 154$ sq in Use 12 in. by 14 in register.

3 ● A third-story bedroom has a calculated heat loss of 12,000 Btu per hour.
a. What size leader pipe should be used for a 175 F register air temperature?
b. What size stack?
c. What size register?

a. Leader area $= \frac{12{,}000}{200} = 60$ sq in. Use leader with diameter of 9 in

b. Stack area $= 0.7 \times 60 = 42$ sq in Use stack 3½ in by 12 in

c. Register gross area $= \frac{60}{0.7} = 85.7$ sq in. Use register 8 in. by 12 in.

4 ● The calculated heat loss of a house is 130,000 Btu per hour. Find the grate area required for the furnace under the following conditions:

Heating value of coal = 12,790 Btu per lb.
Furnace efficiency = 55 per cent.
Combustion rate = 7.5 lb per sq ft per hr.
Ratio of heating surface to grate area of furnace = 20 to 1.
Register temperature = 175 F.
Loss between furnace and registers = 25 per cent.

See Equations 9 and 10:
Grate area $= 0.004205 \times 130{,}000 = 547$ sq in.
Grate diameter = 26.3 in.
Use grate with diameter of 26 in.

5 ● If in Question 4 the conditions were the same except that the ratio of heating surface to grate area of furnace was 24 to 1, what size grate would be required for the furnace?

Grate area $= \frac{0.004205 \times 130{,}000}{1 + 0.02\ (24\text{-}20)} = \frac{547}{1.08} = 506$ sq in.
Grate diameter = 25 4 in
Select grate with diameter of 25 in

6 ● Name the items involved in the design of a furnace heating system.

a. Heat loss from each room, Btu.
b. Area and dimensions of warm-air pipes in basement, inches.
c. Area and dimensions of vertical pipes, inches.
d. Free and gross area and dimensions of warm-air registers, inches.
e. Area and dimensions of recirculating or outside air ducts, inches.
f. Free and gross area and dimensions of recirculating registers, inches.
g. Size of furnace necessary to supply the warm air to overcome the heat loss.
h. Area and dimension of chimney and smoke pipe, inches.

7 ● Discuss the design features of recirculating ducts.

a. Their area should be equal to or greater than that of the supply ducts.
b. They should be streamlined, and have a minimum number of turns.
c. All runs should be as short as possible.
d Account should be taken of all cold walls and window areas in determining sizes and positions of return air inlets.

e. The return line should be pitched downward toward the furnace It should be designed to minimize friction

f. The top of the shoe or boot should never be above the grate level

8 ● Discuss the use of a booster fan. What effect has a booster fan at low operating temperatures? At high ones?

A booster fan is useful in accelerating the air flow past the surface of a low temperature furnace, where only a small weight differential in the air is created, and in unbalancing a gravity system so flow is established The first use involves the entire plant, and increases efficiency about 10 per cent with low temperature operation; the second involves only the leaders in which air flow is accelerated. At high operating temperatures the difference in weight between warm outgoing air and cool incoming air is great enough to make a booster unnecessary with ordinary gravity systems

9 ● Is it desirable to use high side wall locations for warm air registers in gravity circulating systems?

High side wall locations are not recommended on account of the tendency for stratification of the air in the room resulting in high temperatures at the ceiling.

Chapter 24

MECHANICAL WARM AIR FURNACE SYSTEMS

Furnaces, Fans and Motors, Sound Control, Air Washers and Filters, Air Distribution Design, Automatic Controls, Design of Heating System, Selecting the Furnace, Selecting the Fan, Heavy Duty Fan Furnaces, Humidification, Cooling Methods, Cooling System Design

MECHANICAL warm air or fan furnace heating systems[1], which are a special type of central fan systems, are particularly adapted to residences, small office buildings, stores, banks, schools, and churches. Circulation of air is effected by motor-driven fans instead of by the difference in weight between the heated air leaving the top of the casing and the cooled air entering its bottom, as in gravity systems described in Chapter 23. The advantages of mechanical systems, as compared with gravity systems are:

1. The furnace can be installed in a corner of the basement, leaving more basement room available for other purposes.

2. Basement distribution piping can be made smaller and can be so installed as to give full head room in all parts of the average basement, or be completely concealed from view except in the furnace room.

3 Circulation of air is positive, and in a properly designed system can be balanced in such a way as to give a greater uniformity of temperature distribution.

4. Humidity control is more readily attained.

5. The air may be cleaned by air washers or filters, or both.

6. The fan and duct equipment may be utilized for a complete cooling and dehumidifying system for summer, using either ice, mechanical refrigeration, or low temperature water for cooling and dehumidifying, or adsorbers for dehumidifying.

7. The use of the fan increases the volume of air which can be handled, thereby increasing the rate of heat extraction from a given amount of heating surface and insuring sufficient air volume to obtain proper distribution in a large room.

Much of the equipment used in central fan systems is the subject matter of other chapters. It is the purpose of this chapter to discuss the co-ordinated design and to deal in detail only with problems not covered elsewhere which refer particularly to the whole problem of fan warm air furnace heating and air conditioning.

---

[1]See University of Illinois *Engineering Experiment Station* Bulletin No. 266 by A P. Kratz and S. Konzo for details of tests conducted in Warm Air Research Residence.

FURNACES

Furnaces for mechanical warm air systems may be made of cast-iron, steel, or alloy. Cast-iron furnaces are usually made in sections and must be assembled and cemented or bolted together on the job. Steel furnaces are made with welded or riveted seams. The proper design of the furnace depends largely on the kind of fuel to be burned. Accordingly, various manufacturers are making special units for coal, oil and gas. Each type of fuel requires a distinct type of furnace for highest efficiency and economy, substantially as follows:

1. Coal Burning:
   a. *Bituminous*—Large combustion space with easily accessible secondary radiator or flue travel
   b. *Anthracite or coke*—Large fire box capacity and liberal secondary heating surfaces.
2. Oil Burning:
   a. Liberal combustion space.
   b. Long fire travel and extensive heating surface.
3. Gas Burning:
   a. Extensive heating surface.
   b. Close contact between flame and heating surface.

A combustion rate of from 5 to 8 lb of coal per square foot of grate per hour is recommended for residential heaters. A higher combustion rate is permissible with larger furnaces for buildings other than residences, depending upon the ratio of grate surface to heating surface, firing period, and available draft.

Where oil fuel is used, care must be exercised in selecting the proper size and type of burner for the particular size and type of furnace used. It is recommended that the system be designed for blow-through installations, so that the furnace shall be under external pressure in order to minimize the possibility of leakage of the products of combustion into the air circulating system.

In residential furnaces for coal burning, the ratio of heating surface to grate area will average about 20 to 1; in commercial sizes it may run as high as 50 to 1, depending on fuel and draft. Furnaces may be installed singly, each furnace with its own fan, or in batteries of any number of furnaces, using one or more fans.

Furnace Casings

Casings are usually constructed of galvanized iron, 26-gage or heavier, but they may also be constructed of brick. Galvanized iron casings should be lined with black iron liners, extending from the grate level to the top of the furnace and spaced from 1 in. to 1½ in. from the outer casing. Casings for commercial or heavy duty furnaces, if built of galvanized iron, should be insulated with fireproof insulating material at least 2 in. thick. It is generally believed that either brick or sheet metal casing should be equipped with baffles to secure impingement of the air to be heated against the heating surfaces. Brick furnace casings should be supplied with access doors for inspection.

For furnace casings sized for gravity flow of air, where a fan is to be

used, many manufacturers recommend the use of special baffles to restrict the free area within the casing and to force impingement of the air against the heating surfaces. The method of making these baffles for furnaces with top horse-shoe radiators and for furnaces with back crescent radiators is illustrated in Fig. 1.

Either square or round casings may be used. Where square casings are

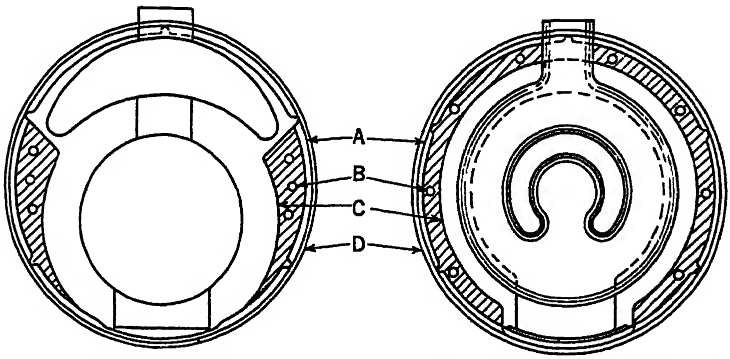


Fig. 1. Usual Method of Baffling Round Casings for Fan Furnace Work

A. Liner, 1 in from casing *B* Hole to vent baffle. *C* Baffle closed top and bottom *D.* Outer casing

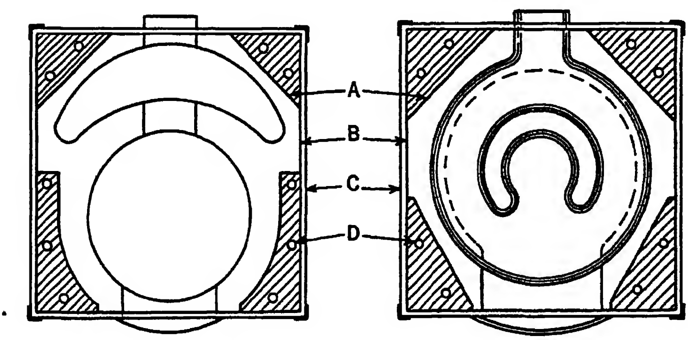


Fig. 2. Method of Baffling Square Furnace Casing for Fan Furnace Work

A Baffle, closed top and bottom *B* Liner, 1 in from casing. *C* Outer casing *D* Hole to vent baffle

used, the corners must be baffled to reduce the net free area and to force impingement of air against the heating surfaces. Fig. 2 shows the usual method of baffling square furnace casings for fan-furnace work.

The hood or bonnet of the casing above the furnace should be as high as basement conditions will allow, to form a plenum chamber over the top of the furnace. This tends to equalize the pressure and temperature of the air leaving the bonnet through the various openings. It is generally considered advisable to take off the warm air pipes from the side of the bonnet near the top, as this method of take-off allows the use of a higher bonnet

and thus provides a larger plenum chamber. Fig. 3 illustrates a complete residence fan furnace installation showing location of fan, furnace, filters, plenum chamber and method of take-off of warm air pipe.

FANS AND MOTORS

Centrifugal type fans are most commonly used, and these may be equipped with either backward or forward curved blades. Low tip speed is desirable for the elimination of air noise, especially where forward curved blades are used. Motors may be mounted on the fan shaft or outside of the fan with belt connection. Multi-speed motors or pulleys are desirable to provide a factor of safety and to allow for increased air circulation. For additional information on fans and motors, see Chapters 17 and 42.

SOUND CONTROL

Special attention should be given to the problem of noise elimination. The fan housing should not be directly connected with metal, either to the furnace casing or to the return air piping. It is common practice to use canvas strips in making these connections. Motors and their mountings must be carefully selected for quiet operation. Electrical conduit and water piping must not be fastened to, nor make contact with fan housing. The installation of a fan directly under a cold air grille is not recommended on account of the noise objection. See also Chapter 18.

AIR WASHERS AND FILTERS

Washers for residence systems may be provided in separate housings to be installed on the inlet or outlet side of the fan, or they may be integral with the fan construction. They operate at water pressures of from 10 to 30 lb and use two or more spray nozzles for washing and humidification. The sprays should be adjusted to completely cover the air passages.

Washers are usually controlled by solenoid valves wired in parallel with the fan motor. The water supply may, in turn, be controlled by a humidity-controlling device located in one of the living rooms, so that the washer will operate at all times when the fan is in operation, unless the relative humidity should rise beyond a desirable percentage. Washers used in connection with commercial or heavy duty plants should be a regulation type of commercial washer.

There are many satisfactory types of filters on the market. These include dry filters, viscous filters, oil filters and other types, some of which must be cleaned, some of which must be cleaned and recharged with oil, and some of which are inexpensive and may be discarded when they become dirty, and replaced with new ones.

The resistance of a filter must be considered in the design of the system since the resistance rises rapidly as the filter becomes dirty, thus impairing the heating efficiency of the furnace, in fact, endangering the life of the furnace itself. Manufacturers' ratings of filters must be carefully regarded, and ample filter area must be provided. Filters must be replaced or cleaned when dirty. See also Chapter 16.

AIR DISTRIBUTION

The conditions of comfort obtained in a room are greatly influenced by the type of register used and the locations of the supply registers and return grilles. In general it has been found that changes in the type, air velocity, and location of the supply register affect the room conditions much more than the changes in the location of the return grilles. Due to the economic considerations involved, it is common practice to locate the supply openings on the inside walls of a residence and the return openings nearest the greatest outside exposure. Many designers prefer, however, to locate the supply registers so that the warm air from the registers *blankets* a cold wall, and mixes with the cold air dropping off from the

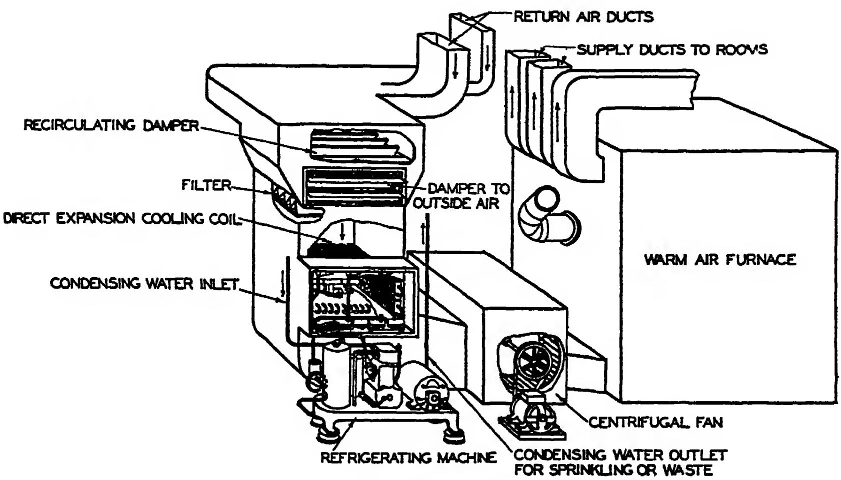


Fig 3 Complete Residence Fan Furnace Installation for Winter Heating and Summer Cooling

exposed walls. This may be accomplished either by the use of a supply register located on the exposed wall with warm air blowing into the room, or by the use of a supply register placed close to an outside wall in such a position that the warm air sweeps the cold wall surface. The ducts leading to supply registers which are located on the exposed walls should be adequately insulated to reduce the heat loss from the ducts.

Register and Grille Openings

Supply registers located in the floor are effective, but as they require frequent attention to keep them clean they should be avoided where another effective register location can be found. Tests conducted in the Warm Air Research Residence[2] have indicated that excellent results are obtainable with the use of a deflecting-diffuser type of baseboard register which throws the air downward toward the floor and diffuses the air at the same time. Unless registers located in the baseboard are well proportioned

[2]Loc Cit. Note 1.

and designed to harmonize with the trim, they may be unsightly. Better air distribution for cooling is obtained when high side wall registers are used, and this same location is satisfactory for heating when the openings are installed at least 7 ft above the floor line, providing the air velocity through the registers are greater than 600 fpm. Registers which are located in side walls above the baseboard or in the ceiling should be of an effective air-diffusing type. All registers should be equipped with dampers, and should be sealed against leakage around the borders or margins.

Velocities through registers may be reduced by the use of registers larger than the connecting pipes. Some suggestions for equalizing velocities over the face area of the register by means of diffusers are illustrated in Fig. 4. Merely to use a larger register may not result in materially reduced velocities unless diffusers are used.

Dampers

Suitable dampers are essential to any trunk or individual duct system, as it is virtually impossible to so lay out a system that it will be absolutely in balance without the use of dampers. Special care must be used in the

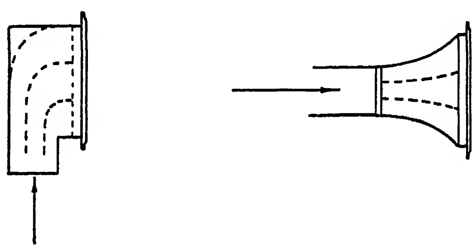


FIG. 4. DIFFUSERS IN TRANSITION FITTINGS TO EQUALIZE VELOCITIES THROUGH REGISTER FACES

design of any system to avoid turbulence and to minimize resistance. Sharp elbows, angles, and offsets should be avoided. See Figs. 1 and 2, Chapter 20

Three types of dampers are commonly used in trunk and individual duct systems. *Volume dampers* are used to completely cut off or reduce the flow through pipes. (See *A* and *B*, Fig. 5.) *Splitter dampers* are used where a branch is taken off from a main trunk. (See *C*, Fig. 5.) *Squeeze dampers* are used for adjusting the volume of air flow and resistance through a given duct. (See *D*, Fig. 5.) It is essential that a damper be provided for each main or duct branch. A positive locking device should be used with each type of damper.

Ducts

The ducts may be either round or rectangular. Rectangular ducts should be as nearly square as possible; the width should not be greater than four times the breadth. The radii of elbows should be not less than one and one-half times the pipe diameter for round pipes, or the equivalent round pipe size in the case of rectangular ducts.

AUTOMATIC CONTROLS

Air stratification, high bonnet temperatures, excessive flue gas temperatures, and heat overrun or lag in the system can be largely eliminated through proper care in the planning and installation of the control system.[3] The essential requirements of the control are:

1. To keep the fire burning when using solid fuel regardless of the weather.

2. To avoid excessive bonnet temperatures with resultant radiant heat losses into the basement.

3. To avoid the overheating of certain rooms through gravity action during off periods of blower operation.

4. To have a sufficient supply of heat available at all times to avoid lag when the room thermostat calls for heat.

5. To prevent cold air delivery when heat supply is insufficient.

6. To avoid heat loss through the chimney by keeping stack temperatures low.

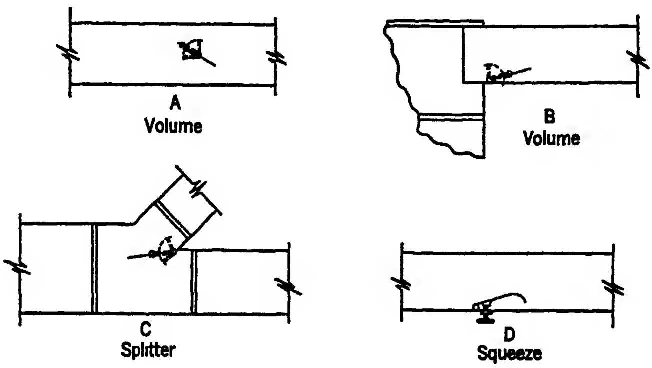


FIG. 5. THREE TYPES OF DAMPERS COMMONLY USED FOR TRUNK AND INDIVIDUAL DUCT SYSTEMS

7. To provide quick response to the thermostat, with protection against overrun.

8. To provide for humidity control.

9. To provide a means of summer control of cooling.

10. To protect against fire hazards.

The following controls are desirable:

1. A *thermostat* located at a point where maximum fluctuation in temperature can be expected, in order to secure frequent operation of fans, drafts, and burners. This location would be near an outside wall but not upon it, in a sun room, or in a room with some unusual exposure. The thermostat, of course, should not be located where it will be affected by direct radiant heat from the sun or from a fireplace, or by direct heat from any warm air duct or register.

2. A *furnacestat* located in the bonnet to permit blower operation only between the temperatures of 100 F and 150 F. In certain extreme cases it may be necessary, or weather conditions may make it advisable, to adjust the high limit to a higher temperature than that given. Another location sometimes used for the furnacestat is in the main duct near the frame opening from the bonnet.

[3]Automatic Controls for Forced-Air Heating Systems, by S. Konzo and A. F. Hubbard (A.S.H.V.E. TRANSACTIONS, Vol. 40, 1934).

3. A *protective limit control* located in the bonnet to shut down the system independently of the thermostat if the bonnet temperature exceeds 225 F.

4 On oil and gas burner installations, a control is usually included which will shut down the system if the fire goes out or if there is a failure of the ignition system.

5 A *humidistat* to regulate the moisture supplied to the rooms

6. On automatic stoker installations, a control is usually included which will start the operation regardless of thermostat settings whenever the bonnet temperature indicates that the fire is dying

METHOD OF DESIGNING FORCED-AIR HEATING SYSTEMS

1. Determine heat loss from each room in Btu per hour (See Chapter 7).

2. Locate warm air registers and return registers on plans of house, beginning with the upper story rooms

3. Sketch in duct layout to connect all registers and grilles with the central unit

4. Determine equivalent length of duct for each register, allowing 10 diameters of straight pipe as equivalent to each 90 deg elbow having an inner radius not less than the diameter of the round pipe or the depth of the rectangular pipe

5. Select a value for temperature of the air at the furnace bonnet It is customary to use some value lying between 150 to 165 F Use lower value if larger number of air recirculations is desired.

6. Determine approximate value of temperature reduction in each duct caused by heat loss from the ducts. A value of from 0 3 to 0 6 F per foot of duct has been obtained from tests conducted in the Research Residence installation for uninsulated duct lengths up to approximately 60 ft.

7 Subtract this temperature reduction from the assumed bonnet air temperature to obtain an approximate value of the register air temperature for each register.

8 Determine the required air volume for each room from the following equation, or from the values listed in Table 1:

$$Q = \frac{H}{60 \times 0.24 \times d\,(t_r - 65)} \qquad (1)$$

where

Q = required air volume, cubic feet per minute.
H = heat loss of room, Btu per hour.
d = density of air at register temperature, pounds per cubic foot.
t_r = register temperature, degrees Fahrenheit
0.24 = specific heat of air.
65 = return air temperature.

For any given register temperature the solution of this equation simplifies to the following form:

$$Q = H \times \text{Factor} \qquad (2)$$

in which the values of the Factor may be obtained from Table 1.

9. Determine register size from the air volume delivered to each room by the following formula:

$$\text{Free area of register, square feet} = \frac{Q}{V} \qquad (3)$$

$$\text{Gross area of register, square feet} = \frac{\text{Free Area}}{R} \qquad (4)$$

where

Q = required air volume, cubic feet per minute.
V = velocity at register face, feet per minute
R = ratio of free area to gross area of register.

TABLE 1 FACTORS CORRESPONDING TO REGISTER TEMPERATURE FOR EQUATION 2

REGISTER TEMPERATURE	FACTOR
110	0.02210
120	0 01840
130	0 01585
140	0 01397
150	0.01253
160	0 01140
170	0.01049

Allowable register velocities to be used in Equation 3 are approximately as follows

Baseboard, non-deflecting type, maximum = 300 fpm
Baseboard, deflecting toward floor, maximum = 500 fpm.
Baseboard, deflecting and diffusing = up to 800 fpm
High sidewall = not less than 600 fpm.

10 Duct systems for forced-air installations may consist of either trunk systems or individual duct systems

Trunk Systems. Determine duct sizes and friction losses as outlined in Chapter 20, except that for residence applications the velocities in the main duct and in the various parts of the system should approximate the values recommended in Table 2.

Individual Duct Systems An individual duct system is one having separate ducts extending from the heating unit to each register In designing such a system select first the duct having the greatest equivalent length. Select a reasonable velocity using Table 2 as a guide From friction chart on p. 366 determine unit friction loss per 100 ft of run, and from this the total friction loss in the duct selected If this total friction loss exceeds a reasonable value a lower velocity should be used.

The remaining ducts are proportioned so that the total pressure in each duct is the same as that calculated for the longest duct. The added resistance necessary in the shorter ducts is accomplished by increasing the velocity in these ducts. No duct should be less than 6 in in diameter, nor should the velocity in any duct exceed approximately 1200 fpm. The final adjustment in a duct system may be made by employing dampers.

Instead of proportioning the ducts as outlined in the preceding paragraph it is more usual in practice to proportion all the ducts so that they have the same velocity as that used in the longest duct and to balance the system by employing dampers in the shorter ducts

Return duct systems are designed making use of the same principles as those used in the design of supply duct systems In this case the design may be based on the volume of air corresponding to the density of air existing in the return ducts, or in order to provide a factor for air leakage, it may be based on the same volume as used for the supply ducts.

TABLE 2. RECOMMENDED VELOCITIES THROUGH DUCTS AND REGISTERS

DESCRIPTION	LOW VELOCITY SYSTEM (FPM)	MEDIUM VELOCITY SYSTEM (FPM)	HIGH VELOCITY SYSTEM (FPM)
Main ducts	500	750	1000
Branch ducts	450	600	750
Wall stacks	350	500	600
Baseboard registers (max)	300	350	400
Wall registers above 5 ft (min)	500	550	600

11 Determine frictional resistance in

a Supply side of system as outlined in item 10
b Return side of system as outlined in item 10
c. Furnace units, casing or hood, which is usually considered as equivalent to 0 03 to 0 10 in of water
d. Accessories such as washers or air filters, from manufacturer's data.
e. Inlet and outlet registers and grilles, from manufacturer's data
f. Other accessory equipment such as cooling coils, from manufacturer's data

Choose a fan which, according to its manufacturer's rating, is capable of delivering a volume of air, expressed in cubic feet per minute, against a frictional resistance, expressed in inches of water, computed by adding together the items listed in the preceding discussion. In practice it is recommended that liberal allowances should be made so that the fan will be capable of delivering air against pressures that may not have been foreseen during the design of the duct system.

12. Select a furnace capable of delivering heat at the register outlets equal to the total heat loss of the structure to be heated

The following formula may be used for coal burning furnaces:

$$G = \frac{H}{f \times p \times E_1 \times E_2 \, [1 + 0\,02\,(R - 20)]} \tag{5}$$

where

G = required grate area, square feet
H = total heat loss from building, Btu per hour
f = calorific value of coal, Btu per pound
p = combustion rate in pounds of fuel per square foot of grate per hour.
E_1 = furnace efficiency based on heat available at bonnet.
E_2 = efficiency of transmission based on ratio of heat delivered at register to heat available at bonnet
R = ratio of heating surface to grate area.

In practice it is customary to use the following constants

f = 12,000 (for specific values, see Table 1, Chapter 27).
p = 7.5 lb.
E_1 = 0.65 lower efficiency must be used with highly volatile solid fuel.
E_2 = 0.85.

The foregoing procedure for determining the size of the furnace to be used applies to continuously heated buildings.

13. Although intermittently heated buildings usually have their heat losses computed according to the standard rules for determining such losses, these rules do not take into account the heat which will be absorbed by the cold material of the building after the air is raised in temperature. This heat absorption must be added to the normal heat loss of the building to determine the load which the heating plant must carry through the warming-up process. It is customary to increase the normal heat loss figure by from 50 to 150 per cent depending upon the heat capacity of the construction material, the higher percentage applying to materials of high heat capacity such as concrete and brick. Fan furnace systems are well adapted for heating intermittently heated buildings as these systems do not require the warming of intermediate piping, radiators, or convectors, the generation of steam, or the heating of hot water.

14. Follow the same methods for an oil furnace as for coal where a conversion unit is to be used, making sure that the ratio of heating surface to grate area exceeds 20 to 1. If it does not, a size larger furnace should be selected Use the manufacturers' Btu ratings of furnaces designed for exclusive use with oil, and select a burner with liberal excess capacity.

15 The selection of the proper size gas furnace for a constantly heated building can be easily made by using the following *American Gas Association* formula

$$R = \frac{H}{0\ 9} \tag{6}$$

where

H = total heat loss from building, Btu per hour.
R = official $A.\ G\ A$ output rating of the furnace, Btu per hour

In the case of converted warm air furnaces a slightly different procedure is necessary, as the Btu input to the conversion burner must be selected rather than the furnace output The proper sizing may be done by means of the following formula:

$$I = 1\ 59\ H \tag{7}$$

where

I = Btu per hour input

The factor 1 59 is the multiplier necessary to care for a 10 per cent heat loss in the distributing ducts and an efficiency of 70 per cent in the conversion burner.

16 Specify location and type of all dampers in both supply air and return air sides of system. Specify controls including location of all thermostats. Arrange for proper control of humidifying equipment.

HEAVY DUTY FAN FURNACES

Fan furnaces for large commercial and industrial buildings are available in sizes ranging from 400,000 to 3,000,000 Btu per hour per unit. Heavy duty heaters may be arranged in combinations of one or more units in a battery. A few possible arrangements are shown in Figs. 6 to 9 inclusive.

Most manufacturers of heavy duty furnaces rate their furnaces in Btu per hour and also in the number of square feet of heating surface. Conservative practice indicates that at no time in the heating-up period should the furnace surface be required to emit more than an average of 3500 Btu per square foot. A higher rate of heat emission tends to increase the heat loss up the chimney, and raise fuel consumption, to shorten the life of the furnace, and to overheat the air. The ratio of heating surface to grate area on furnaces for this type of work should never be less than 30 to 1 and as indicated previously may run as high as 50 to 1.

Control of temperature is secured through (1) controlling the quantity of heated air entering the room, (2) using mixing dampers, or (3) regulating the fuel supply.

The design of heavy duty fan furnace heating systems is in many respects similar to that of the central fan heating systems described in Chapter 9. Ducts are designed by the method outlined in Chapter 20.

HUMIDIFICATION

Mechanical warm air systems offer a means of proportioning and distributing moisture-bearing air; consequently, during the winter months humidifiers may be employed to deliver water vapor to the fan-driven air stream in proper amounts to produce a more humid atmosphere, with increased comfort for people and increased life for household furnishings. Temperatures and relative humidities should be governed within the

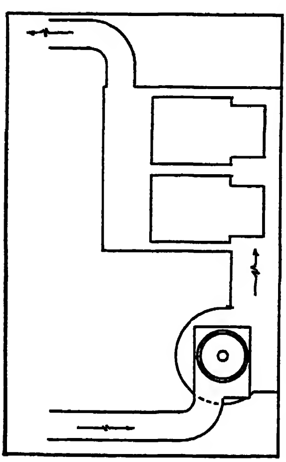


FIG. 6. HEATER ARRANGED FOR COMPLETE RECIRCULATION OF AIR

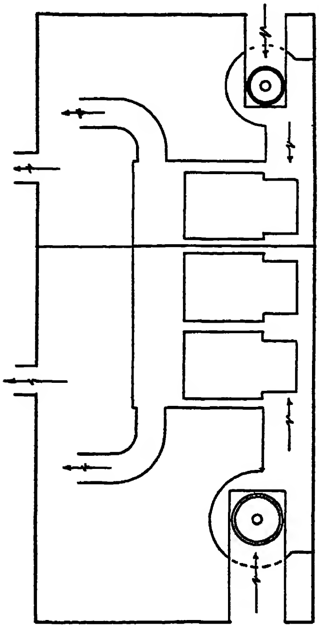


FIG. 7. TWO BATTERIES OF HEATERS AND FANS FOR INDEPENDENT SERVICE USING OUTSIDE AIR AND EXHAUSTING TO ATMOSPHERE

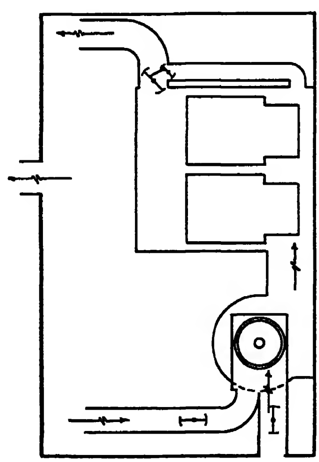


FIG. 8. HEATER ARRANGED FOR PARTIAL RECIRCULATION, ALSO SHOWING MIXING DAMPER FROM WARM AIR AND TEMPERED AIR CHAMBERS, AND PARTIAL EXHAUST TO ATMOSPHERE

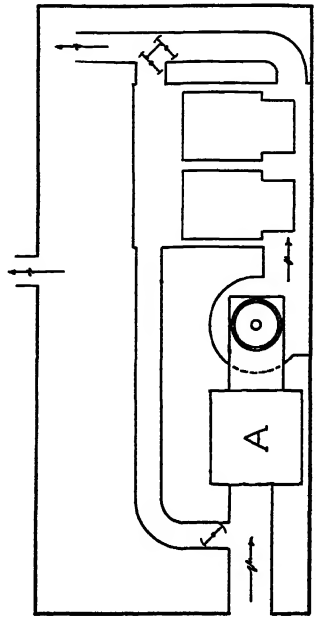


FIG. 9. HEATER ARRANGED FOR USE OF AIR WASHER OR FILTER (A) WITH HEATED AIR TO MIX WITH OUTSIDE AIR FOR TEMPERING, SHOWING MIXING DAMPER FROM WARM AIR AND TEMPERED AIR AND EXHAUST TO ATMOSPHERE

limits of the generally accepted standards. See Chapters 3 and 12 for more detailed information on this point.

In earlier types of furnaces, water evaporating pans were usually placed in the cool portions of the air stream, but modern types usually locate them in air which has been heated by contact with the heating surfaces. To change water into vapor capable of being carried in an air stream as part of the mixture, about 1000 Btu per pound are required. Without the addition of this heat, termed the latent heat of evaporation, water injected into the air will be carried along in the form of tiny globules until it falls out of the stream or is deposited upon some surface.

Furthermore, when dry air is in contact with water for a sufficient length of time without the presence of a sizable body of water or a source other than air from which this latent heat of evaporation can be taken, such heat is supplied from the air. There is, therefore, a trend in present practice toward heating the water in addition to heating the air. Equipment for doing this may make use of sprays, or it may take the form of water circulating coils placed within the combustion chamber and connected by pipes to the humidifier pans where a constant water level is maintained by some separate float device. (See Chapter 12.)

Residence Requirements

The principles underlying humidity requirements and limitations for residences are summarized in *University of Illinois Bulletin No.* 230[4], as follows:

1. Optimum comfort is the most tangible criterion for determining the air conditions within a residence.

2. An effective temperature of 65 deg[5] represents the optimum comfort for the majority of people. Under the conditions in the average residence a dry-bulb temperature of 69 5 F with relative humidity of 40 per cent is the most practical for the attainment of 65-deg effective temperature.

3. Evaporation requirements to maintain a relative humidity of 40 per cent in zero weather depend on the amount of air inleakage to the average residence, and vary from practically nothing to 24 gal of water per 24 hours.

4. Relative humidity of 40 per cent indoors cannot be maintained in rigorous climates without excessive condensation on the windows unless tight-fitting storm sash or the equivalent is installed.

5. The problems of humidity requirements and limitations cannot be separated from considerations of good building construction, and the latter should receive serious attention in the installation of humidifying apparatus.

The following conclusions were drawn from the experimental results reported in the aforementioned bulletin:

1. None of the types of gravity warm air furnace water pans tested proved adequate to evaporate sufficient water to maintain 40 per cent relative humidity in the Research Residence except only in moderately cold weather.

2. The water pans used in the radiator shields tested did not prove adequate to maintain 40 per cent relative humidity in a residence similar to the Research Residence when the outdoor temperature approximated zero degrees Fahrenheit.

[4]See Humidification for Residences, by A. P Kratz (*University of Illinois, Bulletin No* 230)

[5]66 deg is the optimum winter effective temperature recommended by the A.S.H.V.E Committee on Ventilation Standards. See Chapter 3

COOLING METHODS

A slight cooling effect may be obtained under certain conditions by the use of basement air. A more positive cooling effect may be obtained through air washers where the temperature of the water is sufficiently low (55 F or lower), and where a sufficient volume of water can be provided. Unless the temperature of the leaving water is below the dew point temperature of the indoor air at the time the washer is started, both the relative and absolute humidities will be somewhat increased.

Coils of copper finned tubing through which cold water is pumped are available for cooling. They require less space than air washers and have the advantage that no moisture is added to the air when the temperature of the water rises above the dew point. Ample coil surface is necessary with this type of cooling.

It is thoroughly feasible to use ice or mechanical refrigeration in connection with the fan and duct system for the heating installation, and to cool the building by this method, provided the building is reasonably well constructed and insulated. Windows and doors should be tight, and awnings should be supplied on the sunny side of the building. See also Chapters 10 and 11.

Results at Research Residence

The following conclusions may be drawn from the studies thus far completed in the Research Residence, subject to the limitations of the conditions under which the tests were run[6].

1. An uninsulated building of ordinary residential type may require the equivalent of three tons of ice in 24 hr on days when the maximum outdoor temperature reaches 100 F if an effective temperature of approximately 72 deg is maintained indoors.

2. The use of awnings at all windows in east, south, and west exposures may result in savings of from 20 to 30 per cent in the required cooling load

3. The cooling load per degree difference in temperature is not constant but increases as the outdoor temperature increases.

4. The heat lag of the building complicates the estimation of the cooling load under any specified conditions and makes such estimates, based on the usual methods of computation, of doubtful value.

5. The seasonal cooling requirements are extremely variable from year to year, and the ratio between the degree-hours of any two seasons occurring within a 10 year period may be as high as 7.5 to 1. Hence an average value of the degree-hours cooling per season is comparatively meaningless

6. The duct system in a forced-air heating installation can be successfully converted to a system for conveying cool air for the purpose of cooling the structure. No condensation of moisture was observed when the duct temperatures were not less than 65 F.

7. Cooling by means of water at a temperature of 60 F is not satisfactory unless an indoor temperature of less than 80 F is maintained

8. In the selection of cooling coils, the frictional resistance of the coil to flow of air must be given careful consideration.

9. Cooling the structure by introducing large quantities of air from outdoors at night tended to reduce the amount of cooling required on the following day and was a practical means of providing more comfortable conditions in those homes where cooling systems were not available.

---

[6]Study of Summer Cooling in the Research Residence at the University of Illinois, by A P Kratz and S. Konzo (A S H.V E. Transactions, Vol. 39, 1933); Study of Summer Cooling in the Research Residence for the Summer of 1933, by A. P Kratz and S Konzo (A S H V E. Transactions, Vol. 40, 1934).

METHOD OF DESIGNING COOLING SYSTEM

The general procedure for the design of a cooling system in a forced-air installation is as follows:

1. Calculate heat gain for each room or space to be conditioned See Chapters 5 and 8) Allowance for addition of outside air must be included in this calculation

2. Select a temperature of air leaving supply inlets In Research Residence tests[7] a value of from 65 to 70 F was found satisfactory.

3 Determine indoor conditions to be maintained In Research Residence 80 F dry bulb and 45 per cent relative humidity was found satisfactory

4. Determine the quantity of air to be introduced into each room 'See Chapter 10, Equations 1 and 2)

5 Estimate heat loss in duct system between cooling unit and supply registers.

6. Calculate the heat to be removed by the cooling unit, in the form of sensible heat and latent heat.

7. Determine size of ducts in duct system and size of registers, as explained in this chapter under the heading of Method of Designing Forced-Air Heating Systems

8. Determine pressure loss in duct system and select fan as also explained in the same section

9 Select cooling unit from manufacturer's data Specify temperature and pressure of available cooling water, voltage and characteristics of electrical supply, and method of control of apparatus

10 Select cooling coils from manufacturer's data to take care of latent heat load and to give required drop in air temperature with the weight of air flowing. See Chapter 10 on section Surface Cooling Problems.

11 If system is to be used for both winter heating and summer cooling, duct sizes must be checked to insure that velocities and friction losses are reasonable for both conditions of operation Adjustable dampers will be necessary to make changes in air distribution for the two seasons Provision must also be made for changing fan speeds for summer and winter operation

[7]Loc. Cit Note 6.

PROBLEMS IN PRACTICE

1 ● A residence furnace, having a ratio of heating surface to grate area equal to 20 to 1, is to be selected to heat a house which has a computed load of 225,000 Btu per hour. If coal having a calorific value of 12,000 Btu per pound is to be burned, if the furnace will burn 7.5 lb of coal per square foot of grate per hour, and if the furnace efficiency is 65 per cent, determine the square feet of grate area necessary in the furnace to be selected.

Substituting in Equation 5:

$$G = \frac{225{,}000}{12{,}000 \times 7.5 \times 0\,65 \times 0.85} = 4.53 \text{ sq ft of grate area.}$$

A furnace having at least 4 5 sq ft of grate area should therefore be selected

2 ● Why should secondary surface be designed for easy cleaning?

If the combustion is not perfect, soot is formed immediately above the fire and is apt to form a deposit on the secondary surface from which it should be removed. If the secondary surface is so designed that there are horizontal passages, fine gray ash will settle out in these to form an insulation between the hot gases of combustion and the metal of the furnace; consequently, these should be readily cleaned If the passages are vertical they are largely self-cleaning of ash, but provision should be made for easy and thorough cleaning of the collection chamber below them.

3 ● Why is baffling inside the casing necessary on fan systems?

Because the movement of air is independent of its temperature, air must be guided by baffles of one form or another to bring it in contact with the hot surfaces so it will not pass through the casing unheated. On the other hand, if the air is held against a hot surface too long it might become overheated, for the average register temperature on a fan system should not exceed 120 F.

4 ● What practical points should be observed in designing a fan system in order to eliminate noise?

a. Use a large fan so it can be run at slow speed.

b. Set the fan and motor on a solid foundation

c. Insulate the fan and motor from the foundation with rubber, cork, or other springy material according to the principles given in Chapter 18, provided, of course, that such insulation is of value.

d. See that the air velocity is not too high in the ducts. Properly designed splitters in the elbows will avoid high velocities at the turns in cases where the velocity through the ducts themselves is not too high.

e. Use canvas connections between the ducts and any running equipment.

f. Be sure the ducts have a relatively smooth interior and are rigid.

5 ● Why do furnaces designed to burn bituminous coal, oil, or gas require larger combustion spaces than those designed for anthracite?

Anthracite burns largely as fixed carbon whereas gas and oil burn as gases, and as much as 50 per cent of bituminous coal burns as a gas. Ample space must be provided for the intimate mixture of these gases with the oxygen of the air to secure proper combustion.

6 ● A furnace has the following dimensions: Grate diameter, 24 in.; casing diameter for gravity air flow, 56 in.; combustion chamber diameter, 30 in. What is the unobstructed area required for passage of air across the heating surface when a motor-driven blower, operating at an outlet velocity of 1200 fpm, delivers 1600 cfm into the casing near its bottom?

For residence applications using small blowers, an air outlet velocity of about one third of the blower outlet velocity is considered good practice

$$\text{Air-pass velocity} = \frac{1200}{3} = 400 \text{ fpm}$$

$$\text{Air-pass area} = \frac{1600}{4} = 4 \text{ sq ft} = 576 \text{ sq in}$$

7 ● In Question 6 what would be the gap between the chamber and the baffle when the chamber is centered in the casing?

Area of combustion chamber (30-in. diam)	706 9 sq in.
Area of air pass	576 0 sq in.
Total area	1282 9 sq in.

The diameter of a circle with an area of 1282.9 sq in. is 40.4 in. One half of the difference between the diameters is the amount of gap.

$$\text{Gap} = \frac{40.4 - 30.0}{2} = 5.2 \text{ in.} = \text{approximately } 5\tfrac{1}{4} \text{ in}$$

Chapter 25

BOILERS

Cast-Iron Boilers, Steel Boilers, Special Heating Boilers, Gas-Fired Boilers, Hot Water Supply Boilers, Furnace Design, Heating Surface, Testing and Rating Codes, Output, Efficiency, Selection of Boilers, Connections and Fittings, Erection, Operation and Maintenance, Boiler Insulation

STEAM and hot water boilers for low pressure heating work are built in a wide variety of types, many of which are illustrated in the *Catalog Data Section*, and are classified as (1) cast-iron sectional, (2) steel fire tube, (3) steel water tube, and (4) special.

CAST-IRON BOILERS

Cast-iron boilers may be of round pattern with circular grate and horizontal pancake sections joined by push nipples and tie rods, or of rectangular pattern with vertical sections. The latter type many be either of outside header construction where each section is independent of the other and the water and steam connections are made externally through these headers, or assembled with push nipples and tie rods, in which case the water and steam connections are internal.

Cast-iron boilers usually are shipped knocked down to facilitate handling at the place of installation where assembly is made. One of the chief advantages of cast-iron boilers is that the separate sections can be taken into or out of basements and other places more or less inaccessible after the building is constructed. This feature is of importance in making repairs to or replacing a damaged or worn-out boiler and should be given consideration in the original selection. Sufficient space should be provided in the boiler room for assembling the boiler and for disassembling it conveniently if repairs are needed. With the outside header type of boiler a damaged section in the middle of the boiler can be removed without disturbing the other sections so side clearance should be provided.

Capacities of cast-iron boilers range from that required for small residences up to about 18,000 sq ft of steam radiation. For larger loads, cast-iron boilers must be installed in multiple, or a steel boiler must be used. In most cases cast-iron boilers are limited to working pressures of 15 lb for steam and 30 lb for water. Special types are built for hot water supply which will withstand higher local water pressures.

STEEL BOILERS

Two general classifications may be applied to steel boilers: *first*, with regard to the relative position of water and hot gases, distinguished as fire tube or water tube; *second*, with regard to arrangement of furnace and

flues, as (1) horizontal return tubular (HRT) boilers, (2) portable (self-contained) firebox boilers with either water or fire tubes, and (3) water tube boilers of the power type.

Fire tube boilers are constructed so that the water available to produce steam is contained in comparatively large bodies distributed outside of the boiler tubes, the hot gases passing within the tubes. In *water tube* boilers, the water is circulated within the boiler tubes, heat being applied externally to them.

The *HRT boiler* is the oldest type and consists of a horizontal cylindrical shell with fire tubes, enclosed in brickwork to form the furnace and

TABLE 1. PRACTICAL COMBUSTION RATES FOR SMALL COAL-FIRED HEATING BOILERS OPERATING ON NATURAL DRAFT OF FROM 1/8 IN. TO 1/2 IN WATER[a]

KIND OF COAL	SQ FT GRATE	LB OF COAL PER SQ FT GRATE PER HOUR
No 1 Buckwheat Anthracite	Up to 4	3
	5 to 9	3½
	10 to 14	4
	15 to 19	4½
	20 to 25	5
Anthracite Pea	Up to 9	5
	10 to 19	5½
	20 to 25	6
Anthracite Nut and Larger	Up to 4	8
	5 to 9	9
	10 to 14	10
	15 to 19	11
	20 to 25	13
Bituminous	Up to 4	9.5
	5 to 14	12
	15 and above	15.5

[a]Steel boilers usually have higher combustion rates for grate areas exceeding 15 sq ft than those indicated in this table.

combustion chamber. All heating surfaces and the interior of the boiler are accessible for both cleaning and inspection. Horizontal return tubular boilers, especially the larger sizes, should be suspended from structural columns and beams independent of the brick setting. Small HRT boilers sometimes are supported by brackets resting on the brick setting.

Portable firebox boilers are the more generally used type of steel heating boilers, their outstanding characteristic being the water-jacketed firebox which eliminates virtually all brickwork. They are shipped in one piece from the factory and come to the job ready for immediate hook-up to piping. They may be of welded or riveted construction and have either water or fire tubes. Manufacturers' catalogs usually list heating surface as well as grate area. The elimination of brickwork also makes this type the most compact of steel boilers as well as the lowest in first cost.

Water tube boilers. For large heating loads water tube boilers are quite frequently used. They usually require more head room than other types of boilers but require considerably less floor space and make possible a

much higher rate of evaporation per square foot of heating surface, with proper setting, baffling and draft. Water tube boilers used for heating purposes are either completely supported, insulated and encased in steel, or else brick set, supported on structural steel columns and have the brick setting encased in an insulated steel housing to prevent air infiltration and to minimize heat losses. For large heating loads at a high rate of evaporation, such boilers should be operated at pressures above 15 lb per square inch with a pressure-reducing valve on the connection to the heating main.

SPECIAL HEATING BOILERS

A special type of boiler, known as the *magazine feed boiler*, has been developed for the burning of small sizes of anthracite and coke. These are built of both cast-iron and steel, and have a large fuel carrying capacity which results in longer firing periods than would be the case with the standard types using buckwheat sizes of coal. Special attention must be given to insure adequate draft and proper chimney sizes and connections.

Oil-burner boiler units, in which a special boiler has been designed with a furnace shaped to meet the general requirements of oil burners or are specially adapted to one particular burner have been developed by a number of manufacturers. These usually are compact units with the burner and all controls enclosed within an insulated steel jacket. Ample furnace volume is provided for efficient combustion, and the heating surfaces are proportioned for effective heat transfer. Consequently, higher efficiencies are obtainable than with the ordinary coal fired boiler converted to oil firing.

GAS-FIRED BOILERS

Gas boilers have assumed a well-defined individuality. The usual boiler is sectional in construction with a number of independent burners placed beneath the sections. In most boilers each section has its own burner. In all cases the sections are placed quite closely together, much closer than would be possible when burning a soot-forming fuel. The effort of the designer is always to break the hot gas up into thin streams, so that all particles of the heat-carrying gases can come as close as possible to the heat-absorbing surfaces. Because there is no fuel bed resistance and because the gas company supplies the motive power to draw in the air necessary for combustion (in the form of the initial gas pressure), draft losses through gas boilers are low.

HOT WATER SUPPLY BOILERS

Boilers for hot water supply are classified as direct, if the water heated passes through the boiler, and as indirect, if the water heated does not come in contact with the water or steam in the boiler.

Direct heaters are built to operate at the pressures found in city supply mains and are tested at pressures from 200 to 300 lb per square inch. The life of direct heaters depends almost entirely on the scale-making properties of the water supplied. If water temperatures are maintained below 140 F the life of the heater will be much longer than if higher temperatures are used, owing to decreased scale formation and minimized corrosion below 140 F. Direct water heaters in some cases are designed to burn refuse and garbage.

Indirect heaters generally consist of steam boilers in connection with heat exchangers of the coil or tube types which transmit the heat from the steam to the water. This type of installation has the following advantages:

1. The boiler operates at low pressure.
2 The boiler is protected from scale and corrosion
3. The scale is formed in the heat exchanger in which the parts to which the scale is attached can be cleaned or replaced. The accumulation of scale does not affect efficiency although it will affect the capacity of the heat exchanger.
4. Discoloration of water may be prevented if the water supply comes in contact with only non-ferrous metal.

Where a steam heating system is installed, the domestic hot water usually is obtained from an indirect heater placed below the water line of the boiler.

FURNACE DESIGN

Good efficiency and proper boiler performance are dependent on correct furnace design embodying sufficient volume for burning the particular fuel at hand, which requires thorough mixing of air and gases at a high temperature with a velocity low enough to permit complete combustion of all the volatiles. On account of the small amount of volatiles contained in coke, anthracite, and semi-bituminous coal, these fuels can be burned efficiently with less furnace volume than is required for bituminous coal, the combustion space being proportioned according to the amount of volatiles present.

Combustion should take place before the gases are cooled by the boiler heating surface, and the volume of the furnace must be sufficient for this purpose. The furnace temperature must be maintained sufficiently high to produce complete combustion, thus resulting in a higher CO_2 content and the absence of CO. Hydrocarbon gases ignite at temperatures varying from 1000 to 1500 F.

The question of furnace proportions, particularly in regard to mechanical stoker installations, has been given some consideration by various manufacturers' associations. Arbitrary values have been recommended for minimum dimensions. A customary rule-of-thumb method of figuring furnace volumes is to allow 1 cu ft of space for a maximum heat release of 50,000 Btu per hour. This value is equivalent to allowing approximately 1 cu ft for each developed horsepower, and it is approved by most smoke prevention organizations.

The setting height will vary with the type of stoker. In an overfeed stoker, for instance, all the volatiles must be burned in the combustion chamber and, therefore, a greater distance should be allowed than for an underfeed stoker where a considerable portion of the gas is burned while passing through the incandescent fuel bed. The design of the boiler also may affect the setting height, since in certain types the gas enters the tubes immediately after leaving the combustion chamber, while in others it passes over a bridge wall and toward the rear, thus giving a better opportunity for combustion by obtaining a longer travel before entering the tubes.

To secure suitable furnace volume, especially for mechanical stokers or oil burners, it often is necessary either to pit the stoker or oil burner, or

where water line conditions and headroom permit, to raise the boiler on a brick foundation setting.

Smokeless combustion of the more volatile bituminous coals is furthered by the use of mechanical stokers. (See Chapter 28.) Smokeless combustion in hand-fired boilers burning high volatile solid fuel is aided (1) by the use of double grates with down-draft through the upper grate, (2) by the use of a curtain section through which preheated auxiliary air is introduced over the fire toward the rear of the boiler, and (3) by the introduction of preheated air through passages at the front of the boiler. All three methods depend largely on mixing secondary air with the partially burned volatiles and causing this mixture to pass over an incandescent fuel bed, thus tending to secure more complete combustion than is possible in boilers without such provision.

HEATING SURFACE

Boiler heating surface is that portion of the surface of the heat transfer apparatus in contact with the fluid being heated on one side and the gas or refractory being cooled on the other side. Heating surface on which the fire shines is known as *direct* or radiant surface and that in contact with hot gases only, as *indirect* or convection surface. The amount of heating surface, its distribution and the temperatures on either side thereof influence the capacity of any boiler.

Direct heating surface is more valuable than indirect per square foot because it is subjected to a higher temperature and also, in the case of solid fuel, because it is in position to receive the full radiant energy of the fuel bed. The heat transfer capacity of a radiant heating surface may be as high as 6 to 8 times that of an indirect surface. This is one of the reasons why the water legs of some boilers have been extended, especially in the case of stoker firing where the extra amount of combustion chamber secured by an extension of the water legs is important. For the same reason, care should be exercised in building a refractory combustion chamber in an oil-burning boiler so as not to screen any more of this valuable surface with refractories than is necessary for good combustion.

The effectiveness of the heating surface depends on its cleanliness, its location in the boiler, and the shape of the gas passages. Investigations[1] by the U. S. Bureau of Mines show that:

1. A boiler in which the heating surface is arranged to give long gas passages of small cross-section will be more efficient than a boiler in which the gas passages are short and of larger cross-section.
2. The efficiency of a water tube boiler increases as the free area between individual tubes decreases and as the length of the gas pass increases.
3. By inserting baffles so that the heating surface is arranged in series with respect to the gas flow, the boiler efficiency will be increased.

The area of the gas passages must not be so small as to cause excessive resistance to the flow of gases where natural draft is employed.

Heat Transfer Rates

Practical rates of heat transfer in heating boilers will average about 3300 Btu per sq ft per hour for hand-fired boilers and 4000 Btu per sq ft

[1]See *U. S Bureau of Mines Bulletin No* 18, The Transmission of Heat into Steam Boilers.

per hour for mechanically fired boilers when operating at *design load*[2]. When operating at *maximum load*[2] these values will run between 5000 and 6000 Btu per sq ft per hour. Boilers operating under favorable conditions at the above heat transfer rates will give exit gas temperatures that are considered consistent with good practice.

TESTING AND RATING CODES

The Society has adopted three solid fuel testing codes, a solid fuel rating code and an oil fuel testing code. A.S.H.V.E. Standard and Short Form Heat Balance Codes for Testing Low-Pressure Steam Heating Solid Fuel Boilers—Codes 1 and 2—(Revision of June 1929)[3], are intended to provide a method for conducting and reporting tests to determine heat efficiency and performance characteristics. A.S.H.V.E. Performance Test Code for Steam Heating Solid Fuel Boilers—Code No. 3—(Edition of 1929)[3] is intended for use with A.S.H.V.E. Code for Rating Steam Heating Solid Fuel Hand-Fired Boilers[4]. The object of this test code is to specify the tests to be conducted and to provide a method for conducting and reporting tests to determine the efficiencies and performance of the boiler. The A.S.H.V.E. Standard Code for Testing Steam Heating Boilers Burning Oil Fuel[5] is intended to provide a standard method for conducting and reporting tests to determine the heating efficiency and performance characteristics when oil fuel is used with steam heating boilers.

Steel Heating Boilers Ratings

The *Steel Heating Boiler Institute* has adopted a method for the rating of low pressure boilers based on their physical characteristics and expressed in square feet of steam or water radiation or in Btu per hour as given in Table 2. The following requirements are included in this Code:

1. One square foot of steam radiation is to be considered equal to the emission of 240 Btu per hour and one square foot of water radiation is to be considered equal to emission of 150 Btu per hour.
2. The rating of a boiler expressed in square feet of steam radiation in which solid fuel hand fired is used is based on the amount equal to 14 times the heating surface of the boiler in square feet.
3. The rating of a boiler expressed in square feet of steam radiation in which solid fuel mechanically fired, or in which oil or gas is burned, is based on the amount equal to 17 times the heating surface of the boiler in square feet.
4. Heating surface is to be expressed in square feet and include those surfaces in the boiler which are exposed to the products of combustion on one side and water on the other. In measuring surfaces, the outer tube areas are to be considered. When a boiler has the water leg height increased the heating surface noted in the published ratings are not to be increased.
5. A grate area is to be considered as an area of the grate surface expressed in square feet and measured in the plane of the top surface of the grate. For double grate boilers the grate surface is to be considered as the area of the upper grate plus one-quarter of the area of the lower grate
6. The grate area of a boiler for rating as determined in No. 2 is to be not less than that determined by the following formulae:

For boilers with ratings 1800 sq ft to 4000 sq ft of steam radiation:

$$\text{Grate Area} = \sqrt{\frac{\text{Catalogue Rating (in square feet steam radiation)} - 200}{25.5}} \quad (1)$$

[2]For definitions of design load and maximum load see pages 451 and 452.
[3]See A.S.H.V.E. TRANSACTIONS, Vol. 35, 1929. Also Chapter 44.
[4]See A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930. Also Chapter 44.
[5]See A.S.H.V.E. TRANSACTIONS, Vol. 37, 1931. Also Chapter 44.

Table 2 Standard Steel Heating Boiler Ratings[a]

Hand Fired Capacity Rating					Mechanically Fired Capacity Rating			
Steam Radiation Sq Ft	Water Radiation Sq Ft	Btu per Hr	Heating Surface Sq Ft	Grate Area Sq Ft	Steam Radiation Sq Ft	Water Radiation Sq Ft	Btu per Hr	Furnace Volume Cu Ft Oil, Gas or Bituminous Coal
1,800	2,880	432,000	129	7.9	2,190	3,500	525,600	15 7
2,200	3,520	528,000	158	8.9	2,680	4,280	643,200	19 2
2,600	4,160	624,000	186	9 7	3,160	5,050	758,400	22.6
3,000	4,800	720,000	215	10 5	3,650	5,840	876,000	26 1
3,500	5,600	840,000	250	11 4	4,250	6,800	1,020,000	30 4
4,000	6,400	960,000	286	12 2	4,860	7,770	1,166,400	34.8
4,500	7,200	1,080,000	322	13 4	5,470	8,750	1,312,800	39.1
5,000	8,000	1,200,000	358	14 5	6,080	9,720	1,459,200	43 5
6,000	9,600	1,440,000	429	16 4	7,290	11,660	1,749,600	52 1
7,000	11,200	1,680,000	500	18 1	8,500	13,600	2,040,000	60 8
8,500	13,600	2,040,000	608	20 5	10,330	16,520	2,479,200	73 8
10,000	16,000	2,400,000	715	22 5	12,150	19,440	2,916,000	86 8
12,500	20,000	3,000,000	893	25 6	15,180	24,280	3,643,200	108.5
15,000	24,000	3,600,000	1,072	28 4	18,220	29,150	4,372,800	130 2
17,500	28,000	4,200,000	1,250	30 9	21,250	34,000	5,100,000	151 8
20,000	32,000	4,800,000	1,429	33 2	24,290	38,860	5,829,600	173.5
25,000	40,000	6,000,000	1,786	37 4	30,360	48,570	7,286,400	216 9
30,000	48,000	7,200,000	2,143	41 2	36,430	58,280	8,743,200	260 3
35,000	56,000	8,400,000	2,500	44.7	42,500	68,000	10,200,000	303.6

[a]Adopted by the *Steel Heating Boiler Institute* in cooperation with the *Bureau of Standards, United States Department of Commerce Simplified Practice Recommendation R 157-35*

For boilers with ratings 4000 sq ft of steam radiation and larger:

$$\text{Grate Area} = \sqrt{\frac{\text{Catalogue Rating (in square feet steam radiation)} - 1500}{16\ 8}} \qquad (2)$$

7. The volume for furnaces in which solid fuel is burned is to be considered as the cubical content of the space between the bottom of the fuel bed and the first plane of entry into or between the tubes. Volume of furnaces in whcih pulverized fuel liquid or gaseous fuel is burned are to be considered as the cubical content of the space between the hearth and the first plane of entry into or between the tubes No minimum furnace volume is to be specified for mechanical fired boilers burning anthracite.

8 The furnace volume for a boiler, with a rating as determined in No 3 in which oil, gas or bituminous coal stoker fired is burned is not to be less than one cubic foot for every 140 sq ft of steam rating

9. The average height of furnace for the rating determined in No 3 in which bituminous coal, stoker fired is burned is not to be less than that determined graphically in Fig. 1 or mathematically by the following formula

$$H = \sqrt{\frac{R}{22\ 5}} + \sqrt{\frac{R}{A}} \qquad (3)$$

where

H = average furnace height, inches as determined by the following formula;

$$H = \frac{12F}{A} = \frac{12F}{WL}$$

R = stoker fired boiler rating, square foot steam radiation.

A = plan area of firebox, square feet measured at the bottom of the fuel bed

F = furnace volume, cubic feet.

W = average width of furnace, measured at the bottom of the fuel bed, feet

L = length of furnace, feet If the furnace is longer than the fuel bed or contains a bridge wall, the total length of the furnace may be used except that this length is not to exceed $2\frac{1}{2}$ W

BOILER OUTPUT

Boiler output as defined in A.S.H.V.E. Performance Test Code for Steam Heating Solid Fuel Boilers (Code No. 3) is the quantity of heat available at the boiler nozzle with the boiler normally insulated. It should be based on actual tests conducted in accordance with this code. This output is usually stated in Btu and in square feet of equivalent heating surface (radiation). According to the A.S.H.V.E. Standard Code for

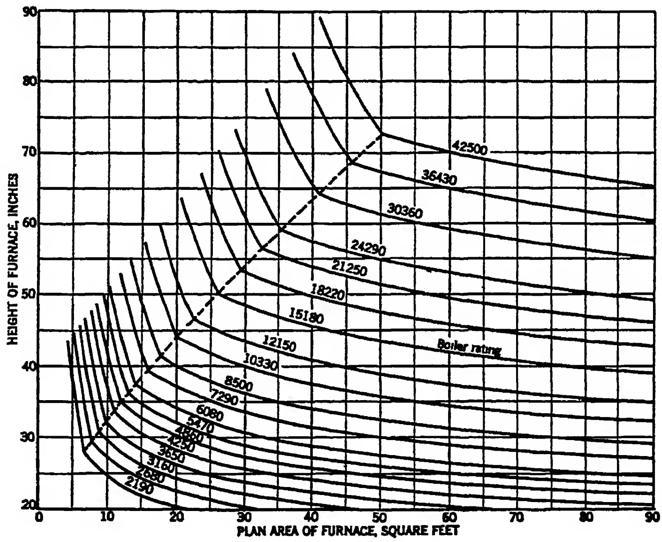


Fig. 1. Furnace Heights for Stoker Fired Boilers and Bituminous Coal Rated in Square Feet Steam Radiation

Rating Steam Heating Solid Fuel Hand-Fired Boilers, the performance data should be given in tabular or curve form on the following items for at least five outputs ranging from maximum down to 35 per cent of maximum: (1) fuel available, (2) combustion rate, (3) efficiency, (4) draft tension, (5) flue gas temperature. The only definite restriction placed on setting the maximum output is that priming shall not exceed 2 per cent. These curves provide complete data regarding the performance of the boiler under test conditions. Certain other pertinent information, such as grate area, heating surface and chimney dimensions is desirable also in forming an opinion of how the boiler will perform in actual service.

The output of large heating boilers is frequently stated in terms of *boiler horsepower* instead of in Btu per hour or square feet of equivalent radiation.

Boiler Horsepower: The evaporation of 34.5 lb of water per hour *from and at 212 F* which is equivalent to a heat output of 970.2 $\times$ 34.5 = 33,471.9 Btu per hour.

Equivalent Evaporation: The amount of water a boiler would evaporate, in pounds per hour, if it received feed water at 212 F and vaporized it at this same temperature and at atmospheric pressure.

It is usually considered that 10 sq ft of boiler heating surface will produce a rated boiler horsepower. A rated boiler horsepower in turn can carry a design load of from 100 to 140 sq ft of equivalent radiation. It is apparent, therefore, that 1 sq ft of boiler heating surface can carry a design load of from 10 to 14 sq ft of equivalent radiation, or somewhat more if the boiler is forced above rating. The application of these values is discussed under the heading Selection of Boilers.

BOILER EFFICIENCY

The term *efficiency* as used for guarantees of boiler performance is usually construed as follows:

1. *Solid Fuels.* The efficiency of the *boiler* alone is the ratio of the heat absorbed by the water and steam in the boiler per pound of combustible burned on the grate to the calorific value of 1 lb of combustible as fired The *combined* efficiency of *boiler, furnace and grate* is the ratio of the heat absorbed by the water and steam in the boiler per pound of fuel as fired to the calorific value of 1 lb of fuel as fired.

2. *Liquid Fuels.* The *combined* efficiency of boiler, furnace and burner is the ratio of the heat absorbed by the water and steam in the boiler per pound of fuel to the calorific value of 1 lb of fuel.

Solid fuel boilers usually show an efficiency of 50 to 75 per cent when operated under favorable conditions at their rated capacities. Information on the combined efficiencies of boiler, furnace and burner has resulted from research conducted at Yale University in cooperation with the A.S.H.V.E. Research Laboratory and the *American Oil Burner Association*[6]. For general information on heating efficiencies see Chapter 29.

SELECTION OF BOILERS

Estimated Design Load: The load, stated in Btu per hour or equivalent direct radiation, as estimated by the purchaser for the conditions of inside and outside temperature for which the amount of installed radiation was determined is the sum of the heat emission of the radiation to be actually installed plus the allowance for the heat loss of the connecting piping plus the heat requirement for any apparatus requiring heat connected with the system (A.S.H.V.E. Standard Code for Rating Steam Heating Solid Fuel Hand-Fired Boilers—Edition of April, 1932).

The estimated design load is the sum of the following three items[7]:

1. The estimated heat emission in Btu per hour of the connected radiation (direct, indirect or central fan) to be installed.

2. The estimated maximum heat in Btu per hour required to supply water heaters or other apparatus to be connected to the boiler.

[6]Study of the Characteristics of Oil Burners and Heating Boilers, by L E Seeley and E J Tavanlar (A S H V E TRANSACTIONS, Vol 37, 1931), A Study of Intermittent Operation of Oil Burners, by L E Seeley and J H Powers (A S H V E TRANSACTIONS, Vol. 38, 1932).

[7]A S H V.E Code of Minimum Requirements for the Heating and Ventilation of Buildings (Edition of 1929).

3 The estimated heat emission in Btu per hour of the piping connecting the radiation and other apparatus to the boiler

Estimated Maximum Load: Construed to mean the load stated in Btu per hour or the equivalent direct radiation that has been estimated by the purchaser to be the greatest or maximum load that the boiler will be called upon to carry. (A.S.H.V.E. Standard Code for Rating Steam Heating Solid Fuel Hand-Fired Boilers—Edition of April, 1932.)

The estimated maximum load is given by[8].

4 The estimated increase in the normal load in Btu per hour due to starting up cold radiation. This percentage of increase is to be based on the sum of Items 1, 2 and 3 and the heating-up factors given in Table 3

Table 3. Warming-up Allowances for Low Pressure Steam and Hot Water Heating Boilers[a, b, c]

Design Load (Representing Summation of Items 1, 2, and 3,[d]		Percentage Capacity to Add for Warming Up
Btu per Hour	Equivalent Square Feet of Radiation[d]	
Up to 100,000	Up to 420	65
100,000 to 200,000	420 to 840	60
200,000 to 600,000	840 to 2500	55
600,000 to 1,200,000	2500 to 5000	50
1,200,000 to 1,800,000	5000 to 7500	45
Above 1,800,000	Above 7500	40

[a]This table is taken from the A S H V E Code of Minimum Requirements for the Heating and Ventilation of Buildings, except that the second column has been added for convenience in interpreting the design load in terms of equivalent square feet of radiation

[b]See also Time Analysis in Starting Heating Apparatus, by Ralph C Taggert (A S H V E Transactions, Vol. 19, 1913); Report of A S.H V.E Continuing Committee on Codes for Testing and Rating Steam Heating Solid Fuel Boilers (A S H V E Transactions, Vol 36, 1930); Selecting the Right Size Heating Boiler, by Sabin Crocker (*Heating, Piping and Air Conditioning*, March, 1932)

[c]This table refers to hand-fired solid fuel boilers A factor of 25 per cent over design load is adequate when oil or gas are used as fuels

[d]240 Btu per square foot

Other things to be considered are:

5. Efficiency with hard or soft coal, gas, or oil firing, as the case may be
6. Grate area with hand-fired coal, or fuel burning rate with stokers, oil, or gas
7. Combustion space in the furnace.
8. Type of heat liberation, whether continuous or intermittent, or a combination of both.
9. Miscellaneous items consisting of draft available, character of attendance, possibility of future extension, possibility of breakdown, headroom in the boiler room

Radiation Load

The connected radiation (Item 1) is determined by calculating the heat losses in accordance with data given in Chapters 5, 6 and 7, and dividing by 240 to change to square feet of equivalent radiation as explained in Chapter 30. For hot water, the emission commonly used is 150 Btu per square foot, but the actual emission depends on the temperature of the medium in the heating units and of the surrounding air. (See Chapter 30.)

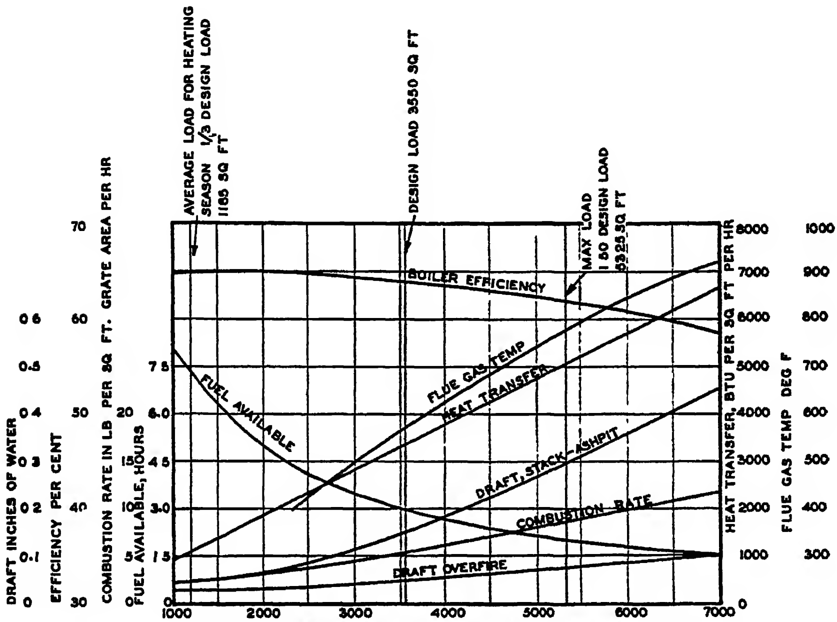
Although it is customary to use the actual connected load in equivalent square feet of radiation for selecting the size of boiler, this connected load usually represents a reserve in heating capacity to provide for infiltration in the various spaces of the building to be heated, which reserve, however,

[8]Loc Cit Note 8

is not in use at all places at the same time, or in any one place at all times. For a further discussion of this subject see Chapter 6.

Hot Water Supply Load

When the hot water supply (Item 2) is heated by the building heating boiler, this load must be taken into consideration in sizing the boiler. The



BOILER DATA—

GRATE AREA, SQ FT.	18 0
HEATING SURFACE, SQ FT	254
WEIGHT, LB.	9160
FUEL CAPACITY, LB.	651
FUEL AVAILABLE, LB.	414
FUEL DEPTH, IN	10

FUEL—BITUMINOUS 3/4" LUMP

ANALYSIS—

VOLATILE MATTER	34 68%
FIXED CARBON	55 44
ASH	9.87
SULPHUR	2.86
MOISTURE	3 00
BTU PER LB.	13,655

Fig. 2. Typical Performance Curves for a 36-in. Cast-Iron Sectional Steam Heating Boiler, Based on the A.S H.V.E. Code for Rating Steam Heating Solid Fuel Hand-Fired Boilers

allowance to be made will depend on the amount of water heated and its temperature rise. A good approximation is to add 4 sq ft of equivalent radiation for each gallon of water heated per hour through a temperature range of 100 F. For more specific information, see Chapter 35.

Piping Tax (Item 3)

It is common practice to add a flat percentage allowance to the equivalent connected radiation to provide for the heat loss from bare and covered pipe in the supply and return lines. The use of a flat allowance of 25 per cent for steam systems and 35 per cent for hot water systems is preferable to ignoring entirely the load due to heat loss from the supply

and return lines, but better practice, especially when there is much bare pipe, is to compute the emission from both bare and covered pipe surface in accordance with data in Chapter 36. With direct radiation served by bare supply and return piping the percentages may be higher than those stated, while in the case of unit heaters where the output is concentrated in a few locations, the piping tax may be 10 per cent or less.

Warming-Up Allowance

The warming-up allowance represents the load due to heating the boiler and contents to operating temperature and heating up cold radiation and piping. (See Item 4.) The factors to be used for determining the allowance to be made should be selected from Table 3 and should be applied to the estimated design load as determined by Items 1, 2 and 3.

Performance Curves for Boiler Selection

In the selection of a boiler to meet the estimated load, the A.S.H.V.E. Standard Code for Rating Steam Heating Solid Fuel Hand-Fired Boilers recommends the use of performance curves based on actual tests conducted in accordance with the A.S.H.V.E. Performance Test Code for Steam Heating Solid Fuel Boilers (Code No. 3), similar to the typical curves shown in Fig. 2. It should be understood that performance data apply to test conditions and that a reasonable allowance should be made for decreased output resulting from soot deposit, poor fuel or inefficient attention.

Selection Based on Heating Surface and Grate Area

Where performance curves are not available, a good general rule for conventionally-designed boilers is to provide 1 sq ft of boiler heating surface for each 14 sq ft of equivalent radiation (240 Btu per square foot) represented by the design load consisting of connected radiation, piping tax and domestic water heating load. As stated in the section on Boiler Output, this is equivalent to allowing 10 sq ft of boiler heating surface per boiler horsepower. In this case it is assumed that the maximum load including the warming-up allowance will be provided for by operating the boiler in excess of the design load, that is, in excess of the 100 per cent rating on a boiler-horsepower basis.

Due to the wide variation encountered in manufacturers' ratings for boilers of approximately the same capacity, it is advisable to check the grate area required for heating boilers burning solid fuel by means of the following formula:

$$G = \frac{H}{C \times F \times E} \tag{4}$$

where

G = grate area, square feet.

H = required total heat output of the boiler, Btu per hour (see Selection of Boilers, p. 451).

C = combustion rate in pounds of dry coal per square foot of grate area per hour, depending on the kind of fuel and size of boiler as given in Table 1.

F = calorific value of fuel, Btu per pound.

E = efficiency of boiler, usually taken as 0.60.

Example 1. Determine the grate area for a required heat output of the boiler of

500,000 Btu per hour, a combustion rate of 6 lb per hour, a calorific value of 13,000 Btu per pound, and an efficiency of 60 per cent

$$G = \frac{500{,}000}{6 \times 13{,}000 \times 0.60} = 10.7 \text{ sq ft}$$

The boiler selected should have a grate area not less than that determined by Formula 4. With small boilers where it is desired to provide sufficient coal capacity for approximately an eight-hour firing period plus a 20 per cent reserve for igniting a new charge, more grate area may be required depending upon the depth of the fuel pot.

Selection of Steel Heating Boilers

Boiler ratings previously described under the *Steel Heating Boiler Institute's* Boiler Rating Code are intended to correspond with the estimated design load based on the sum of items 1, 2 and 3 outlined on pages 451 to 453. Insulated residence type boilers for oil or gas may carry a net load expressed in square feet of steam radiation of not more than 17 times the square feet of heating surface in the boiler, provided the boiler manufacturer guarantees the boiler to be capable of operating at a maximum output of not less than 150 per cent of net load rating with overall efficiency of not less than 75 per cent with at least two different makes of each type of standard commercial burner recommended by the boiler manufacturer. If the heat loss from the piping system exceeds 20 per cent of the installed radiation, the excess is to be considered as a part of the net load.

TABLE 4 BOILER RATINGS BASED ON NET LOAD[a]

HAND FIRED RATINGS		MECHANICALLY FIRED RATINGS	
Steam Radiation Sq Ft	Net Load[b] Steam Radiation Sq Ft	Steam Radiation Sq Ft	Net Load[b] Steam Radiation Sq Ft
1,800	1,389	2,190	1,695
2,200	1,702	2,680	2,089
2,600	2,020	3,160	2,461
3,000	2,335	3,650	2,853
3,500	2,732	4,250	3,335
4,000	3,135	4,860	3,830
4,500	3,540	5,470	4,330
5,000	3,945	6,080	4,834
6,000	4,770	7,290	5,850
7,000	5,608	8,500	6,885
8,500	6,885	10,330	8,490
10,000	8,197	12,150	10,125
12,500	10,417	15,180	12,650
15,000	12,500	18,220	15,183
17,500	14,584	21,250	17,708
20,000	16,667	24,290	20,242
25,000	20,834	30,360	25,300
30,000	25,000	36,430	30,359
35,000	29,167	42,500	35,417

[a]Adopted by the *Steel Heating Boiler Institute* in cooperation with the *Bureau of Standards, United States Department of Commerce Simplified Practice Recommendation R 157-35*

[b]The net load is made up by the sum of the estimated design load, items 1 and 2 (pages 451 to 453). All net loads are expressed in 70 F For hand fired boiler ratings less than 1800 sq ft of steam or 2880 sq ft of water and mechanically fired boiler ratings of 2190 sq ft of steam or 3500 sq ft of water, apply the factor 1.3 to the net load to determine the boiler size. For water boilers use the equivalent net load for steam boilers of similar physical size.

When the estimated heat emission of the piping (connecting radiation, and other apparatus to the boiler) is not known the net load to be considered for the boiler may be determined from Table 4.

Selection of Gas-Fired Boilers

Gas-heating appliances should be selected in accordance with factors given in Table 2, Chapter 28, which include an allowance for heating up cold radiation, and for the piping tax. These factors are for thermostatically-controlled systems; in case manual operation is desired, a warming-up allowance of 100 per cent is recommended by the *A.G.A.* A gas boiler selected by the use of the *A.G.A.* factors will be the minimum size boiler which can carry the load. From a fuel economy standpoint, it may be advisable to select a somewhat larger boiler and then throttle the gas and air adjustments as required. This will tend to give a low stack temperature with high efficiency and at the same time provide reserve capacity in case the load is underestimated or more is added in the future.

Conversions

The conversion of a coal or oil boiler to gas burning is simpler than the reverse since little furnace volume need be provided for the proper combustion of gas. When a solid fuel boiler of 500 sq ft or less capacity is converted to gas burning, the necessary gas heat units should be approximately double the connected load. The presumption for a conversion job is that the boiler is installed and probably will not be made larger; therefore, it is a matter of setting a gas-burning rate to obtain best results with the available surface. Assuming a combustion efficiency of 75 per cent for a conversion installation the boiler output would be $2 \times 0.75 = 1.5$ times the connected load, which allows 50 per cent for piping tax and pickup. In converting large boilers, the determination of the required Btu input should not be done by an arbitrary figure or factor but should be based on a detailed consideration of the requirements and characteristics of the connected load.

An efficient conversion installation depends upon the proper size of flue connection. Often the original smoke breeching between the boiler and chimney is too large for gas firing, and in this case, flue orifices can be used, which are discs provided with an opening of the size for the gas input used in this boiler. The size should be based on 1 sq in. of flue area for each 7500 hourly Btu input.

If dampers are found in the breeching they should be locked in position so that they will not interfere with the normal operation of the gas burners at maximum flow. In the case of large boiler conversions, automatic damper regulators proportion the position of the flue dampers to the amount of gas flowing and may be substituted for existing dampers. Generally in residence conversions automatic dampers are not of the proportioning type but close the flue during the off periods of the gas burners. Automatic shutoff dampers should be located between the backdraft diverter and the chimney flue. Automatic dampers are usually designed to operate with electric contact mechanism, but frequently an arrangement is utilized which functions with mechanical fluid or gas pressure.

As it will usually be found that several boilers will meet the speci-

fications, the final selection of the boiler may be influenced by other considerations, some of which are

1. Dimensions of boiler.
2. Durability under service.
3. Convenience in firing and cleaning
4. Adaptability to changes in fuel and kind of attention.
5. Height of water line.

In large installations, the use of several smaller boiler units instead of one larger one will obtain greater flexibility and economy by permitting the operation, at the best efficiency, of the required number of units according to the heat requirements.

Boiler rooms should, if possible, be situated at a central point with respect to the building and should be designed for a maximum of natural light. The space in front of the boilers should be sufficient for firing, stoking, ash removal and cleaning or renewal of flues, and should be at least 3 ft greater than the length of the boiler firebox.

A space of at least 3 ft should be allowed on at least one side of every boiler for convenience of erection and for accessibility to the various dampers, cleanouts and trimmings. The space at the rear of the boiler should be ample for the chimney connection and for cleanouts, and with large boilers the rear clearance should be at least 3 ft in width.

The boiler room height should be sufficient for the location of boiler accessories and for proper installation of piping. In general the ceiling height for small steam boilers should be at least 3 ft above the normal boiler water line. With vapor heating, especially, the height above the boiler water line is of vital importance.

When steel boilers are used, space should be provided for the removal and replacement of tubes.

CONNECTIONS AND FITTINGS

The velocity of flow through the outlets of low pressure steam heating boilers should not exceed 15 to 25 fps if fluctuation of the water line and undue entrainment of moisture are to be avoided. Steam or water outlet connections preferably should be the full size of the manufacturers' tapping and should extend vertically to the maximum height available above the boiler. For gravity circulating steam heating systems, it is recommended that a Hartford Loop, described in Chapter 32, be utilized in making the return connection.

Particular attention should be given to *fitting connections* to secure conformity with the *A.S.M.E.* Boiler Construction Code for Low Pressure Heating Boilers. Attention is called in particular to pressure gage piping, water gage connections and safety valve capacity.

Steam gages should be fitted with a water seal and a shut-off consisting of a cock with either a tee or lever handle which is parallel to the pipe when the cock is open. *Steam gage connections* should be of copper or brass when smaller than 1 in. *I.P.S.*[9] if the gage is more than 5 ft from the

[9]*A.S.M.E.* Code, Identification of Piping Systems.

boiler connection, and also in any case where the connection is less than ½ in. *I.P.S.*

Each steam or vapor boiler should have at least one *water gage* glass and two or more *gage cocks* located within the range of the visible length of the glass. The water gage fittings or gage cocks may be direct connected to the boiler, if so located by the manufacturer, or may be mounted on a separate water column. No connections, except for combustion regulators, drains or steam gages, should be placed on the pipes connecting the water column and the boiler. If the water column or gage glass is connected to the boiler by pipe and fittings, a cross, tee or equivalent, in which a cleanout plug or a drain valve and piping may be attached, should be placed in the water connection at every right-angle turn to facilitate cleaning. The water line in steam boilers should be carried at the level specified by the boiler manufacturer.

Safety valves should be capable of discharging all the steam that can be generated by the boiler without allowing the pressure to rise more than 5 lb above the maximum allowable working pressure of the boiler. This should be borne in mind particularly in the case of boilers equipped with mechanical stokers or oil burners where the amount of grate area has little significance as to the steam generating capacity of the boiler.

Where a *return header* is used on a cast-iron sectional boiler to distribute the returns to both rear tappings, it is advisable to provide full size plugged tees instead of elbows where the branch connections enter the return tappings. This facilitates cleaning sludge from the bottom of the boiler sections through the large plugged openings. An equivalent clean-out plug should be provided in the case of a single return connection.

Blow-off or drain connections should be made near the boiler and so arranged that the entire system may be drained of water by opening the drain cock. In the case of two or more boilers separate blow-off connections must be provided for each boiler on the boiler side of the stop valve on the main return connection.

Water service connections must be provided for both steam and water boilers, for refilling and for the addition of make-up water to boilers. This connection is usually of galvanized steel pipe, and is made to the return main near the boiler or boilers.

For further data on pipe connections for steam and hot water heating systems, see Chapters 32 and 33 and the *A.S.M.E.* Boiler Construction Code for Low Pressure Heating Boilers.

Smoke Breeching and Chimney Connections. The breeching or smoke pipe from the boiler outlet to the chimney should be air-tight and as short and direct as possible, preference being given to long radius and 45-deg instead of 90-deg bends. The breeching entering a brick chimney should not project beyond the flue lining and where practicable it should be grouted up from the inside of the chimney. A thimble or sleeve grout usually is provided where the breeching enters a brick chimney.

Where a battery of boilers is connected into a breeching each boiler should be provided with a tight damper. The breeching for a battery of boilers should not be reduced in size as it goes to the more remote boilers. Good connections made to a good chimney will usually result in a rapid response by the boilers to demands for heat.

ERECTION, OPERATION, AND MAINTENANCE

The directions of the boiler manufacturer always should be read before the assembly or installation of any boiler is started, even though the contractor may be familiar with the boiler. All joints requiring boiler putty or cement which cannot be reached after assembly is complete must be finished as the assembly progresses.

The following precautions should be taken in all installations to prevent damage to the boiler:

1 There should be provided proper and convenient drainage connections for use if the boiler is not in operation during freezing weather

2. Strains on the boiler due to movement of piping during expansion should be prevented by suitable anchoring of piping and by proper provision for pipe expansion and contraction.

3 Direct impingement of too intense local heat upon any part of the boiler surface, as with oil burners, should be avoided by protecting the surface with firebrick or other refractory material.

4 Condensation must flow back to the boiler as rapidly and uniformly as possible. Return connections should prevent the water from backing out of the boiler.

5 Automatic boiler feeders and low water cut-off devices which shut off the source of heat if the water in the boiler falls below a safe level are recommended for boilers mechanically fired

Boiler Troubles

A complaint regarding boiler operation generally will be found to be due to one of the following:

1. *The boiler fails to deliver enough heat.* The cause of this condition may be: (*a*) poor draft, (*b*) poor fuel, (*c*) inferior attention or firing, (*d*) boiler too small, (*e*) improper piping, (*f*) improper arrangement of sections, (*g*) heating surfaces covered with soot; and (*h*) insufficient radiation installed.

2. *The water line is unsteady.* The cause of this condition may be: (*a*) grease and dirt in boiler, (*b*) water column connected to a very active section and, therefore, not showing actual water level in boiler; and (*c*) boiler operating at excessive output.

3. *Water disappears from gage glass* This may be caused by: (*a*) priming due to grease and dirt in boiler, (*b*) too great pressure difference between supply and return piping preventing return of condensation; (*c*) valve closed in return line; (*d*) connection of bottom of water column into a very active section or thin waterway, and (*e*) improper connections between boilers in battery permitting boiler with excess pressure to push returning condensation into boiler with lower pressure.

4 *Water is carried over into steam main.* This may be caused by: (*a*) grease and dirt in boiler; (*b*) insufficient steam dome or too small steam liberating area; (*c*) outlet connections of too small area, (*d*) excessive rate of output; and (*e*) water level carried higher than specified.

5. *Boiler is slow in response to operation of dampers* This may be due to (*a*) poor draft due to air leaks into chimney or breeching, (*b*) inferior fuel; (*c*) inferior attention; (*d*) accumulation of clinker on grate; (*e*) boiler too small for the load.

6. *Boiler requires too frequent cleaning of flues.* This may be due to: (*a*) poor draft; (*b*) smoky combustion, (*c*) too low a rate of combustion, and (*d*) too much excess air in firebox causing chilling of gases.

7. *Boiler smokes through fire door.* This may be due to: (*a*) defective draft in chimney or incorrect setting of dampers, (*b*) air leaks into boiler or breeching; (*c*) gas outlet from firebox plugged with fuel, (*d*) dirty or clogged flues; and (*e*) improper reduction in breeching size.

Cleaning Steam Boilers

All boilers are provided with flue clean-out openings through which the

heating surface can be reached by means of brushes or scrapers. Flues of solid fuel boilers should be cleaned often to keep the surfaces free of soot or ash. Gas boiler flues and burners should be cleaned at least once a year. Oil burning boiler flues should be examined periodically to determine when cleaning is necessary.

The grease used to lubricate the cutting tools during erection of new piping systems serves as a carrier for sand and dirt, with the result that a scum of fine particles and grease accumulates on the surface of the water in all new boilers, while heavier particles may settle to the bottom of the boiler and form sludge. These impurities have a tendency to cause foaming, preventing the generation of steam and causing an unsteady water line.

This unavoidable accumulation of oil and grease should be removed by blowing off the boiler as follows: If not already provided, install a surface blow connection of at least 1¼ in. nominal pipe size with outlet extended to within 18 in. of the floor or to sewer, inserting a valve in line close to boiler. Bring the water line to center of outlet, raise steam pressure and while fire is burning briskly open valve in blow-off line. When pressure recedes close valve and repeat process adding water at intervals to maintain proper level. As a final operation bring the pressure in the boiler to about 10 lb, close blow-off, draw the fire or stop burner, and open drain valve. After boiler has cooled partly, fill and flush out several times before filling it to proper water level for normal service. The use of soda, or any alkali, vinegar or any acid is not recommended for cleaning heating boilers because of the difficulty of complete removal and the possibility of subsequent injury, after the cleaning process has been completed.

Insoluble compounds have been developed which are effective, but special instructions on the proper cleaning compound and directions for its use in a boiler, as given by the boiler manufacturer, should be carefully followed.

It is common practice when starting new installations to discharge heating returns to the sewer during the first week of operation. This prevents the passage of grease, dirt or other foreign matter into the boiler and consequently may avoid the necessity of cleaning the boiler. During the time the returns are being passed to the sewer, the feed valve should be cracked sufficiently to maintain the proper water level in the boiler.

Care of Idle Heating Boilers

Heating boilers are often seriously damaged during summer months due chiefly to corrosion resulting from the combination of sulphur from the fuel with the moisture in the cellar air. At the end of the heating season the following precautions should be taken:

1. All heating surfaces should be cleaned thoroughly of soot, ash and residue, and the heating surfaces of steel boilers should be given a coating of lubricating oil on the fire side.
2. All machined surfaces should be coated with oil or grease.
3. Connections to the chimney should be cleaned and in case of small boilers the pipe should be placed in a dry place after cleaning.
4. If there is much moisture in the boiler room, it is desirable to drain the boiler to prevent atmospheric condensation on the heating surfaces of the boiler when they are below the dew-point temperature. Due to the hazard of some one inadvertently building

a fire in a dry boiler, however, it is safer to keep the boiler filled with water A hot water system usually is left filled to the expansion tank

5 The grates and ashpit should be cleaned.

6 Clean and repack the gage glass if necessary.

7. Remove any rust or other deposit from exposed surfaces by scraping with a wire brush or sandpaper After boiler is thoroughly cleaned, apply a coat of preservative paint where required to external parts normally painted

8. Inspect all accessories of the boiler carefully to see that they are in good working order. In this connection, oil all door hinges, damper bearings and regulator parts

BOILER INSULATION

Insulation for cast-iron boilers is of two general types: (1) plastic material or blocks wired on, cemented and covered with canvas or duck; and (2) blocks, sheets or plastic material covered with a metal jacket furnished by the boiler manufacturer. Self-contained steel firebox boilers usually are insulated with blocks, cement and canvas, or rock wool blankets; HRT boilers are brick set and do not require insulation beyond that provided in the setting. It is essential that the insulation on a boiler and adjacent piping be of non-combustible material as even slow-burning insulation constitutes a dangerous fire hazard in case of low water in the boiler.

REFERENCES

A S.H.V.E. Code of Minimum Requirements for the Heating and Ventilation of Buildings.

A.S.H.V.E. Standard and Short Form Heat Balance Codes for Testing Low-Pressure Steam Heating Solid Fuel Boilers (Codes 1 and 2).

A.S.H.V.E. Performance Test Code for Steam Heating Solid Fuel Boilers (Code No. 3).

A.S.H.V.E. Standard Code for Rating Steam Heating Solid Fuel Hand-Fired Boilers.

Heating, Ventilating and Air Conditioning, by Harding and Willard, Revised Edition, 1932.

A.S.M.E. Boiler Construction Code for Low Pressure Heating Boilers.

Heating, Piping and Air Conditioning Contractors National Association Standards (boiler selection tables).

House Heating, published by *American Gas Association*.

Handbook of Oil Burning, published by *American Oil Burner Association*.

Heating and Air Conditioning, by Allen and Walker (fourth edition).

Selecting the Right Size Boiler, by Sabin Crocker (*Heating, Piping and Air Conditioning*, February, March, April, 1932).

PROBLEMS IN PRACTICE

1 ● What basic requirements of boiler design are to be accomplished with a combination boiler and oil burner unit?

Combination units vary widely but in general, the basic requirements of design depends upon a combustion chamber of proper design and arranged for the flame shape with adequate heating surface for the complete combustion of the fuel.

2 ● Name the construction materials that distinguish two types of low pressure heating boilers.

a Cast-iron.

b. Steel.

3 ● What is the normal rating range of each type of boiler?

a. Cast-iron boilers are rated at from 200 to 18,000 sq ft EDR

b. Steel boilers are rated at from 300 to 50,000 sq ft EDR

4 ● a. What is meant by direct boiler heating surface?
b. What is meant by indirect boiler heating surface?

a. Direct boiler heating surface is that boiler surface upon which the fire shines, namely, the walls of the firebox and the crown sheet.

b. Indirect boiler heating surface is that boiler surface not exposed to the direct rays of the fire and over which heated gases pass after they have been in contact with the direct surface. Indirect surface is generally known as convective surface.

5 ● What is the average heat transmission rate in heating boilers in Btu per sq ft of heating surface per hour?

3500 for coal burning boilers; 4200 for oil burning boilers

6 ● What factors contribute to economical fuel operation in low pressure boilers burning coal or oil?

a. Proper furnace volume for complete combustion.

b. Arrangement of heating surfaces in series to create a turbulent and scrubbing contact of gases against the convective surfaces.

c. Rapid internal water circulation which will remove steam bubbles from the water side of heating surfaces and allow other steam bubbles to be formed Rapid disengagement of steam bubbles increases the steam generating efficiency of each unit area of heating surface, and thereby lowers flue gas temperatures

7 ● What equipment is usually directly attached to a low pressure heating boiler?

For coal burning steam boilers water column, water gage, tri-cocks, steam gage, lever pop safety valve, boiler damper regulator

For coal burning hot water boilers. damper regulator, altitude gage, thermometer, relief valve

For oil burning boilers, the damper regulators are omitted and the following additional equipment is usually attached: automatic water feeder, low water cutout, a pressure control, and a water temperature control. These are generally furnished by the oil burner manufacturer and do not come with the boiler.

8 ● What general precautions regarding the boiler should be taken to make sure a proposed heating installation will work properly?

a. Select the right size and type of boiler.

b. Be sure the combustion space is proper for the type of fuel burned.

c. Allow sufficient space around the boiler for cleaning.

d. Secure proper height and area of chimney and connecting breeching.

e. Clean the boiler thoroughly and provide surface blowoff connections and bottom blowoff connections for periodic cleaning after operation is begun.

f. See that the boiler heating surface is cleaned at regular periods.

g. Check flue gas temperatures and make a flue gas analysis at least once a month.

h. Secure information and advice from boiler manufacturer.

9 ● Below what temperature should the water in direct water heaters be maintained to reduce scale formation and corrosion?

140 F.

10 ● a. What is the heat equivalent of a boiler horsepower?
b. How many square feet of heating surface are usually required per boiler horsepower?

a. 33,471.9 Btu per hour.

b. 10 sq ft.

Chapter 26

CHIMNEYS AND DRAFT CALCULATIONS

Natural Draft, Mechanical Draft, Characteristics of Natural Draft Chimneys, Determining Chimney Sizes, General Equation, Chimney Construction, Chimneys for Gas Heating

THE design and construction of a chimney is so important a part of the heating engineer's work that a general knowledge of draft characterictics and calculations is essential.

Draft, in general, may be defined as the pressure difference between the atmospheric pressure and that at any part of an installation through which the gases flow. Since a pressure difference implies a head, draft is a static force. While no element of motion is inferred, yet motion in the form of circulation of gases throughout an entire boiler plant installation is the direct result of draft. This motion is due to the pressure difference, or unbalanced pressure, which compels the gases to flow. Draft is often classified into two kinds according to whether it is created thermally or artificially, *viz*, (1) natural or thermal draft, and (2) artificial or mechanical draft.

Natural Draft

Natural draft is the difference in pressure produced by the difference in weight between the relatively hot gases inside a natural draft chimney and an equivalent column of the cooler outside air, or atmosphere. Natural draft, in other words, is an unbalanced pressure produced thermally by a natural draft chimney as the pressure transformer and a temperature difference. The intensity of natural draft depends, for the most part, upon the height of the chimney above the grate bar level and also the temperature difference between the chimney gases and the atmosphere.

A typical natural draft system consists essentially of a relatively tall chimney built of steel, brick, or reinforced concrete, operating with the relatively hot gases which have passed through the boilers and accessories and from which all the heat has not been extracted. Hot gases are an essential element in the operation of a natural draft system, although inherently a heat balance loss.

A natural draft chimney performs the two-fold service of assisting in the creation of draft by aspiration and also of discharging the gases at an elevation sufficient to prevent them from becoming a nuisance.

Natural draft is quite advantageous in installations where the total loss of draft due to resistances is relatively low and also in plants which have practically a constant load and whose boilers are seldom operated above

their normal rating. Natural draft systems have been, and are still being, employed in the operation of large plants during the periods when the boilers are operated only up to their normal rating. When the rate of operation is increased above the normal rating, some form of mechanical draft is employed as an auxiliary to overcome the increased resistances or draft losses. Natural draft systems are used almost exclusively in the smaller size plants where the amount of gases generated is relatively small and it would be expensive to install and operate a mechanical draft system.

The principal advantages of natural draft systems may be summarized as follows: (1) simplicity, (2) reliability, (3) freedom from mechanical

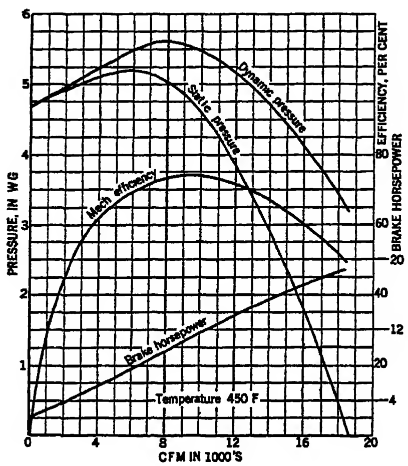


Fig. 1. General Operating Characteristics of Typical Induced Draft Fan

parts, (4) low cost of maintenance, (5) relatively long life, (6) relatively low depreciation, and (7) no power required to operate. The principal disadvantages are: (1) lack of flexibility, (2) irregularity, (3) affected by surroundings, and (4) affected by temperature changes.

Mechanical Draft

Artificial draft, or mechanical draft, as it is more commonly called, is a difference in pressure produced either directly or indirectly by a forced draft fan, an induced draft fan, or a Venturi chimney as the pressure transformer. The intensity of mechanical draft is dependent for the most part upon the size of the fan and the speed at which it is operated. The element of temperature does not enter into the creation of mechanical draft and therefore its intensity, unlike natural draft, is independent of the temperature of the gases and the atmosphere. Mechanical draft includes the induced and Venturi types of draft systems in which the pressure difference is the result of a suction, and also the forced draft system in which the pressure difference is the result of a blowing. Mechanical draft systems tend to produce a vacuum or a plenum, as the system used in its production creates a pressure difference below, or above, atmospheric

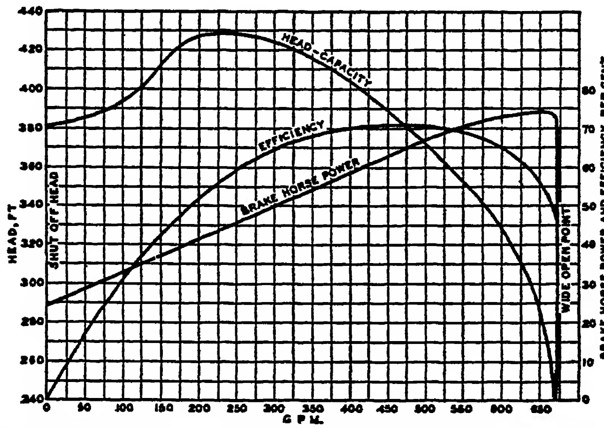


FIG. 2. OPERATING CHARACTERISTICS OF TYPICAL CENTRIFUGAL PUMP

pressure, respectively. A mechanical draft system may be used either in conjunction with, or as an adjunct to, a natural draft system.

CHARACTERISTICS OF CHIMNEYS

In order to analyze the performance of a natural draft chimney, it may be advantageous to compare its general operating characteristics with those of a centrifugal pump and also of a centrifugally-induced draft fan, there being a similarity among the three. Figs. 1, 2 and 3 show the general operating characteristics of a typical centrifugally-induced draft fan, a typical centrifugal pump, and a typical natural draft chimney, respectively. The draft-capacity curve of the chimney corresponds to

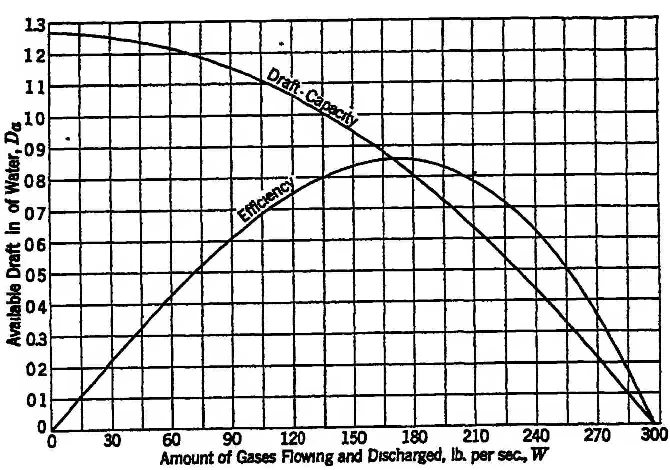


FIG. 3. TYPICAL SET OF OPERATING CHARACTERISTICS OF A NATURAL DRAFT CHIMNEY

the head-capacity curve of the pump and also to the dynamic-head capacity curve of the fan.

When the gases in the chimney are stationary, the draft created is termed the *theoretical draft.* When the gases are flowing, the theoretical intensity is diminished by the draft loss due to friction, the difference between the two being termed the total *available draft.* The general equation for this net total available draft intensity of a natural draft chimney with a circular section is as follows:

$$D_a = 2.96HB_0\left(\frac{W_0}{T_0} - \frac{W_C}{T_C}\right) - \frac{0.00126W^2T_CfL}{D^5B_0W_C} \quad (1)$$

where

D_a = available draft, inches of water
H = height of chimney above grate bars, feet.
B_0 = barometric pressure corresponding to altitude, inches of mercury
W_0 = unit weight of a cubic foot of air at 0 F and sea level atmospheric pressure, pounds per cubic foot.
W_C = unit weight of a cubic foot of chimney gases at 0 F and sea level atmospheric pressure, pounds per cubic foot
T_0 = absolute temperature of atmosphere, degrees Fahrenheit.
T_C = absolute temperature of chimney gases, degrees Fahrenheit.
W = amount of gases generated in the combustion chamber of the boiler and passing through the chimney, pounds per second.
f = coefficient of friction.
L = length of friction duct of the chimney, feet.
D = minimum diameter of chimney, feet.

The first term of the right hand expression of Equation 1 represents the theoretical draft intensity, and the second term, the loss due to friction.

Example 1. Determine the available draft of a natural draft chimney 200 ft in height and 10 ft in diameter operating under the following conditions: atmospheric temperature, 62 F; chimney gas temperature, 500 F, sea level atmospheric pressure, B_0 = 29.92 in. of mercury; atmospheric and chimney gas density, 0 0863 and 0 09, respectively; coefficient of friction, 0.016; length of friction duct, 200 ft The chimney discharges 100 lb of gases per second

Substituting these values in Equation 1 and reducing.

$$D_a = 2.96 \times 200 \times 29.92 \times \left(\frac{0.0863}{522} - \frac{0.09}{960}\right) - \frac{0.00126 \times 100^2 \times 960 \times 0.016 \times 200}{10^5 \times 29.92 \times 0.09}$$

$$= 1.27 - 0.14 = 1.13 \text{ in.}$$

Fig. 3 shows the variation in the available draft of a typical 200 ft by 10 ft chimney operating under the general conditions noted in Example 1. When the chimney is under static conditions and no gases are flowing, the available draft is equal to 1.27 in. of water, the theoretical intensity. As the amount of gases flowing increases, the available intensity decreases until it becomes zero at a gas flow of 297 lb per second, at which point the draft loss due to friction is equal to the theoretical intensity. The draft-capacity curve corresponds to the head-capacity curve of centrifugal pump characteristics and the dynamic-head-capacity curve of a fan. The point of maximum draft and zero capacity is called shut-off draft, or point of impending delivery, and corresponds to the point of shut-off head of a centrifugal pump. The point of zero draft and maximum capacity is

called the wide open point and corresponds to the wide open point of a centrifugal pump. A set of operating characteristics may be developed for any size chimney operating under any set of conditions by substituting the proper values in Equation 1 and then plotting the results in the manner shown in Fig. 3.

In substituting the values for the various factors in Equation 1, care should be exercised that the selections be as near the actual conditions as is practically possible The following notes will serve as a guide for these selections:

1. The *barometric pressure* varies inversely as the altitude of the plant above sea level. Fig. 4 gives the barometric pressure corresponding to various elevations as computed from the equation:

$$E_1 = 62{,}737 \log_{10} \frac{29.92}{B_o} \qquad (2)$$

where

E_1 = altitude of plant above sea level, feet.

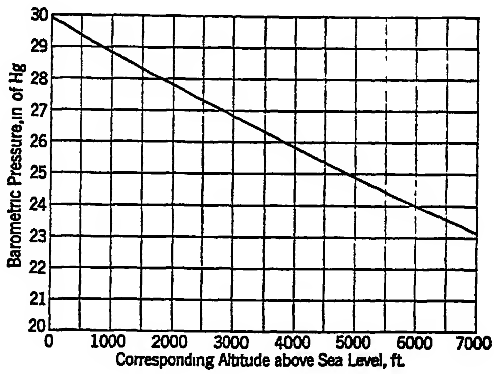


FIG. 4. RELATION BETWEEN BAROMETRIC PRESSURE AND ALTITUDE

In general, the barometric pressure decreases approximately 0.1 in. of mercury per 100 ft increase in elevation.

2. The *unit weight of a cubic foot* of chimney gases at 0 F and sea level barometric pressure is given by the equation:

$$W_C = 0.131CO_2 + 0.095\ O_2 + 0.083\ N_2 \qquad (3)$$

In this equation CO_2, O_2 and N_2 represent the percentages of the parts by volume of the carbon dioxide, oxygen and nitrogen content, respectively, of the gas analysis. For ordinary operating conditions, the value of W_C may be assumed at 0.09

The density effect on the chimney gases due to superheated water vapor resulting from moisture and hydrogen in the fuel, or due to any air infiltrations in the chimney proper are here disregarded. Though water vapor content is not disclosed by Orsat analysis, its presence tends to reduce the actual weight per cubic foot of chimney gases.

3. The *atmospheric temperature* is the actual observed temperature of the outside air at the time the analysis of the operating chimney is made. The mean atmospheric temperature in the temperate zone is approximately 62 F.

4. The *chimney gas temperature* does not vary appreciably from the gas temperature as it leaves the breeching and enters the chimney. For average operating conditions, the chimney gas temperature will vary between 500 F and 650 F except in the case when

economizers and recuperators are used, when the temperature will vary between 300 F and 450 F. If a chimney has been properly constructed, properly lined and has no air infiltration due to open joints, the temperature of the gases throughout the chimney will not differ appreciably from the foregoing figures. In most up-to-date heating plants, the temperature may be read from instruments or ascertained from a pyrometer. The analysis of this section is predicated on the assumption of constant gas temperature and no air infiltration throughout the height of the chimney.

5 The *coefficient of friction* between the chimney gases and a sooted surface has been taken by many workers in this field as a constant value of 0.016 for the conditions involved. This value, of course, would be less for a new unlined steel stack than for a brick or brick-lined chimney, but in time the inside surface of all chimneys regardless of the materials of construction becomes covered with a layer of soot, and thus the coefficient of friction has been taken the same for all types of chimneys and in general constant for all conditions of operation For reasons of simplicity and convenience to

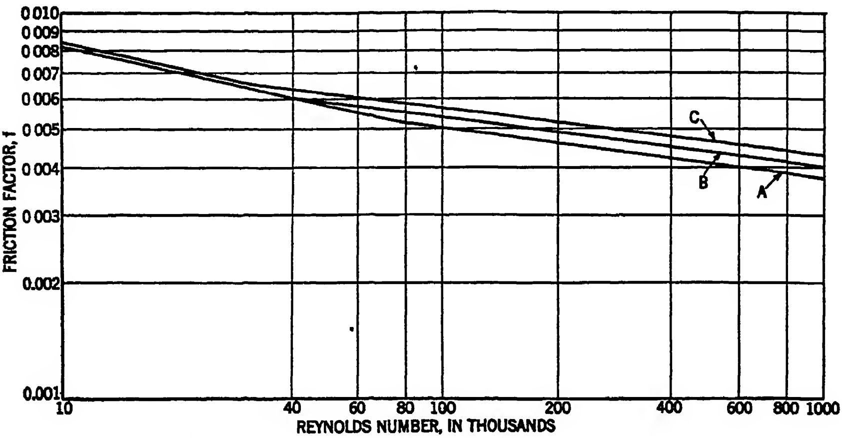


Fig. 5. Variation of Friction Factor f with Reynolds Number

the reader, this constant value of 0.016 has been employed in the development of the various special equations and charts shown in this chapter.

However, much to be recommended as an alternate method is the practise of separately determining duct friction factors as a function of the flow conditions, specifically as a function of the Reynolds number and the relative duct roughness The Reynolds criterion is based on the physical properties of the gas, the duct dimensions, and the gas velocity. The gas velocity for a chimney is usually well above the critical velocity. It is likely that this procedure of using a separately determined variable friction factor for chimney flow will give results that are to be preferred over those based on a set constant.

The Reynolds number, a dimensionless ratio, may be stated as follows:

$$C_r = \frac{DV\rho}{\mu} \tag{4}$$

where

D = chimney diameter, feet

V = velocity of hot gas, feet per second.

ρ = mass density of the chimney gas per cubic foot

μ = viscosity of the gas in pounds-second per square foot taken at the gas temperature.

In another form:

$$C_r = \frac{1\,27\,W}{D\mu g} = \frac{0\,0396\,W}{D\mu} \tag{5}$$

where

W = weight of gas passed per second
g = acceleration of gravity.

The value of μ for chimney gases is usually taken as that of air or nitrogen, and for the variation of μ with temperature, the Sutherland equation may be employed as follows, giving μ in pounds-second per square foot:

$$\mu = \mu_o \left[\frac{273 + C}{T_c + C}\right] \left[\frac{T_c}{273}\right]^{1\,5}$$

where

T_c = chimney gas temperature, degrees Centrigrade.
μ_o = gas viscosity at 0 C
C = constant for specific gas

Using International Critical Table values, for air $\mu_o = 35\,6 \times 10^{-8}$, $C = 124$, for nitrogen $\mu_o = 34\,5 \times 10^{-8}$; and $C = 110$

Values for the viscosity of air and of nitrogen (the principal component of chimney gases) for the different temperatures follow, in which the values given in pounds-second per square foot are to be multiplied by 10^{-8}

Temp F	300	400	500	600	700	800
Air	49 7	54.5	58.5	62.5	66 7	70 5
Nitrogen	47 7	52 2	56 0	59 8	63 5	67 0

Example 2. To determine the Reynolds number C_r for a flow of 118 lb gas per second up a 12 ft diameter chimney at a temperature of 500 F. The gas may be assumed to have the same viscosity as nitrogen at 500 F. Using Equation 5.

$$C_r = 0\,0396\,\frac{W}{D\mu} = \frac{0\,0396 \times 118}{12 \times 56.0 \times 10^{-8}} = 698{,}000$$

The variation of the friction factor f with the Reynolds number is shown in Fig. 5[1]. Three curves are shown: A, B, and C, where the choice of the friction factor curve depends on the relative surface roughness, and this for usual chimney construction may be selected by size since surface conditions in service are always undeterminant. For sizes up to 3 ft in diameter, Curve C may be used; from 3 to 6 ft, Curve B; and from 6 ft upwards, Curve A Thus for the previous example with C_r = 698,000 and 12 ft diameter, f would be taken from Curve A as 0.0039.

6. The *length of the friction duct* is the vertical distance between the bottom of the breeching opening and the top of the chimney. Ordinarily this distance is approximately equal to the height of the chimney above the grate level.

7. Assuming no air infiltration the *amount of gases flowing and being discharged* is, of course, equal to the amount of gases generated in the combustion chamber of the boiler. The total products of combustion in pounds per second for a grate fired boiler may be computed from the equation:

$$W = \frac{C_g G W_{tp}}{3600} \tag{6}$$

where

C_g = pounds of fuel burned per square foot of grate surface per hour.
G = total grate surface of boilers, square feet.
$C_g \times G$ = total weight of fuel burned per hour.
W_{tp} = total weight of products of combustion per pound of fuel.

A similar computation may be made in the case of gas, oil, or stoker-fired fuel.

[1]See also Flow of Fluids in Closed Circuits, by R. J. S. Pigott (*Mechanical Engineering*, August, 1933).

Fig. 6 is a typical chimney performance chart giving the available draft intensities for various amounts of gases flowing and sizes of chimney. This chart is based on an atmospheric temperature of 62 F, a chimney gas temperature of 500 F, a unit chimney gas weight of 0.09 lb per cubic foot, sea level atmospheric pressure, a coefficient of friction of 0.016, and a friction duct length equal to the height of the chimney above the grate level. These curves may be used for general operating conditions. For specific operating conditions, a new chart should be constructed from Equation 1.

It has been the usual custom, and still is to a lamentably great extent,

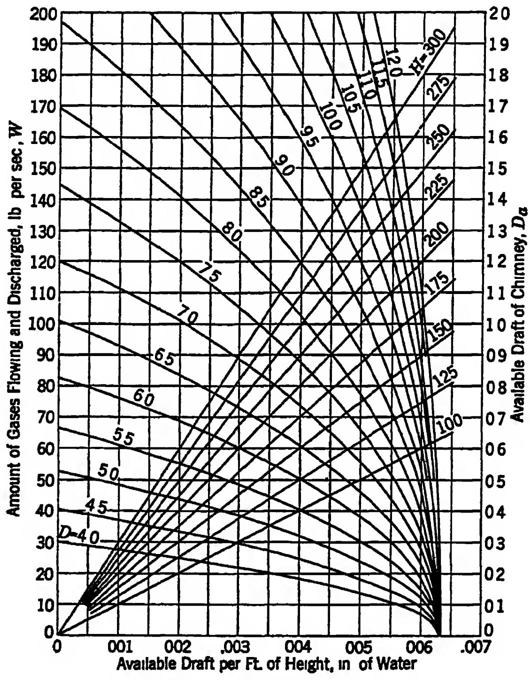


FIG. 6. CHIMNEY PERFORMANCE CHART[a]

[a]*To solve a typical example:* Proceed horizontally from a Weight Flow Rate point to intersection with diameter line; from this intersection follow vertically to chimney height line, from this intersection follow horizontally to the right to Available Draft scale Starting from a point of Available Draft, take steps in reverse order

to select the required size of a natural draft chimney from a table of chimney sizes based only on boiler horsepowers. After the ultimate horsepower of the projected plant had been determined, the chimney size in the table corresponding to this figure was then selected as the proper size required. Generally, no further attempt was made to determine if the height thus selected was sufficient to help create the required draft demanded by the entire installation, or the diameter sufficiently large to enable the chimney quickly, efficiently, and economically to dispose of the gases. Since the operating characteristics of a natural draft chimney are

similar in all respects to those of a centrifugal pump, or a centrifugal fan, it is no more possible to select a proper size chimney from such a table, even with correction factors appended, than it is to select the proper size pump from tables based only on the amount of water to be delivered.

DETERMINING CHIMNEY SIZES

The required diameter and height of a natural draft chimney are given by the following equations:

$$H = \frac{D_r}{2.96B_o\left(\frac{W_o}{T_o} - \frac{W_c}{T_c}\right) - \frac{0.184fW_cB_oV^2}{T_cD}} \tag{7}$$

$$D = 0.288\sqrt{\frac{WT_c}{B_oW_cV}} \tag{8}$$

where

H = required height of chimney above grate bar level, feet.
D = required minimum diameter of chimney, feet (constant for entire height).
V = chimney gas velocity, feet per second.
D_r = total required draft demanded by the entire installation outside of the chimney, inches of water

Equations 7 and 8 give the required size of a natural draft chimney with all of the operating factors taken into consideration. Values for all of the factors with the exception of the chimney gas velocity may be either observed or computed. It is, of course, necessary to assume an arbitrary value for the velocity in order to arrive at some definite size. For any one set of operating conditions there will be as many sizes of chimneys as there are values of reasonable velocities to assume. Of the number of sizes corresponding to the various assumed velocities, there is one size which will be least expensive. Since the cost of a chimney structure, regardless of the kind of material used in the construction, varies as the volume of material in the structure, the cost criterion then may be represented by the approximate equation:

$$Q = \pi tHD \tag{9}$$

where

Q = volume of material, cubic feet.
t = average wall thickness, feet.

For all practical purposes, the value of πt may be taken as a constant regardless of the size of the structure. Hence, in general, the volume, and consequently the cost, of a chimney structure may be based on the factor HD as a criterion. Therefore, the value of the chimney gas velocity which will result in the least value of HD for any one set of operating conditions will produce a structure which will be the most economical to use, because its cost will be least.

The problem at hand is to deduce an equation for the chimney gas velocity which will result in a combination of a height and a diameter whose product HD will be least. The solution is obtained by equating the

product of Equations 7 and 8 to HD, differentiating this product with respect to V and equating the resulting expression to zero. This procedure results in the following expression:

$$V_e = \left(\frac{0.772 T_c \left(\frac{W_o}{T_o} - \frac{W_c}{T_c} \right) \sqrt{\frac{W T_c}{B_o W_c}}}{f W_c} \right)^{2/5} \tag{10}$$

where V_e = economical chimney gas velocity, feet per second.

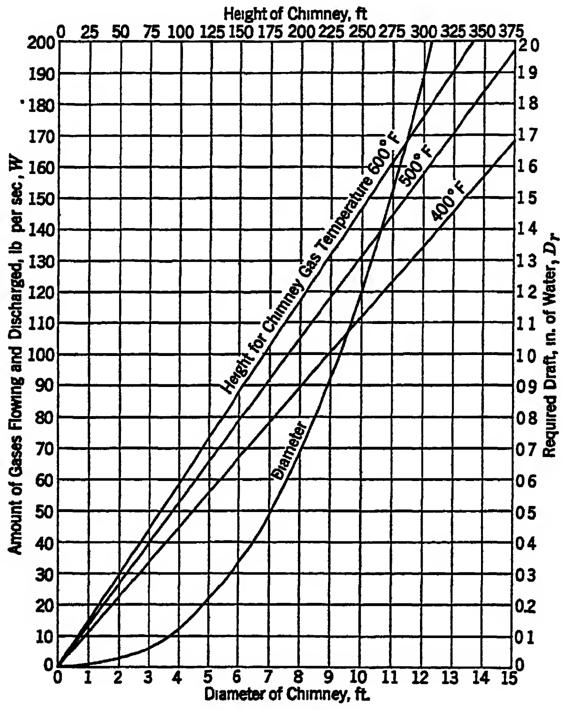


Fig. 7. Economical Chimney Sizes[a]

[a]Diameter values also for gas temperatures of 400, 500 and 600 F.

Equation 10 gives the economical velocity of the chimney gases for any set of operating conditions, and represents the velocity which will result in a chimney the size of which will cost less than that of any other size as determined by any other velocity for the same operating conditions. After the value of the economical velocity has been determined, the corresponding height and diameter can then be determined from Equations 7 and 8, respectively, and the economical size will then be attained. Equations 7, 8 and 10 may be simplified considerably for average operating conditions in an average size steam plant by assuming typical conditions.

Average chimney gas temperature, 500 F........................T_c = 960
Mean atmospheric temperature, 62 F..............................T_o = 522
Average coefficient of friction, 0 016...........................f = 0 016
Average chimney gas density, 0 09..............................W_c = 0 09
Sea level elevation, with barometer of 29 92...................B_o = 29 92

Substituting these values in Equations 10, 8 and 7, respectively, and reducing, the results are substantially:

$$V_e = 13.7W^{1/5} \tag{11}$$

$$D = 1\,5W^{2/5} \tag{12}$$

$$H = 190D_r \tag{13}$$

Fig. 7 gives the economical chimney sizes for various amounts of gases flowing and for required draft intensities as computed from Equations 11, 12 and 13. They are based on the operating factors used in reducing Equations 7, 8 and 10 to their simpler form. The sizes shown by the curves in the chart should be used for general operating conditions only, or for installations where the required data necessary for an exact determination are difficult or impossible to secure. Whenever it is possible to secure accurate data, or the anticipated operating conditions are fairly well known, the required size should be determined from Equations 7, 8 and 10. The recommended minimum inside dimensions and heights of chimneys for small and medium size installations are given in Table 1.

GENERAL EQUATION

The general draft equation for a steam producing plant may be stated as follows:

$$D_t - h_f = h_F + h_B + h_{Bd} + h_C + h_{Br} + h_V + h_O + h_E + h_R \tag{14}$$

where

D_t = theoretical draft intensity created by pressure transformer, inches of water.
h_f = draft loss due to friction in pressure transformer, inches of water.
h_F = draft loss through the fuel bed, inches of water.
h_B = draft loss through the boiler and setting, inches of water.
h_{Br} = draft loss through the breeching, inches of water.
h_V = draft loss due to velocity, inches of water.
h_{Bd} = draft loss due to bends, inches of water.
h_C = draft loss due to contraction of opening, inches of water.
h_O = draft loss due to enlargement of opening, inches of water.
h_E = draft loss through the economizer, inches of water.
h_R = draft loss through recuperators, regenerators, or air heaters, inches of water

The left hand member of Equation 14 represents the total amount of available draft created by the pressure transformer, that is, the natural draft chimney, Venturi chimney, or fan, and is equal to the theoretical intensity less the internal losses incidental to operation. The right hand member represents the sum of all of the various losses of draft throughout the entire boiler plant installation outside of the pressure transformer itself. The left hand member expresses the available intensity and is analogous to the head developed by a centrifugal pump in a water works system, while the right hand member expresses the required draft in-

tensity and is analogous to the total dynamic head in a water works system. For a general circulation of gases

$$D_a = D_r \tag{15}$$

where

D_a = available draft intensity, inches of water
D_r = required draft, inches of water.

The *draft loss through the fuel bed* (h_F), or the amount of draft required to effect a given or required rate of combustion, varies between wide limits and represents the greater portion of the required draft. In coal-fired

TABLE 1. RECOMMENDED MINIMUM CHIMNEY SIZES FOR HEATING BOILERS AND FURNACES[a]

Warm Air Furnace Capacity in Sq In of Leader Pipe	Steam Boiler Capacity Sq Ft of Radiation	Hot Water Heater Capacity Sq Ft of Radiation	Nominal Dimensions of Fire Clay Lining in Inches	Rectangular Flue		Round Flue		Height in Ft Above Grate
				Actual Inside Dimensions of Fire Clay Lining in Inches	Actual Area Sq In.	Inside Diameter of Lining in Inches	Actual Area Sq In	
790	590	973	8½x13	7 x11½	81			35
1000	690	1,140				10	79	
	900	1,490	13x13	11¼x11¼	127			
	900	1,490	8½x18	6¾x16¼	110			
	1,100	1,820				12	113	40
	1,700	2,800	13x18	11¼x16¼	183			
	1,940	3,200				15	177	
	2,130	3,520	18x18	15¾x15¾	248			
	2,480	4,090	20x20	17¼x17¼	298			45
	3,150	5,200				18	254	50
	4,300	7,100				20	314	
	4,600	7,590	20x24	17x21	357			
	5,000	8,250	24x24	21x21	441			55
	5,570	9,190		24x24[b]	576			60
	5,580	9,200				22	380	
	6,980	11,500				24	452	65
	7,270	12,000		24x28[b]	672			
	8,700	14,400		28x28[b]	784			
	9,380	15,500				27	573	
	10,150	16,750		30x30[b]	900			
	10,470	17,250		28x32[b]	896			

[a]This table is taken from the A.S.H.V.E. Code of Minimum Requirements for the Heating and Ventilation of Buildings (Edition of 1929).
[b]Dimensions are for unlined rectangular flues.

installations, the draft loss through the fuel bed is dependent upon the following factors: (1) character and condition of the fuel, clean or dirty; (2) percentage of ash in the fuel; (3) volume of interstices in the fuel bed, coarseness of fuel; (4) thickness of the fuel bed, rate of combustion; (5) type of grate or stoker used; (6) efficiency of combustion.

There is a certain intensity of draft with which the best results will be obtained for every kind of coal and rate of combustion. Fig. 8 gives the intensity of draft, or the vacuum in the combustion chamber required to burn various kinds of coal at various rates of combustion. Expressed in

other words, these curves represent the amount of draft required to force the necessary amount of air through the fuel bed in order to effect various rates of combustion. It will be noted that the amount of draft increases as the percentage of volatile matter diminishes, being comparatively low for the lower grades of bituminous coals and highest for the high grades and small sizes of anthracites. Also, when the interstices of the coal are large and the particles are not well broken up, as with bituminous coals, much less draft is required than when the particles are small and are well broken up, as with bituminous slack and the small sizes of anthracites. In general, the draft loss through the fuel bed increases as: (1) the percentage of volatile matter diminishes; (2) the percentage of fixed carbon

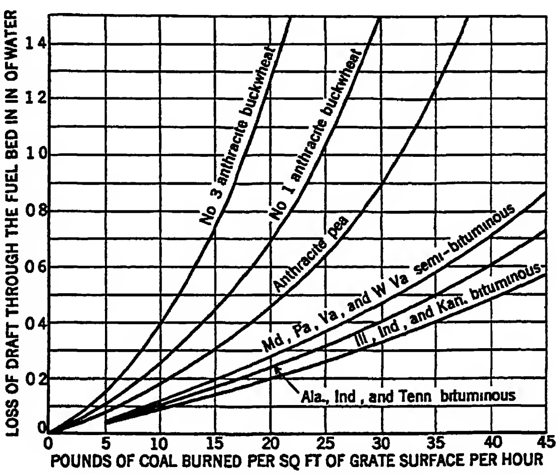


FIG 8. DRAFT REQUIRED AT DIFFERENT RATES OF COMBUSTION FOR VARIOUS KINDS OF COAL

increases; (3) the thickness of the bed increases; (4) the percentage of ash increases; (5) the volume of the interstices diminishes.

In making the preliminary assumptions for the draft loss through the fuel bed, due allowances should be made for a possible future change in the grade of fuel to be burned and also in the rate of combustion. A value should be selected for this loss which will represent not only the highest rate of combustion which will be encountered, but also the grade of coal which has the greatest resistance through the fuel bed and which may be burned at a later date.

In powdered-fuel and oil-fired installations, there will be no draft loss through the fuel bed since there is none and, consequently, this factor becomes zero in the general draft equation. All other factors being constant, the height of the chimney in installations of this character will be less than the height in coal-fired installations, and in the case of mechanical draft installations the driving units need not be as large since the head against which the fan is to operate is not as great in the former as in the latter.

The *draft loss through the boiler and setting* (h_B) also varies between wide limits and, in general, depends upon the following factors:

1. Type of boiler.
2. Size of boiler.
3. Rate of operation.
4. Arrangement of tubes.
5. Arrangement of baffles.
6. Type of grate.
7. Design of brickwork setting.
8. Excess air admitted.
9. Location of entrance into breeching.

Curves showing the draft loss through the boiler are usually based on the load or quantity of gases passing through the boiler, expressed in terms of percentage of normal rate of operation. Owing to the great variety of boilers of different designs and the various schemes of baffling, it is impossible to group together a set of curves for the draft loss through the boiler which may even be used generally. It is therefore necessary to secure this information from the manufacturer of the particular type of boiler and baffle arrangement under consideration.

When a boiler is installed and in operation, the draft loss depends upon the amount of gases flowing through it. This, in turn, depends upon the proportion of excess air admitted for combustion. Primarily, the amount of excess air is measured by the CO_2 content; the less the amount of CO_2, the greater the amount of excess air and hence the greater the draft loss.

The loss of draft through the boiler will vary directly as the size of the boiler and the length of the gas passages within. The loss also varies as the number of tubes high, but not in a direct ratio inasmuch as the loss due to the reversal of flow at the ends of the baffles remains constant regardless of the height of the boiler. The arrangement of the tubes, whether the gases flow parallel to or at right angles to the tubes, has an appreciable effect on the loss. The arrangement of the baffles influences the draft loss greatly, the loss through a boiler with five passes being greater than the loss through one of three or four passes. A poor design and a rough condition of the brickwork will increase the loss greatly, whereas a proper design and a smooth condition will keep the loss at a minimum. The loss through the boiler will be less when the breeching entrance is located at or near the top of the boiler than when it is located at or near the bottom since the gases have a shorter distance to travel in the former instance.

The *draft loss through the breeching* (h_{Br}) is given by the general equation:

$$h_{Br} = \frac{0.000194 W^2 T_c f L}{A^2 B_o W_c C_{br}} \tag{16}$$

where

W = the amount of gases flowing, pounds per second.
T_c = absolute temperature of breeching gases, degrees Fahrenheit.
f = coefficient of friction.
L = length of breeching, feet.
A = area of breeching, square feet.
B_o = atmospheric pressure corresponding to altitude, inches of mercury.
W_c = weight of a cubic foot of breeching gases at 0 F and sea level atmospheric pressure, pounds per cubic foot.
C_{br} = hydraulic radius of breeching section.

It has been the general custom to *lump off* the intensity of the breeching loss at 0.10 in. of water per 100 ft of breeching length regardless of its size or shape or the amount and temperature of the gases flowing through it. This practice is hazardous and has no more foundation in fact than that of determining the friction head in a water works system without taking into consideration the size of the pipe or the amount of water flowing through it. When the length of the breeching is relatively short, any variation in any one of the factors in the equation will have no appreciable effect on the draft loss. However, when the breeching is relatively long, the draft loss is affected greatly by the various factors, particularly by the size and shape as well as by the weight of gases flowing.

The *draft loss due to velocity* (h_V) is given by the equation

$$h_V = \frac{0.000194W^2T_c}{A^2B_0W_c} \tag{17}$$

and represents the amount of draft required to accelerate the gases from zero velocity to the velocity at which the gases are flowing, or in other words, from a static gas condition of zero flow to the amount of gases flowing throughout the installation. This loss corresponds to the velocity head in water works systems.

The *draft loss due to bends* (h_{Bd}) is equivalent to the loss due to the velocity head for a 90-deg bend. In changing direction of flow, the gas velocity decreases to zero with a loss of velocity head and then increases to its proper value at the expense of a loss in pressure head, the net result being a loss in pressure head equal to the velocity head at the bend. This loss is given by the equation:

$$h_{Bd} = \frac{0.000194W^2T_c}{A^2B_0W_c} \tag{18}$$

The friction at a right-angle bend is sometimes expressed as the equivalent of a straight length of flue of a certain length for a certain diameter, similar to the procedure used in estimating the loss due to bends in piping systems conducting water. Most flues, however, particularly breechings, are built square or rectangular in section and no general equation based on the shape of the flue can be conveniently expressed.

The *draft loss due to sudden contraction of an area* (h_C) is given by the equation:

$$h_C = \frac{0.000194K_cW^2T_c}{A_s^2B_0W_c} \tag{19}$$

where

K_c = coefficient of sudden contraction based on $\frac{A_s}{A_1}$, the ratio of the areas of the smaller to the larger section $= 0.5\left(1 - \frac{A_s}{A_1}\right)$

A_s = area of the smaller section.

When the flue or passage through which the gases flow is suddenly contracted, a considerable portion of the static head in the larger section

is converted into velocity head and a draft loss of some consequence, particularly in a short breeching, takes place. A sudden contraction should always be avoided where possible. At times, however, due to obstructions or limited head-room, it is necessary to alter the size of the breeching, but a sudden contraction may be avoided by gradually decreasing the area over a length of several feet.

The *draft loss due to a sudden enlargement of an area* (h_O) is given by the equation:

$$h_O = \frac{0.000194 K_O W^2 T_c}{A_s^2 B_O W_c} \tag{20}$$

where

K_0 = coefficient of sudden enlargement based on $\frac{A_s}{A_1}$, the ratio of the areas of the smaller to the larger section $= \left(1 - \frac{A_s}{A_1}\right)^2$

When the flue or passage through which the gases flow is suddenly enlarged, a portion of the velocity head is converted into static head in the larger section and, like the loss due to sudden contraction, a loss of some consequence, particularly in short breechings, takes place. A sudden enlargement in a breeching may be avoided by gradually increasing the area over a length of several feet. In large masonry chimneys, the area of the flue at the region of the breeching entrance is considerably larger than the area of the breeching at the chimney, and a sudden enlargement exists.

The *draft loss through the economizer* (h_E) should be obtained from the manufacturer but for general purposes it may be computed from the following general equation:

$$h_E = \frac{6.6 W_n^2 N T_c}{10^{12}} \tag{21}$$

where

W_n = pounds of gases flowing per hour per linear foot of pipe in each economizer section.

N = number of economizer sections.

An economizer in a steam plant affects the draft in two ways, (1) it offers a resistance to the flow of gases, and (2) it lowers the average chimney gas temperature, thereby decreasing the available intensity. In the case of a natural draft installation, both of these factors result in a relative increase in the height of the chimney and, in the case of a large plant, they may add as much as 20 or 30 ft to the height. The decrease in the temperature of the gases after they have passed through the economizer has an extremely important effect on the performance of a natural draft chimney and also upon the performance of a fan.

CONSTRUCTION DETAILS

For general data on the construction of chimneys reference should be made to the Standard Ordinance for Chimney Construction of the

National Board of Fire Underwriters. Briefly summarized, these provisions are as follows for heating boilers and furnaces:

The construction, location, height and area of the chimney to which a heating boiler or warm-air furnace is connected affect the operation of the entire heating system. Most residence chimneys are built of brick and may be either lined or unlined, but in either case the *walls must be air-tight* and there should be *only one smoke opening* into the chimney. Cleanout, if provided, must be *absolutely air-tight* when closed.

The walls of brick chimneys shall be not less than 3¾ in. thick (width of a standard size brick) and shall be lined with fire-clay flue lining. Fire-clay flue linings shall be manufactured from suitable refractory clay, either natural or compounded, and shall be adapted to withstand high temperatures and the action of flue gases. They shall be of standard commercial thickness, but not less than ⅝ in All fire-clay flue linings shall meet the standard specification of the *Eastern Clay Products Association.* The flue sections shall be set in special mortar, and shall have the joints struck smooth on the inside The masonry shall be built around each section of lining as it is placed, and all spaces between masonry and linings shall be completely filled with mortar. No broken flue lining shall be used Flue lining shall start at least 4 in below the bottom of smoke-pipe intakes of flues, and shall be continued the entire heights of the flues and project at least 4 in. above the chimney top to allow for a 2 in projection of lining. The wash or splay shall be formed of a rich cement mortar To improve the draft the wash surface should be concave wherever practical

Flue lining may be omitted in brick chimneys, provided the walls of the chimneys are not less than 8 in. thick, and that the inner course shall be a refractory clay brick. All brickwork shall be laid in spread mortar, with all joints push-filled. Exposed joints both inside and outside shall be struck smooth No plaster lining shall be permitted.

Chimneys shall extend at least 3 ft above flat roofs and 2 ft above the ridges of peak roofs when such flat roofs or peaks are within 30 ft of the chimney The chimney shall be high enough so that the wind from any direction shall not strike the top of the chimney from an angle above the horizontal. The chimney shall be properly capped with stone, terra cotta, concrete, cast-iron, or other approved material; but no such cap or coping shall decrease the flue area.

There shall be but one connection to the flue to which the boiler or furnace smoke-pipe is attached. The boiler or furnace smoke-pipe shall be thoroughly grouted into the chimney and shall not project beyond the inner surface of the flue lining.

The size or area of flue lining or of brick flue for warm-air furnaces depends on height of chimney and capacity of heating system For chimneys not less than 35 ft in height above grate line, the net internal dimensions of lining should be at least 7 x 11½ in. for a total leader pipe area up to 790 sq in Above 790 and up to 1,000 sq in of leader pipe area the lining should be at least 11¼ x 11¼ in inside In case of brick flues not less than 35 ft in height with no linings, the internal dimensions should be at least 8 x 12 in. up to 790 sq in of leader area, and at least 12 x 12 in for leader capacities up to 1,000 sq in. Chimneys under 35 ft in height are unsatisfactory in operation and hence should be avoided.

CHIMNEYS FOR GAS HEATING

The burning of gas differs from the burning of coal in that the force which supplies the air for combustion of the gas comes largely from the pressure of the gas in the supply pipe, whereas air is supplied to a bed of burning coal by the force of the chimney draft. If, with a coal-burning boiler, the draft is poor, or if the chimney is stopped, the fire is smothered and the combustion rate reduced. In a gas boiler or furnace such a condition would interfere with the combustion of the gas, but the gas would continue to pass to the burners and the resulting incomplete combustion would produce a dangerous condition. In order to prevent incomplete combustion from insufficient draft, all gas-fired boilers and furnaces should have a back-draft diverter in the flue connection to the chimney.

A study of a typical *back-draft diverter* shows that partial or complete

chimney stoppage will merely cause some of the products of combustion to be vented out into the boiler room, but will not interfere with combustion. In fact, gas-designed appliances must perform safely under such a condition to be approved by the *American Gas Association* Laboratory. Other functions of the back-draft diverter are to protect the burner and pilot from the effects of down-drafts, and to neutralize the effects of variable chimney drafts, thus maintaining the appliance efficiency at a

TABLE 2 SUGGESTED GENERAL DIMENSIONS FOR VERTICAL BACK-DRAFT DIVERTER

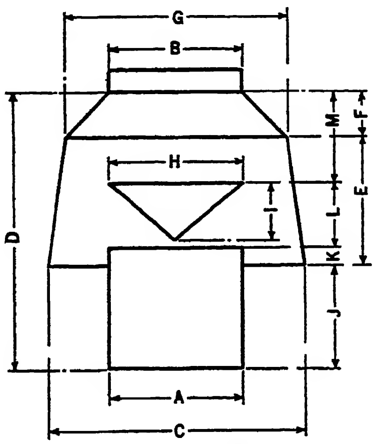


Table of Dimensions (In)

Pipe Size	A	B	C	D	E	F	G	H	I	J	K	L	M
3	3	3	5 5	7 0	3.8	0 7	4 4	3 0	1.5	2.5	0 7	1 5	2.3
4	4	4	7 2	9.5	5.0	1.0	6 0	4.0	2.0	3.5	1.0	2.0	3.0
5	5	5	9.4	10.8	5 3	1 5	8.0	5.0	2.3	4.0	0.9	2.4	3.5
6	6	6	11.5	12.0	5.6	1.9	9 8	6.0	2 5	4 5	0.8	2.7	4.0
7	7	7	13.5	13.9	6.4	2.3	11.6	7.0	2 9	5.3	0.9	3.1	4 6
8	8	8	15.5	15.8	7.1	2.7	13.4	8.0	3.2	6.0	1.0	3.5	5.3
9	9	9	17.5	17.5	7.7	3.1	15.2	9.0	3 5	6 7	1.0	4.0	5.8
10	10	10	19.7	18.8	7.9	3.6	17.2	10.0	3.8	7.3	1 0	4.3	6.2
11	11	11	22.2	20.7	8.4	4.3	19.6	11.0	4 1	8 0	1.5	4.6	6 6
12	12	12	24.7	22.2	8.7	5.0	22.0	12 0	4 4	8.5	1.7	5 0	7 0

substantially constant value. Converted boilers or furnaces, as well as gas-designed appliances, should be provided with back-draft diverters.

Since back-draft diverters have a special function to perform in protecting gas burning appliances, it is necessary that they should be built to the proper size as shown in Table 2. Work is now in progress on the development of a horizontal diverter for use where there is not enough room to install a vertical type of diverter. Information on the approved proportion of such equipment may be secured from the *American Gas Association* Testing Laboratory.

TABLE 3. MINIMUM ROUND CHIMNEY DIAMETERS FOR GAS APPLIANCES (INCHES)

HEIGHT OF CHIMNEY FEET	GAS CONSUMPTION IN THOUSANDS OF BTU PER HOUR								
	100	200	300	400	500	750	1000	1500	2000
20	4 50	5 70	6 60	7.30	8 00	9 40	10.50	12 35	13.85
40	4.25	5 50	6 40	7.10	7.80	9 15	10 25	12 10	13.55
60	4 10	5.35	6 20	6 90	7.60	8 90	10 00	11 85	13.25
80	4.00	5.20	6 00	6.70	7.35	8 65	9 75	11 50	12.85
100	3.90	5.00	5.90	6.50	7.20	8.40	9.40	11 00	12 40

As is the case with the complete combustion of almost all fuels, the products of combustion for gas are carbon dioxide (CO_2) and water vapor with just a trace of sulphur trioxide (SO_3). Sulphur usually burns to the trioxide in the presence of an iron oxide catalyst. The volume of water vapor in the flue products is about twice the volume of the carbon dioxide when coke oven or natural gas is burned. Because of the large quantity of water vapor which is formed by the burning of gas, it is quite important that all gas-fired central heating plants be connected to a chimney having a good draft. Lack of chimney draft causes stagnation of the products of combustion in the chimney and results in the condensation of a large amount of the water vapor. A good chimney draft draws air into the chimney through the openings in the back-draft diverter, lowers the dew point of the mixture, and reduces the tendency of the water vapor to condense.

The flue connections from a gas-fired boiler or furnace to the chimney should be of a non-corrosive material. In localities where the price of gas requires the use of highly efficient appliances, the material used for the flue connection not only should be resistant to the corrosion of water, but should resist the corrosion of dilute solutions of sulphur trioxide in water. Sheet aluminum, as well as some other materials, seems to serve this purpose very well.

When condensation in a chimney proves troublesome, it may be necessary to provide a drain to a dry well or sewer. The cause of the excessive condensation should be investigated and remedied if possible. This may be done by raising the flue temperature slightly or increasing the size of the back-draft diverter. The protection of unlined chimneys has been investigated and the results indicate that after the loose material has been removed, the spraying with a water emulsion of asphalt-chromate provides an excellent protection.

A chimney for a gas-fired boiler or furnace should be constructed in accordance with the principles applicable to other boilers. Where the wall forming a smoke flue is made up of less than an 8-in. thickness of brick, concrete, or stone, a burnt fire clay flue tile lining should be used. Care should be used that the lengths of flue tile meet properly with no openings at the joints. Cement mortar should be used for the entire chimney.

Table 3 gives the minimum cross-sectional diameters of round chim-

neys (in inches) for various amounts of heat supplied to the appliance, and for various chimney heights. This is in accordance with *American Gas Association* recommendations.

PROBLEMS IN PRACTICE

1 ● What are the principle factors influencing the intensity of natural draft?

The intensity of natural draft depends largely upon the height of chimney above the grate bar level and the temperature difference between the chimney gases and the atmosphere.

2 ● What two kinds of draft need be considered?

Natural draft caused by temperature differences, and artificial draft caused by mechanical forcing.

3 ● What is the effective height of a chimney?

The height from the grate level to the top of the chimney is the effective height in producing natural draft

4 ● What dual purpose does a tall chimney fulfill?

A tall chimney primarily creates the necessary draft to move the air required for the combustion process and to move the products of combustion, and secondarily it discharges the gases at a high elevation to prevent them from becoming a nuisance.

5 ● What is the direct influence of the height on the design of a chimney?

The immediate purpose of height is to provide that draft intensity under the conditions of chimney gas temperature such that it will be adequate to overcome all the frictional resistances of the installation, as well as to provide for the actual gas movement

6 ● Of what importance is chimney cross-sectional area in stack design?

The area should be as large as is economically feasible in order that the frictional loss for the chimney height should not destroy the effectiveness of the height-created draft in overcoming the necessary frictional resistances of the boiler and its flue connections

7 ● Of what importance is the Reynolds number in chimney design?

It permits the selection of a more specific value of the chimney friction factor, rather than a general one, to correspond with conditions of size, nature of the gas, rate of gas flow, and condition of the surface.

8 ● a. Name the principal advantages of natural draft.
b. Name the principal disadvantages of natural draft.

a. Simplicity, reliability, freedom from mechanical parts, low cost of maintenance, relatively long life, relatively low depreciation, operation with no power requirement.

b. Lack of flexibility, irregularity, dependence on surroundings, susceptibility to temperature changes.

9 ● How is mechanical draft created?

By forced draft, by induced-draft fans, or by a Venturi chimney

10 ● Distinguish between theoretical and available draft.

Theoretical draft is the difference in pressure inside and outside the base of a chimney when it is under operating temperatures but when there are no gases flowing. Available draft is less than theoretical draft by the friction loss due to the flow of gases through the chimney.

Chapter 27

FUELS AND COMBUSTION

Classification of Coal, Air for Combustion, Draft Required, Combustion of Anthracite, Firing Bituminous Coal, Burning Coke, Hand Firing, Classification and Use of Oil, Classification and Use of Gas

THE choice of fuel for heating is a question of economy, cleanliness, fuel availability, operation requirements, and control. The principal fuels to be considered are coal, oil, and gas.

CLASSIFICATION OF COALS

The complex composition of coal makes it difficult to classify it into clear-cut types. Its chemical composition is some indication but coals having the same chemical analysis may have distinctly different burning characteristics. Users are mainly interested in the available heat per pound of coal, in the handling and storing properties, and in the burning characteristics. A description of the relationship between the qualities of coals and these characteristics requires considerable space; a treatment applicable to heating boilers is given in *U. S. Bureau of Mines Bulletin* 276.

A classification of coals is given in Table 1, and a brief description of the kinds of fuels is given in the following paragraphs, but it should be recognized that there are no distinct lines of demarcation between the kinds, and that they graduate into each other:

Anthracite is a clean, dense, hard coal which creates very little dust in handling. It is comparatively hard to ignite but it burns freely when well started. It is non-caking, it burns uniformly and smokelessly with a short flame, and it requires little attention to the fuel beds between firings It is capable of giving a high efficiency in the common types of hand-fired furnaces.

Semi-anthracite has a higher volatile content than anthracite, it is not as hard and ignites somewhat more easily, otherwise its properties are similar to those of anthracite.

Semi-bituminous coal is soft and friable, and fines and dust are created by handling it. It ignites somewhat slowly and burns with a medium length of flame. Its caking properties increase as the volatile matter increases, but the coke formed is relatively weak. Having only half the volatile matter content of the more abundant bituminous coals it can be burned with less production of smoke, and it is sometimes called *smokeless coal*

The term *bituminous coal* covers a large range of coals and includes many types having distinctly different composition, properties and burning characteristics. The coals range from the high-grade bituminous coals of the East to the poorer coals of the West. Their caking properties range from coals which completely melt, to those from which the volatiles and tars are distilled without change of form, so that they are classed as non-caking or free-burning. Most bituminous coals are strong and non-friable enough to permit of the screened sizes being delivered free from fines. In general, they ignite

easily and burn freely, the length of flame varies with different coals, but it is long. Much smoke and soot are possible especially at low rates of burning

Sub-bituminous coals occur in the western states, they are high in moisture when mined and tend to break up as they dry or when exposed to the weather; they are liable to ignite spontaneously when piled or stored. They ignite easily and quickly and have a medium length flame, are non-caking and free-burning; the lumps tend to break into small pieces if poked; very little smoke and soot are formed.

Lignite is of woody structure, very high in moisture as mined, and of low heating value, it is clean to handle. It has a greater tendency than the sub-bituminous coals to disintegrate as it dries, and it also is more liable to spontaneous ignition. Freshly mined lignite, because of its high moisture, ignites slowly. It is non-caking The char left after the moisture and volatile matter are driven off burns very easily, like charcoal. The lumps tend to break up in the fuel bed and pieces of char falling into the ashpit continue to burn. Very little smoke or soot is formed

Coke is produced by the distillation of the volatile matter from coal. The type of coke depends on the coal or mixture of coals used, the temperatures and time of distillation and, to some extent, on the type of retort or oven, coke is also produced as a residue from the destructive distillation of oil

TABLE 1. CLASSIFICATION OF COALS BY RANK[f]

Legend: F.C. = Fixed Carbon. V.M = Volatile Matter. Btu = British thermal units.

CLASS	GROUP	LIMITS OF FIXED CARBON OR BTU MINERAL-MATTER-FREE BASIS	REQUISITE PHYSICAL PROPERTIES
I Anthracite	1. Meta-anthracite	Dry F C, 98 per cent or more (Dry V M, 2 per cent or less)	
	2. Anthracite	Dry F C, 92 per cent or more and less than 98 per cent (Dry V M, 8 per cent or less and more than 2 per cent)	
	3. Semi-anthracite	Dry F C., 86 per cent or more and less than 92 per cent (Dry V.M, 14 per cent or less and more than 8 per cent)	Non-agglutinating[a]
II Bituminous[c]	1. Low volatile bituminous coal	Dry F C, 77 per cent or more and less than 86 per cent (Dry V M, 23 per cent or less and more than 14 per cent)	
	2 Medium volatile bituminous coal	Dry F C, 69 per cent or more and less than 77 per cent (Dry V M, 31 per cent or less and more than 23 per cent)	
	3. High volatile *A* bituminous coal	Dry F C, less than 69 per cent (Dry V M, more than 31 per cent), and moist[b] Btu, 14,000[d] or more	
	4. High volatile *B* bituminous coal	Moist[b] Btu, 13,000 or more and less than 14,000[d]	
	5 High volatile *C* bituminous coal	Moist Btu, 11,000 or more and less than 13,000[d]	Either agglutinating or non-weathering[e]
III Sub-bituminous	1 Sub-bituminous *A* coal	Moist Btu, 11,000 or more and less than 13,000[d]	Both weathering and non-agglutinating
	2 Sub-bituminous *B* coal	Moist Btu 9500 or more and less than 11,000[d]	
	3. Sub-bituminous *C* coal	Moist Btu 8300 or more and less than 9500[d]	
IV Lignitic	1. Lignite	Moist Btu less than 8300	Consolidated
	2. Brown coal	Moist Btu less than 8300	Unconsolidated

[a]If agglutinating, classify in low-volatile group of the bituminous class

[b]Moist Btu refers to coal containing its natural bed moisture but not including visible water on the surface of the coal

[c]Pending the report of the Subcommittee on Origin and Composition and Methods of Analysis, it is recognized that there may be non-caking varieties in each group of the bituminous class.

[d]Coals having 69 per cent or more fixed carbon on the dry, mineral-matter-free basis shall be classified according to fixed carbon, regardless of Btu

[e]There are three varieties of coal in the High-volatile *C* bituminous coal group, namely, Variety 1, agglutinating and non-weathering; Variety 2, agglutinating and weathering, Variety 3, non-agglutinating and non-weathering

[f]Adapted from A.S.T M Standards on Coal and Coke, p 68, *American Society for Testing Materials*, Philadelphia, 1934

High-temperature cokes Coke as usually available is of the high-temperature type, and contains between 1 and 2 per cent volatile matter. High-temperature cokes are subdivided into *beehive coke* of which comparatively little is now sold for domestic use, *by-product coke*, which covers the greater part of the coke sold, and *gas-house coke*. The differences among these three cokes are relatively small, their denseness and hardness decrease and friability increases in the order named. In general, the lighter and more friable cokes ignite and burn the more easily.

Low-temperature cokes are produced at low coking temperatures, and only a portion of the volatile matter is distilled off. Cokes as made by various processes under development have contained from 10 to 15 per cent volatile matter. In general, these cokes ignite and burn more readily than high-temperature cokes. The properties of various low-temperature cokes may differ more than those of the various high-temperature cokes because of the differences in the quantities of volatile matter and because some may be light and others briquetted.

The sale of *petroleum cokes* for domestic furnaces has been small and is generally confined to the Middle West. They vary in the amount of volatile matter they contain, but all have the common property of a very low ash content, which necessitates the use of refractory pieces to protect the grates from being burned.

In order to obtain perfect combustion a definite amount of air is required for each pound of fuel fired. A deficiency of air supply will result in combustible products passing to the stack unburned. An excess of air absorbs heat from the products of combustion and results in a greater loss of sensible heat to the stack.

Total Air Required. The theoretical amount of air required per pound of fuel for perfect combustion is dependent upon the analysis of the fuel;

TABLE 2. POUNDS OF AIR PER POUND OF FUEL AS FIRED

ANTHRACITE	COKE	SEMI-BITUMINOUS	BITUMINOUS	LIGNITE
9.6	11 2	11.2	10 3	6 2

however, for estimating purposes the theoretical air required for different grades of fuel may roughly be taken from Table 2. An excess of about 50 per cent over the theoretical amount is considered good practice under usual operating conditions.

The amount of excess air, based upon the laws of combustion, can be determined by its relation to the percentage of CO_2 (carbon dioxide) in the products of combustion. This relationship is shown by the curves (Fig. 1) for high and low volatile coals and for coke. In hand-fired furnaces with long periods between firings the combustion goes through a cycle in each period and the quantity of excess air present varies.

Secondary Air. The division of the total into primary and secondary air necessary to produce the same rate of burning and the same excess air depends on a number of factors which include size of fuel, depth of fuel bed, and diameter of firepot. The ratio of the secondary to the primary air increases with decrease in the size of the fuel pieces, with increase in the depth of the fuel bed, and with increase in the area of the firepot; the ratio also increases with increase in rate of burning.

Size of the fuel is a very important factor in fixing the quantity of secondary air required for non-caking coals. With caking coals it is not

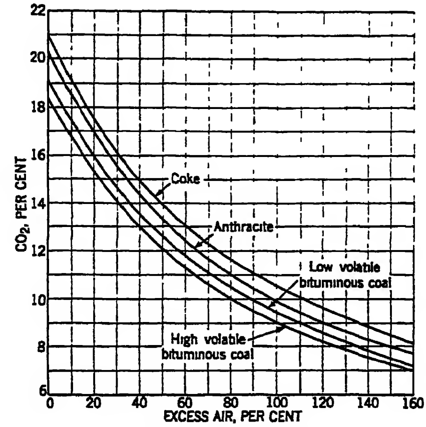
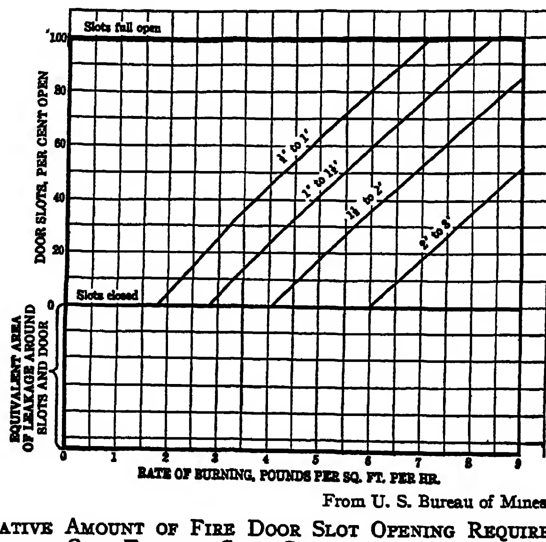


Fig. 1. Relation Between CO_2 and Excess Air in Gases of Combustion

so important because small pieces fuse together and form large lumps. Fortunately a smaller size fuel gives more resistance to air flow through the fuel bed and thus automatically causes a larger draft above the fuel bed, which draws in more secondary air through the same slot openings. In spite of this, a small size fuel requires a larger opening of the door slots; for a certain size for each fuel no slot opening is required, and for larger sizes too much excess air gets through the fuel bed.

It is impossible to establish a single rule for the correct slot opening for all types and sizes of fuels and for all rates of burning. Furthermore, the



From U. S. Bureau of Mines

Fig. 2. Relative Amount of Fire Door Slot Opening Required in a Given Furnace to Give Equally Good Combustion for High Temperature Coke of Various Sizes When Burned at Various Rates

size of slot opening is dependent on whether the ashpit damper is open or closed. It is better to have too much than too little secondary air; the opening is too small if there is a puff of flame when the firing door is opened.

Fig. 2, taken from the *U. S. Bureau of Mines Report of Investigations* No. 2980, shows the relationship of the slot opening, for a domestic furnace, to the size of coke and the rate of burning; these openings are with the ashpit damper wide open, and would be less if the available draft permits of its being partly closed. The same openings are satisfactory for anthracite.

Bituminous coals require a large amount of secondary air during the period subsequent to a firing in order to consume the gases and to reduce the smoke. The smoke produced is a good indicator, and that opening is best which reduces the smoke to a minimum. Too much secondary air will cool the gases below the ignition point, and prove harmful instead of beneficial. The following suggestions will be helpful:

1. In cold weather, with high combustion rates, the secondary air damper should be half open all the time.

2. In very mild weather, with a very low combustion rate, the secondary air damper should be closed all the time.

3. For temperatures between very mild and very cold, the secondary air damper should be in an intermediate position

4. For ordinary house operation, secondary air is needed after each firing for about one hour.

Draft Requirements

The draft required to effect a given rate of burning the fuel as measured at the smokehood is dependent on the following factors:

1. Kind and size of fuel.
2. Combustion rate per square foot of grate area per hour.
3. Thickness of fuel bed
4. Type and amount of ash and clinker accumulation.
5. Amount of excess air present in the gases.
6. Resistance offered by the boiler passes to the flow of the gases.
7. Accumulation of soot in the passes.

Insufficient draft will necessitate additional manipulation of the fuel bed and more frequent cleanings to keep its resistance down. Insufficient draft also restricts the control by adjustment of the dampers.

The quantity of excess air present has a marked effect on the draft required to produce a given rate of burning, and it is often possible to produce a higher rate by increasing the thickness of the fuel bed.

Combustion of Anthracite[1]

An anthracite fire should never be poked, as this serves to bring ash to the surface of the fuel bed where it melts into clinker.

Egg size is suitable for large firepots (grates 24 in. and over) if the fuel

[1]See reports published by *The Anthracite Institute Laboratory*, Primos, Pennsylvania

can be fired at least 16 in. deep. The air spaces between the pieces of coal are large, and for best results this coal should be fired deeply.

Stove size coal is the proper size of anthracite for many boilers and furnaces used for heating buildings. It burns well on grates at least 16 in. in diameter and 12 in. deep. The only instructions needed for burning this type of fuel are that the grate should be shaken daily, the fire should never be poked or disturbed, and the fuel should be fired deeply and uniformly.

Chestnut size coal is in demand for firepots up to 20 in. in diameter, with a depth of from 10 to 15 in.

Pea size coal is often an economical fuel to burn. It is relatively low in price. When fired carefully, pea coal can be burned on standard grates. It is well to have a small amount of a larger fuel on hand when building new fires, or when filling holes in the fuel bed. Care should be taken to shake the grates only until the first bright coals begin to fall through the grates. The fuel bed, after a new fire has been built, should be increased in thickness by the addition of small charges until it is at least level with the sill of the fire door. This keeps a bed of ignited coal in readiness against the time when a sudden demand for heat shall be made on the heater.

Pea size coal requires a strong draft and therefore the best results generally will be obtained by keeping the choke damper open, the cold-air check closed, and by controlling the fire with the air-inlet damper only. Pea size can also be fired in layers with stove or egg size anthracite and its use in this manner will reduce the fuel costs and attention required.

Buckwheat size coal for best results requires more attention than pea size coal, and in addition the smaller size of the fuel makes it more difficult to burn on ordinary grates. Greater care must be taken in shaking the grates than with pea coal on account of the danger of the fuel falling through the grate. In house heating furnaces the coal should be fired lightly and more frequently than pea coal. When banking a buckwheat coal fire it is advisable after coaling to expose a small spot of hot fire by putting a poker down through the bed of fresh coal. This will serve to ignite the gas that will be distilled from the fresh coal and prevent an explosion of gas within the firepot, which in some cases depending upon the thickness of the bed of fresh coal is severe enough to blow open the doors and dampers of the furnace. A good draft is required and consequently the fire is best controlled by the air-inlet damper only. Where frequent attention can be given and care exercised in manipulation of the grates this fuel can be burned satisfactorily without the aid of any special equipment.

In general it will be found more satisfactory with buckwheat coal to maintain a uniform heat output and consequently to keep the system warm all the time, rather than to allow the system to cool off at times and then to attempt to burn the fuel at a high rate while warming up. A uniform low fire will minimize the clinker formation and keep the clinker in an easily broken up condition so that it readily can be shaken through the grate.

Forced draft and special grates or retorts frequently are used with this fuel for best results.

No. 2 buckwheat anthracite, or rice size, is used only with forced draft equipment on mechanical stokers. No. 3 buckwheat anthracite, or barley, has no application in domestic heating.

Firing Bituminous Coal

Bituminous coal should never be fired over the entire fuel bed at one time. A portion of the glowing fuel should always be left exposed to ignite the gases leaving the fresh charge.

Air should be admitted over the fire through a special secondary air device, or through a slide in the fire door or by opening the fire door slightly. If the quantity of air admitted is too great the gases will be cooled below the ignition temperature and will fail to burn. The fireman can judge the quantity of air to admit by noting when the air supplied is just sufficient to make the gases burn rapidly and smokelessly above the fuel bed.

The red fuel in the firebox, before firing, excepting only a shallow layer of coke on the grate, should be pushed to one side or forward or backward to form a hollow in which to throw the fresh fuel. Some manufacturers recommend that all red fuel be pushed to the rear of the firebox and that the fresh fuel be fired directly on the grate and allowed to ignite from the top. The object of this is to reduce the early rapid distillation of gases and to reduce the quantity of secondary air required for smokeless combustion.

It is well to have the bright fuel in the firebox so placed that the gases from the freshly fired fuel, mixed with the air over the fuel bed, pass over the bed of bright fuel on the way to the flues. The bed of bright fuel then supplies the heat to raise the mixture of air and gas to the ignition temperature, thereby causing the gaseous matter to burn and preventing the formation of smoke.

The fuel bed should be carried as deep as the size of fuel and the available draft permit, in order to have as much coked fuel as possible for pushing to the rear of the firebox at the time of firing. A deep fuel bed allows the longest firing intervals.

If the coal is of the caking kind the fresh charge will fuse into one solid mass which can be broken up with the stoking bar and leveled from 20 min to one hour after firing, depending on the temperature of the firebox. Care should be exercised when stoking not to bring the bar up to the surface of the fuel as this will tend to bring ash into the high temperature zone at the top of the fire, where it will melt and form clinker. The stoking bar should be kept as near the grate as possible and should be raised only enough to break up the fuel. With fuels requiring stoking it may not be necessary to shake the grates, as the ash is usually dislodged during stoking.

The output obtained from any heater with bituminous coal will usually exceed that obtainable with anthracite, since soft coal burns more rapidly than hard coal and with less draft. Soft coal, however, will require frequent attention to the fuel bed, because it burns unevenly, even though the fuel bed may be level, forming holes in the fire which admit too much air, chilling the gases over the fuel bed and reducing the available draft.

Semi-bituminous coal is fired as bituminous coal, and because of its caking characteristics it requires practically the same attention. The *Pocahontas Operators Association* recommends the central cone method of firing, in which the coal is heaped on to the center of the bed forming a cone the top of which should be level with the middle of the firing door. This allows the larger lumps to fall to the sides, and the fines to remain in the center and be coked. The poking should be limited to breaking down the coke without stirring, and to gently rocking the grates. It is recommended that the slides in the firing door be kept closed, as the thinner fuel bed around the sides allows enough air to get through.

Burning Coke

Coke is a very desirable fuel and usually will give satisfaction as soon as the user learns how to control the fire. Coke ignites and burns very rapidly with less draft than anthracite coal. In order to control the air admitted to the fuel it is very important that all openings or leaks into the ashpit be closed tightly. A coke fire responds more rapidly than an anthracite fire to the opening of the dampers. This is an advantage in warming up the system, but it also makes it necessary to watch the dampers more closely in order to prevent the fire from burning too rapidly. A deep fuel bed always should be maintained when burning coke. The grates should be shaken only slightly in mild weather and should be shaken only until the first red particles drop from the grates in cold weather. Since coke weighs only about half as much as anthracite per cubic foot only about half as much can be put in the firepot, so it will be necessary to fire oftener. The best size of coke for general use, for small firepots where the fuel depth is not over 20 in., is that which passes over a 1 in. screen and through a 1½ in. screen. For large firepots where the fuel can be fired over 20 in. deep, coke which passes over a 1 in. screen and through a 3 in. screen can be used, but a coke of uniform size is always more satisfactory. Large sizes of coke should be either mixed with fine sizes or broken up before using.

Dustless Coal

The practice of treating the more friable coals to allay the dust they create is increasing. The coal is sprayed with a solution of calcium chloride or a mixture of calcium and magnesium chlorides. Both these salts are very hygroscopic and their moisture under normal atmospheric conditions keeps the surface of the coal damp, thus reducing the dust during delivery and in the cellar, and obviating the necessity of sprinkling the coal in the bin.

The coal is sometimes treated at the mine, but more usually by the local distributor just before delivery. The solution is sprayed under high pressure, using from 2 to 4 gal or from 5 to 10 lb of the salt per ton of coal, depending on its friability and size.

Pulverized Coal

Installations of pulverized coal burning plants in heating boilers are of the unit type, in which the pulverized coal is delivered into the furnace immediately after grinding, together with the proper amount of preheated

air. With this apparatus, where the necessary furnace volume is obtainable, high efficiencies can be obtained.

A 150-hp boiler has generally been considered the smallest size for which pulverized fuel is feasible. Complications are introduced if an installation with a single boiler has to take care of very light loads.

Hand Firing

Hand firing is the oldest and the most widely used method of burning coal for heating purposes. To keep the fuel bed in proper condition where hand firing is used, the following general rules should be observed:

1. Remove ash from fuel bed by shaking the grates whenever fresh fuel is fired. This removes ash from the fire, enables the air to reach the fuel, and does away with the formation of clinker which is melted ash

2. Supply the boiler with a deep bed of fuel. Nothing is gained by attempting to fire a small amount of fuel A deep bed of fuel secures the most economical results.

3. Remove ash from ashpit at least once daily. Never allow ash to accumulate up to the grates. If the ash prevents the air from passing through, the grate bars will burn out and much clinker trouble will be experienced.

The principal requirements for a *hand-fired furnace* are that it shall have enough grate area and combustion space. The amount of grate area required is dependent upon the desired combustion rate.

The furnace volume is influenced by the kind of coal used. Bituminous coals, on account of their long-flaming characteristic, require more space in which to burn the gases of combustion completely than do the coals low in volatile matter. For burning high volatile coals provision should be made for mixing the combustible gases thoroughly so that combustion is complete before the gases come in contact with the relatively cool heating surfaces. An abrupt change in the direction of flow tends to mix the gases of combustion more thoroughly.

CLASSIFICATION OF OILS

Uniform oil specifications were prepared in 1929 by the *American Oil Burner Association*, in cooperation with the *American Petroleum Institute*, the *U. S. Bureau of Standards*, the *American Society for Testing Materials* and other interested organizations. Oil fuels were classified into six groups, as indicated by Table 3. When these specifications were prepared, it was generally accepted that the first three grades were adapted to domestic use, while the last three were suitable only for commercial and industrial burners.

Today domestic installations may use No. 4 oil of the so-called heavy oil group, when and if said oil very closely follows the specifications of No. 3. Up-to-date listing by the *Underwriter's Laboratories* should be referred to before a No. 4 grade of fuel is used which merely meets *Commercial Standards CS 12-35*.

Since the specifications as originally drawn provide for maximum limits only for the several grades, this differentiation has not proved stable. Realizing how unsatisfactory it is to have specifications which permit the substitution of one grade for another, the *U. S. Bureau of Standards* in cooperation with the *American Society for Testing Materials* is figuring

TABLE 3 COMMERCIAL STANDARD FUEL OIL SPECIFICATIONS[a]

A. Detailed Requirements for Domestic Fuel Oils

Grade of Oil	Approx Btu per Gal [b]	Flash Point Min	Flash Point Max.	Water and Sediment, Maximum	Pour Point,[c] Maximum	Distillation Test		Viscosity Maximum
No. 1 Domestic Fuel Oil A light distillate oil for use in burners requiring a high grade fuel	139,000	110 F or legal	165 F	0 05%	15 F	10% point, maximum 420 F	End point, maximum 600 F	
No 2 Domestic Fuel Oil A medium distillate oil for use in burners requiring a high grade fuel	141,000	125 F or legal	190 F	0 05%	15 F	10% point, maximum 440 F	90% point, maximum 620 F	
No. 3 Domestic Fuel Oil A distillate fuel oil for use in burners where a low viscosity oil is required.	143,400	150 F or legal	200 F	0 1%	15 F	10% point, maximum 460 F	90% point, maximum 675 F	Saybolt Universal at 100 F 55 seconds

B. Detailed Requirements for Industrial Fuel Oils

Grade of Oil	Approx. Btu per Gal.[b]	Flash Point, Min. Max.	Water and Sediment, Maximum	Pour Point,[c] Maximum	Viscosity, Maximum
No. 4 Industrial Fuel Oil An oil known to the trade as a light fuel oil for use in burners where a low viscosity industrial fuel oil is required.	144,500	150 F See Note[d]	1 0%	See Note[e]	Saybolt Universal at 100 F 125 seconds
No. 5 Industrial Fuel Oil Same as Federal Specifications Board specification for bunker oil "B" for burners adapted to the use of industrial fuel oil of medium viscosity.	146,000	150 F	1.0%		Saybolt Furol at 122 F 100 seconds
No 6 Industrial Fuel Oil Same as Federal Specifications Board specification for bunker oil "C" for burners adapted to oil of high viscosity.	150,000	150 F	Water sediment 1.75% 0 25%		Saybolt Furol at 122 F 300 seconds

[a]Adapted from "Fuel Oils," p 2, *U. S. Department of Commerce, Bureau of Standards, Commercial Standard CS12-33*, Washington, 1933.

[b]Government specifications do not give Btu per gallon, but they are noted here for information only.

[c]Lower or higher pour points may be specified whenever required by conditions of storage and use. However, these specifications shall not require a pour point less than 0 F under any conditions

[d]Whenever required, as for example in burners with automatic ignition, a maximum flash point may be specified However, these specifications shall not require a flash point less than 250 F under any conditions

[e]Pour point may be specified whenever required by conditions of storage and use However, these specifications shall not require a pour point less than 15 F under any conditions

on a new set of specifications providing for definite limits for each grade. When these specifications are adopted, it is expected that the *National Board of Fire Underwriters* will retest all burners using oils of the maximum specifications for the grade so that if a burner is approved for a certain grade it will burn any oil meeting the specifications for that particular grade.

Several burners adapted to industrial use have recently been listed for automatic operation with No. 5 oil. Usually oils No. 5 or 6 require preheating for proper operation, but where conditions are favorable, No. 5 can be used without the equipment that this entails.

There are two reasons for the trend to lower grades of oil. While the lighter oils contain slightly more heat units per pound, the weight per gallon increases more rapidly than the decrease in heat units per pound, and oil is bought by the gallon. As a consequence, while a No. 1 oil may contain 139,000 Btu per gallon, oil No. 5 may test 146,000 Btu per gallon, or 6 per cent more. Usually there is a differential of 3 to 4 cents between the No. 1 and No. 5 oils, so that the economy of buying the heavier fuels is apparent; there remains the economic utilization of the heat content of the heavier oils.

The cost of oil fuel is dependent also upon the amount that can be delivered at one time, and the method of delivery. Common practice has split the tank of the truck delivering oils for domestic use into compartments of 150 to 500-gal capacity, and these *unit dumps* are made the basis of price. Where a truck can be connected to a storage-tank *fill* and quickly discharge its oil by pump, the price obviously can be less than where a smaller quantity must be drawn off in 5-gal cans and poured. For similar reasons an installation that can be supplied from a tank car on a siding provides for a lower unit fuel cost than one where the oil must be trucked, even in the large trucks holding 2,000 gal or more that are used for distributing the heavier oils.

GAS CLASSIFICATION

Gas is broadly classified as being either *natural* or *manufactured.* Natural gas is a mechanical mixture of several combustible and inert gases rather than a chemical compound. Manufactured gas as distributed is usually a combination of certain proportions of gases produced by two or more processes, and is often designated as *city gas.*

When gas is burned a large amount of water vapor is produced as one of the products of combustion. This ordinarily escapes up the chimney, carrying away with it a certain amount of heat. However, when the heat value of gas is determined in an ordinary calorimeter, this water vapor is condensed and the latent heat of vaporization that is given up during the condensation is reported as a portion of the heat value of the gas. The heat value so determined is termed the *gross* or *higher* heat value and this is what is ordinarily meant when the heat value of gas is specified. The heat that is reclaimed by the condensation of the water vapor amounts to about 10 per cent of the total heat value. It is impractical to utilize the entire higher heat value of the gas in any house-heating appliance, because to do so it would be necessary to cool the products of

combustion down below their dew point, which is ordinarily in the neighborhood of 130 F.

The actual dew point in the chimney is different from the theoretical value because excess air is admitted not only at the burner but also at the backdraft diverter which lowers the dew point.

Natural gas is the richest of the gases and contains from 80 to 95 per cent methane, with small percentages of the other combustible hydrocarbons. In addition, it contains from 0.5 to 5.0 per cent of CO_2, and from 1 to 12 or 14 per cent of nitrogen. The heat value varies from 700 to 1,500 Btu per cubic foot, the majority of natural gases averaging about 1,000 Btu per cubic foot. Table 4 shows typical values for the four main oil fields, although values from any one field vary materially.

Table 4 also gives the calorific values of the more common types of manufactured gas. Most states have legislation which controls the distribution of gas and fixes a minimum limit to its heat content. The gross or higher calorific value usually ranges between 520 and 545 Btu per cubic foot, with an average of 535. A given heat value may be maintained and yet leave considerable latitude in the composition of the gas so that as distributed the composition is not necessarily the same in different districts, nor at successive times in the same district. There are limits to the

TABLE 4. REPRESENTATIVE PROPERTIES OF GASEOUS FUELS, BASED ON GAS AT 60 F AND 30 IN HG

Gas	Btu per Cu Ft		Specific Gravity, Air = 1 00	Air Required for Combustion, (Cu Ft)	Products of Combustion				Theoretical Flame Temperature, (Deg Fahr)
					Cubic Feet			Ultimate CO_2 Dry Basis	
	High (Gross)	Low (Net)			CO_2	H_2O	Total with N_2		
Natural gas—California	1200	1087	0.67	11.26	1.24	2.24	12.4	12 2	3610
Natural gas—Mid-Continental	967	873	0.57	9 17	0.97	1 92	10 2	11.7	3580
Natural gas—Ohio	1130	1025	0.65	10 70	1 17	2.16	11.8	12 1	3600
Natural gas—Pennsylvania	1232	1120	0 71	11 70	1.30	2 29	12.9	12.3	3620
Retort coal gas	575	510	0.42	5.00	0.50	1.21	5 7	11.2	3665
Coke oven gas	588	521	0.42	5.19	0.51	1 25	5 9	11.0	3660
Carburetted water gas	536	496	0.65	4.37	0.74	0 75	5.0	17.2	3815
Blue water gas	308	281	0.53	2.26	0.46	0.51	2 8	22.3	3800
Anthracite producer gas	134	124	0.85	1 05	0.33	0.19	1.9	19 0	3000
Bituminous producer gas	150	140	0.86	1.24	0 35	0.19	2.0	19.0	3160
Oil gas	575	510	0.35	4 91	0.47	1.21	5.6	10 7	3725

variation allowable, because the specific gravity of the gas depends on its composition, and too great a change in the specific gravity necessitates a change in the adjustment of the burners of small appliances.

Table 4 shows that a large proportion of the products of combustion when gas is burned may consist of water vapor, and that the greater the proportion of water vapor, the lower the maximum attainable CO_2 by gas analysis. The table also shows that a low calorific value does not necessarily mean a low flame temperature since, for example, natural gas has a theoretical flame temperature of 3600 F and blue water gas of 3800 F, although it has a calorific value less than one third that of natural gas.

The quantity of air given in Table 4 is that required for theoretical combustion, but with a properly designed and installed burner the excess air can be kept low. The division of the air into primary and secondary is a matter of burner design and the pressure of gas available, and also of the type of flame desired.

PROBLEMS IN PRACTICE

1 ● Differentiate between the general characteristics of hard and soft coals.

Hard coals contain fixed carbon in large proportions and in addition more ash is present especially in the smaller sizes Soft coals have an increasing percentage of carbon in combination with hydrogen which is volatile and will distill off under high temperature, producing smoke.

2 ● Name several important properties of coal from a utilization standpoint.

a. Caking tendency, whether none, weak, or strong

b. Quantity of volatile matter.

c. Friability.

d Fusibility of the ash.

3 ● What are the main data commonly available that fix the qualities of coal, and do these tell the whole story?

a. Calorific value, Btu per pound.

b. Proximate analysis giving percentages of moisture, volatile matter, fixed carbon, ash, and sulphur.

c. Temperature at which the ash softens.

d. Screen sizes.

Other important qualities not usually given are the friability of the coal, its caking tendency, and the qualities of the volatile matter The percentage of ash and its fusion temperature do not tell how the ash is distributed or how much of it is less fusible lumps of slate or shale

4 ● Are there available complete and sufficient data on gas and oils to fix their burning properties and furnace requirements?

Yes. Because gas and oils are of simple and uniform composition, data are available to fix their burning properties and furnace requirements, but the ability to control their combustion is somewhat less determinable.

5 ● What effect does moisture in fuels have on their efficiency?

With any solid fuel, latent and sensible heat are lost at the stack when moisture is dried out of the fuel in burning, and when its hydrogen is burned. Therefore, such fuels as sub-bituminous coal and lignite, which are high in moisture content, have a low efficiency. However, these efficiencies may be improved if the stack gases are cooled to room temperature, by heating the feed water, for example.

6 ● What are the advantages of a sized fuel for heating furnaces?

Because a sized fuel encourages a more uniform flow of air through the bed, the burning will be more uniform, and the bed will be less liable to develop holes and will require less attention. Uniformity of fuel size is more desirable as the area of the bed becomes smaller; it is less important with fuels that cake, but with sized fuels the caking will be more uniform and the air flow through the bed will be steadier In addition, ash and pieces of slate are less likely to be segregated and to form lumps of clinker.

7 ● Does the size of a fuel affect the quantity of air required to burn it at a given rate?

The total air required to give the same gas analysis at the stack is independent of the size of the fuel burned, but for non-caking fuels the ratio of the air passing through the fuel bed to the total air entering the burner base decreases, for the same thickness of bed, as the size of the fuel becomes smaller; this decrease is very rapid for sizes less than one inch. For coals that cake, this ratio will depend on the way the caked bed is broken up and on the size of the resulting pieces

8 ● Is the volatile matter which is given off when coals are burned of the same nature in all coals?

No. The products given off by coals when they are heated differ materially in the ratios by weight of the gases to the oils and tars. No heavy oils or tars are given off by anthracite, and very small quantities are given off by semi-anthracite. As the volatile matter in the coal increases to as much as 40 per cent of ash-free and moisture-free coal, increasing amounts of oils and tars are given up. For coals of higher volatile content, the relative quantity of oils and tars decreases, so it is low in the sub-bituminous coals and in lignite.

9 ● Is smoke a primary product in the burning of fuels?

Visible smoke may include very small particles of carbon, oil, tar, water (condensed steam), and ash. Of these, the oils, tars, and ash are mainly primary products, and the water is partly primary. The carbon, which usually comprises the greater part of the smoke, results from the breaking up by heat of oils, tars, and such gases as methane, so it may be considered a secondary product.

10 ● Is the sulphur in coals detrimental to combustion?

Not so far as is known, but its complete combustion gives only 25 per cent as much heat as is given by the same weight of carbon Sulphur is undesirable because it causes corrosion of flues and stacks, and also because its gases pollute the atmosphere, and damage buildings and vegetation.

11 ● How do deposits of soot on the surfaces of a boiler or heater affect the quantity of fuel burned?

There are two effects. The soot acts as an insulating layer over the surface and reduces the heat transmission to the water or air, the *Bureau of Mines Report of Investigations* No. 3272 shows that the loss of seasonal efficiency is not as great as has been believed and should not be over 6 per cent because the greater part of the heat is transmitted through the firepot. The soot clogs the passages and reduces the draft, the loss of efficiency from this action may be much more, and also the lack of draft results in unsatisfactory heating.

Chapter 28

AUTOMATIC FUEL BURNING EQUIPMENT

Residential Stokers, Apartment House Stokers, Commercial Stokers, Domestic Oil Burners, Commercial Oil Burners, Gas-Fired Appliances, Gas Boilers, Warm Air Furnaces, Space Heaters, Conversion Burners, Gas Appliances

AUTOMATIC, mechanical equipment for the efficient combustion of coal, oil, and gas is considered in this chapter.

MECHANICAL STOKERS

Coal can be burned more efficiently on a mechanical stoker than by hand firing. The burning of coal involves uniformity of stoking, proper distribution over the fuel bed, admission of air as required to the fuel bed, and means for removing ash. The proper burning of the fuel on the grate is the function of the stoker and depends upon the stoker design.

The burning of the volatile gases above the fuel bed is a matter of furnace design. The requirements are the same regardless of the type of stoker. Proper care should be taken to provide furnaces sufficiently liberal in volume and with grates at a sufficient distance from the heating surface in order to permit proper combustion of gases. The standards that have been most universally adopted for the proportioning of furnaces are those of the *Midwest Stoker Association.*

Stokers may be divided into four types according to their construction, namely, (1) overfeed flat grate, (2) overfeed inclined grate, (3) underfeed side cleaning type, and (4) underfeed rear cleaning type. They may also be classified according to their uses. The following classification has been recommended by the *Stoker Manufacturers Association.*

Class 1. Up to 60 lb coal feed per hour (Household).

Class 2. 60 to 500 lb coal feed per hour (Apartment house and small commercial).

Class 3. 500 to 1200 lb coal feed per hour and less than 36 sq ft of grate area (General commercial heating and small high pressure steam plants).

Class 4. Over 1200 lb coal feed per hour and over 36 sq ft grate area (Large commercial and high pressure steam plants).

Overfeed Flat Grate Stokers

This type is represented by the various chain- or traveling-grate stokers. These stokers receive fuel at the front of the grate in a layer of uniform thickness and move it back horizontally to the rear of the furnace. Air is

supplied under the moving grate to carry on combustion at a sufficient rate to complete the burning of the coal near the rear of the furnace. The ash is carried over the back end of the stoker into an ashpit beneath. This type of stoker is suitable for small sizes of anthracite or coke breeze and also for bituminous coals, the characteristics of which make it desirable to burn the fuel without disturbing it. This type of stoker requires an arch over the front of the stoker to maintain ignition of the incoming fuel. Frequently, a rear combustion arch is required to maintain ignition until the fuel is fully consumed. A typical traveling-grate stoker is illustrated in Fig. 1.

Another and distinct type of overfeed flat-grate stoker is the spreader or sprinkler type in which coal is distributed either mechanically or by air over the entire grate surface. This type of stoker has a wide application

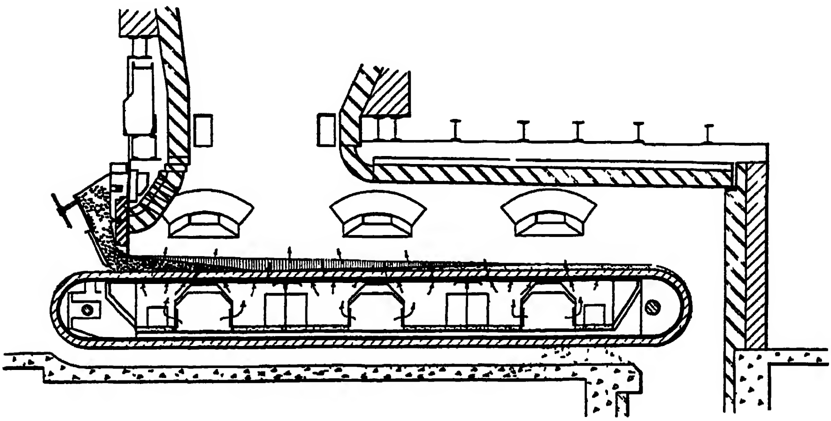


Fig. 1. Overfeed Traveling-Grate Stoker

on small sized fuels and on certain special fuels such as lignites, high-ash coals, and coke breeze.

Overfeed Inclined Grate Stokers

In general the combustion principle is similar to the flat grate stoker, but this stoker (Fig. 2) is provided with rocking grates set on an incline to advance the fuel during combustion. Also this type is provided with an ash plate where ash is accumulated and from which it is dumped periodically. This type of stoker is suitable for all types of coking fuels but preferably for those of low volatile content. Its grate action has the tendency to keep the fuel bed well broken up thereby allowing for free passage of air. Because of its agitating effect on the fuel it is not so desirable for badly clinkering coals. Furthermore, it should usually be provided with a front arch to care for the volatile gas.

Underfeed Side Cleaning Stokers

In this type (Fig. 3), the fuel is fed in at the front of the furnace to one or more retorts, is advanced away from the retort as combustion progresses, while finally the ash is disposed of at the sides. This type of

stoker is suitable for all coking coals while in the smaller sizes it is suitable for small sizes of anthracites. In this type of stoker the fuel is delivered to a retort beneath the fire and is raised into the fire. During this process the volatile gas is released, is mixed with air, and passes through the fire

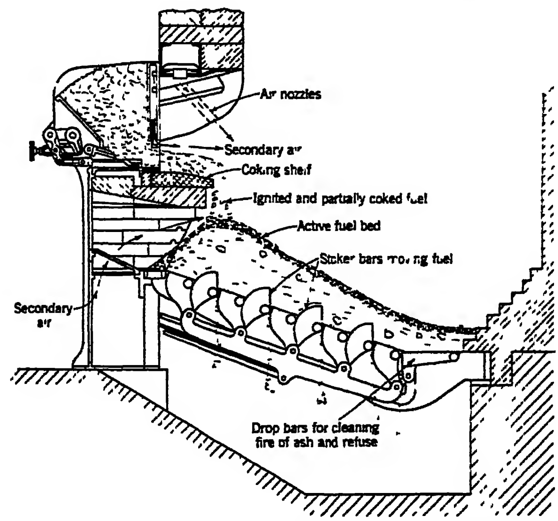


FIG 2. OVERFEED INCLINED GRATE STOKER

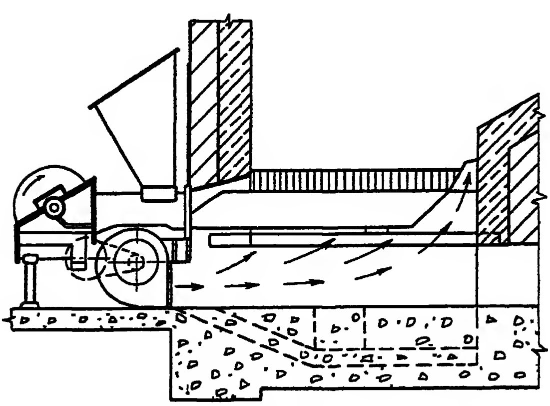


FIG. 3. UNDERFEED PLUNGER TYPE STOKER

where it is burned. The ash may be continuously discharged as in the small stoker or may be accumulated on a dump plate and periodically discharged. This stoker requires no arch as it automatically provides for the combustion of the volatile gas.

Underfeed Rear Cleaning Stokers

This type of stoker carries on combustion in much the same manner as the side cleaning type, but consists of several retorts placed side by side

and filling up the furnace width, while the ash disposal is at the rear. In principle, its operation is the same as the side cleaning underfeed.

Class 1 Stokers, Household

A common type of stoker in this class consists of a round retort having tuyeres at the top where all of the air for combustion is admitted. Coal is fed from a storage hopper (Fig. 4) outside of the boiler by means of a

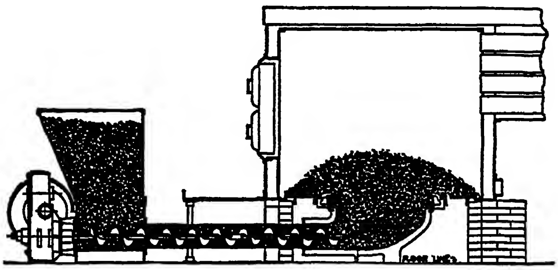


Fig. 4. Underfeed Screw Stoker, Hopper Type

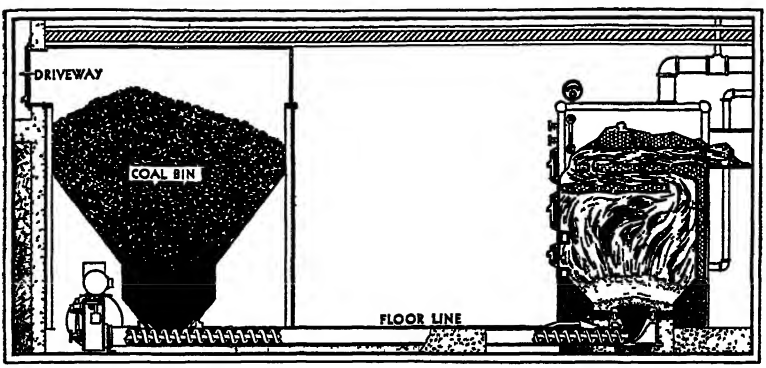


Fig 5. Underfeed Screw Stoker, Bin Type

worm into the bottom of this retort and beneath the fire. The equipment includes a blower which is driven by the same motor that drives the stoker.

Some household stokers are provided with automatic grate-shaking mechanism together with screw conveyers for removing the ash from the ashpit and depositing it in an ash receptacle outside the boiler. Certain types can also be provided with a coal conveyer which takes coal from the storage bin and maintains a full hopper at the stoker. In some cases the stoker hopper functions as the coal bin as shown in Fig. 5, and an extended worm is used to convey the fuel to the boiler. They may feed coal to the furnace either intermittently or with a continuous flow regulated auto-

matically to suit conditions. Where the boiler is provided with indirect coils for heating the domestic hot water, the stoker may be arranged so that it can be used the entire year to maintain a continuous hot water supply.

Household stokers are made for all classes of fuel—anthracite, bituminous and semi-bituminous. The *United States Department of Commerce* has issued commercial standards for household anthracite burners, which may be secured by application. The requirements stated in these standards are described in the following paragraphs.

Operating Requirements

Efficiency. The over-all efficiency of the unit at all points above 50 per cent of maximum coal feed shall be above 50 per cent when installed in a round sectional cast-iron boiler having three intermediate sections and 1½ in of asbestos insulation or its equivalent in good condition of repair, operating at 50 per cent or more of the boiler capacity. The efficiency shall be maintained for any continuous period of 4 hours during any test or observation run.

Ash Loss. Combustible in ash shall not exceed 7.5 per cent of the Btu content of the coal as fired at any rate of coal feed above 50 per cent of maximum. Methods of test according to Code No. 3 of the A.S.H.V.E.[1] are to be followed in all details applicable to stoker testing.

Clinker. Ash removing systems should at all times be capable of disposing of any clinker which may be formed under any conditions of operation with the coals prescribed.

Combustion Rate. A combustion rate of at least 13 lb per square foot of horizontal projected area of ash ring per hour must be continuously maintained for at least 9 hours with the above conditions of efficiency, ash and clinker.

Flue Gas. Flue gas shall be not below 6 per cent in carbon dioxide with a reasonably tight boiler at any rate of operation above 50 per cent of maximum coal feed.

Maximum Rating. The maximum rating, in terms of gross square feet of water or steam radiation which the burner will supply, when intended for installation in the average existing cast-iron boiler, shall be 90 per cent of the maximum steam produced in a round cast-iron boiler in good repair having three intermediate sections and the equivalent of 1½ in. of asbestos insulation. However, in no case shall the maximum rating be greater than 29 sq ft of direct steam radiation for each pound of coal fired per hour, and in no case shall ratings be based upon efficiency figures below 50 per cent.

The maximum rating as defined in the preceding paragraph shall be based upon combustion of Pennsylvania anthracite having the following approximate analysis:

Volatile matter 3.5 to 9 per cent; ash content not to exceed 15 per cent; sulphur content under 1.5 per cent; ash fusing temperature 2750 F, or above (volatile, ash and sulphur content on dry basis in accordance with *A.S.T.M.* method D271-33); Btu content 12,000 or above, properly sized as follows: A No. 1 buckwheat should pass through a round mesh screen having 9/16 in. holes and over a similar screen having 5/16 in. holes. The undersizing should not exceed 15 per cent and the oversizing should not exceed 10 per cent. A No. 2 buckwheat (rice) should pass through a round mesh screen having holes 5/16 in. in diameter and over a like screen having holes of 3/16 in. in diameter. The undersizing should not exceed 15 per cent and the oversizing should not exceed 10 per cent.

Coal Storage. It is recommended that the coal bin or closet be constructed so as to be dustproof.

Electrical Consumption. The electrical consumption shall not exceed 18 kwh per 2000 lb of coal burned at any rate of coal feed above 50 per cent of the maximum.

Operation Upon Other Sizes of Coal The foregoing specifications have been drafted for operating with the Nos. 1 and 2 buckwheat sizes of anthracite. In the event that other sizes are recommended, ratings shall be based upon the same efficiency and ash loss requirements.

[1] A.S.H.V.E. Performance Test Code for Steam Heating Solid Fuel Boilers (Code 3), (A.S.H.V.E. Transactions, Vol. 35, 1929).

Banking. The burner shall be so constructed or controlled as to maintain a fire during an indefinite banking period.

Acceleration. When the burner resumes operation after a 12 hour banking period, the time required for the stack temperature to reach a normal maximum shall not exceed 60 min.

Class 2 Stokers, Apartment House, Small Commercial

This class is used extensively for heating plants in apartments and hotels, and also for small industrial plants such as laundries, bakeries, and creameries. The majority of stokers used in this field are of the underfeed type. The principal exception is an overfeed type having step action grates in a horizontal plane and so arranged that they are alternately moving and stationary, and are designed to advance the fuel during combustion to an ash plate at the rear.

All of the stokers are provided with a coal hopper outside of the boiler. In the underfeed types, the coal feed from this hopper to the furnace may be accomplished by a continuously revolving worm or by an intermittent plunger. The drive for the coal feed may be an electric motor, or a steam or hydraulic cylinder. With an electric motor, the connection between the driver and the coal feed may be through a variable speed gear train which provides two or more speeds for the coal feed; or it may be through a simple gear train and a variable speed driver for the change in speed of the coal feed; or a simple gear train with a coal feed having an adjustment for varying the travel of the feeding device. With a steam or hydraulic cylinder, the power piston is connected directly to the coal feeding plunger.

The stokers in this class vary also in their retort design according to the fuels and load conditions. The retort is placed approximately in the middle of the furnace and is provided with tuyere openings at the top on all sides. In the plunger-feed type the retort extends from the inside of the front wall entirely to the rear wall or to within a short distance of the rear wall. This type of retort has tuyeres on the sides and at the rear.

These stokers also differ in the grate surface surrounding the retort. In many of the worm-feed stokers this grate is entirely a dead plate on which the fuel rests while combustion is completed. In the dead-plate type, all of the air for combustion is furnished by the tuyeres at the retort. Because of this, combustion is well advanced over the retort so that it may easily be completed by the air which percolates through the fuel bed. With the dead-plate type of grate the ash is removed through the fire doors and it is therefore desirable that the fuel used shall be one in which the ash is readily reduced to a clinker at the furnace temperature, in order that it may be removed with the least disturbance of the fuel bed.

In other stokers in this class, the grates outside of the retort are air-admitting and some stokers have shaking grates. These grates permit a large part of the ash to be shaken into the ash pit beneath, while the clinkers are removed through the fire doors. With this type of grate, the main air chamber extends only under the retort while the side grates receive air by natural draft from the ash pit.

In still other stokers of this class, the main air chamber extends beyond the retort and is covered with fuel-bearing, air-supplying grates. With this type of grate, the fuel is supplied with air from the main air chamber

throughout combustion. Also with this type of grate, dump plates are provided beyond the grates where the ash accumulates and from which it can be dropped periodically into the ash pit beneath.

Stokers in this class are compactly built in order that they may fit into standard heating boilers and still leave room for sufficient combustion space above the grates. The height of the grate is approximately the same as that of the ordinary grates of boilers, so that it is usually possible to install such stokers with but minor changes in the existing equipment. In some districts, there are statutory regulations governing such settings.

These stokers vary in furnace dimensions from 30 in. square to approximately 66 in. square. The capacity of the stokers is measured by the amount of coal that can be burned per hour. In general, manufacturers recommend that, for continuous operation, the coal burning rate shall not exceed 25 lb of coal per square foot of grate per hour, while for short peaks this rate may be increased to 30 lb per hour. Although these stokers were designed to burn bituminous coal, types are available for the semi-bituminous coals such as Pocahontas and New River. They can also be used to burn the small sizes of anthracite but at a somewhat

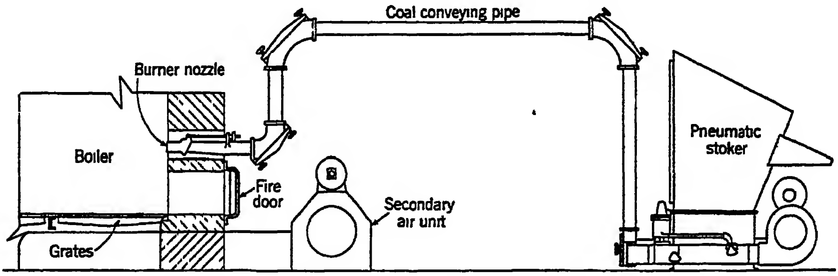


FIG. 6. PNEUMATIC STOKER WITH CONVEYING PIPE

lower rate. It is often customary to have the janitor or some other attendant care for the boiler as one of his duties. Under these conditions the heating plant does not receive the same careful attention as it would if a man devoted his entire attention to the fire. With periodic hand firing, the boiler is operated inefficiently much of the time. With a stoker, the boiler is operated at the rate that the conditions require so long as there is coal in the hopper. With hand firing, it is customary to use a more expensive size of fuel, while with a stoker the smaller sizes are used at a considerable saving in the cost per ton. Because the stoker responds promptly to automatic regulation, it is possible to maintain a reasonably constant standard. Also because the stoker feeds the fuel regularly and in small quantities without losses due to opening doors, it is more efficient than hand firing. This increase in efficiency depends entirely on conditions, with a minimum of about 10 per cent and a maximum of about 25 per cent.

Another type of stoker which may be used in connection with small size coal is a pneumatic type as shown in Fig. 6. The equipment may be arranged with a pipe conveying the fuel from a storage space directly to the burner nozzle with secondary air supplied from a separate unit located near the boiler.

Class 3 Stokers, General Commercial, Small High Pressure Plants

The general commercial heating and high pressure plant with boilers burning from 500 lb of coal per hour to 1200 lb of coal per hour, is a field distinct from the large commercial and industrial field. The prevalent type of stoker employed is the single retort underfeed side-cleaning type. In this field is found the greatest financial return on stoker investments due to the fact that 90 per cent of the fuel is purchased in the retail market where prices are on the average 40 per cent higher for the same fuel than if the coal were purchased in carload quantities. This is, therefore, an extremely important class of stokers. Savings of 40 per cent to 50 per cent are not at all unusual, and many plants show as much as a 50 per cent return on the investment.

Automatic Stoker Controls

The industry developed by stokers in Classes 1, 2 and 3 has been due as much to the application of proper controls as to the stoker itself, as the types of stokers used in these classes are not basically new, while the industry is distinctly new, originating in 1923. The usual controls applied are as follows:

a. Thermostats (plain and clock).
b. Limit Controls (steam, vapor, hot water, etc.).
c. Stack Temperature or Time Controls (for actuating fires periodically).
d. Relay (for low voltage controls).
e. Safety or Overload Cutout (for protection against overload).

Class 4 Stokers, Large Commercial, High Pressure Steam Plants

This class includes stokers with grate areas above 36 sq ft and with hourly burning rates of over 1200 lb of coal per hour. The prevalent stokers in this field are:

a. Overfeed flat grate stokers.
b. Overfeed inclined grate stokers.
c. Underfeed side cleaning stokers.
d. Underfeed rear cleaning stokers.

Overfeed inclined grate stokers are seldom built in sizes of over 500 hp and are not as extensively used as other types of stokers.

Underfeed side-cleaning stokers are made in sizes up to approximately 500 hp and in this field are extensively used. These stokers are not so varied in design as those in the smaller classes although the principle is much the same. Practically all of them are of the front coal feed type, either power driven or steam driven. Dump plates at the side are manually operated. These stokers are heavily built and designed to operate continuously at high boiler ratings with a minimum amount of attention. Because of the fact that all volatile gas must pass through the fire before reaching the combustion chamber, these stokers will operate smokelessly under ordinary conditions. Also because of the fact that these stokers are always provided with forced draft, they are the most desirable type for fluctuating loads or high boiler ratings.

In the design of the grates for supporting the fuel between the retort and the ash plates, the stokers differ in providing for movement of the

fuel during combustion. Some stokers are designed with fixed grates of sufficient angle to provide for this movement as the bed is agitated by the incoming fuel, while others have alternate moving and stationary bars in this area and provide for this movement mechanically. In either type, with proper operation, all refuse will be deposited at the dump plate Recent developments in this type of stoker provide for sliding distributor blocks along the bottom of the retorts which give flexibility in providing proper distribution of fuel over the grate area and assist in preventing coke masses when strong coking coals are used. Another difference in these stokers is that some use a single air chamber under the whole grate area thus having the same air pressure under the ignition area as under the rest of the grate, while others have a divided air chamber using the full air pressure under the ignition area and a reduced air pressure under the remainder of the grate. These stokers vary in size from approximately 5 sq ft to a maximum of 8½ sq ft.

The most prevalent type of rear-cleaning underfeed stoker is the multiple retort design. Occasionally double or triple retort side-cleaning underfeeds are made. The multiple retort underfeed stoker is made for the largest sizes of boilers for large industrial plants and central stations. This stoker has reached a very fine stage of development mechanically and in the matter of air supply and control. In some instances zoned air-control has been applied both longitudinally and transversely to the grate surface. Ash dumps on smaller sizes are sometimes manually operated,

Table 1. Recommended Setting Heights for Heating Boilers Equipped with Mechanical Stokers[a]

Firebox Boilers

Actual Load	2500	5000	7500	10000	12500	15000	20000	25000	30000
A	18″	18″	20″	20″	22″	22″	24″	24″	24″
B	42″	48″	54″	60″	66″	72″	78″	84″	84″

A = Distance from bottom of Water Leg to floor. *B* = Distance from Crown Sheet to bottom of Water Leg.

Compact Welded Boilers

Actual Load	2500	5000	7500	10000	12500	15000	20000	25000	30000
A	18″	18″	20″	20″	22″	22″	24″	24″	24″
B	30″	33″	36″	42″	45″	48″	54″	60″	60″

A = Distance from bottom of Water Leg to floor *B* = Distance from Crown Sheet to bottom of Water Leg.

H R. T Boilers

Hp	50	75	100	125	150	175	200	225	250	275	300
A	5′-0″	5′-6″	6′-0″	6′-6″	7′-0″	7′-0″	7′-6″	8′-0″	8′-6″	9′-0″	9′-0″

A = Distance from bottom of shell to floor *Hp* = Installed horsepower

In the case of the Firebox or Compact Welded type boilers the desired setting height can be obtained by combining *A* and *B* dimensions The load ratings shown for this class of boilers are actual developed loads in square feet of equivalent cast iron steam radiation and are not manufacturers' ratings.

The setting heights given for H. R. T. boilers may be used for developed loads up to 50 per cent above normal rating.

[a]From Data Prepared by the *Midwest Stoker Association.*

while larger sizes are power operated. The number of retorts and dimensions of furnaces are practically unlimited.

The V-type stoker is practically obsolete although many are still in operation. In this stoker, the grates are inclined downward from both sides of the furnace to a low point at the middle where there is either a dump plate for periodic disposal of the ash or a rotary ash grate for continuous discharge of ash. In this stoker, the fuel is fed into a hopper at the top of the grate on each side of the furnace and advanced down the grates to the center where the refuse is accumulated. This stoker is always provided with a combustion arch over the entire furnace for the purpose of assuring thorough combustion of the solid fuel and providing a furnace temperature sufficiently high to burn the volatile gases. Because of this high furnace temperature and because so little of the boiler surface is exposed to the fire to assist in carrying off the heat by radiation, this stoker is characterized by severe clinkering in the ash area. With all types of overfeed stokers, the most desirable installations are in boilers which are operated with comparatively uniform loads and moderate rates of combustion, since, even with good combustion arches, fluctuating loads or high combustion rates result in smoke.

Table 1 gives recommended setting heights for heating boilers equipped with mechanical stokers.

DOMESTIC OIL BURNERS

An oil burner is a mechanical device for producing heat automatically and safely from liquid fuels. This heat is produced in the furnace or firepot of hot water or steam boilers or warm air furnaces and is absorbed by the boiler, and thus made available for distribution to the house through the heating system. Heat production is thus only one of many functions to be performed by a heating system.

Efficient heat production with any kind of fuel requires that all combustible matter in the fuel shall be completely consumed and that it shall be done with no more excess air than necessary. The combustion of oil is a rather rapid chemical reaction. Excess air provides an over supply of oxygen so that all of the oil, composed of carbon and hydrogen, will be completely oxidized and thus produce all the heat possible. The use of unnecessary excess air means no increase or decrease in the heat produced but it does mean that some air is needlessly heated and thrown away. This loss cannot be counteracted by any other part of the heating system and is therefore chargeable to the fuel-burning device.

Oil is a highly concentrated fuel composed exclusively of hydrogen and carbon. In its liquid form oil cannot burn. It must be converted into a gas or vapor by some means. If the excess air is to be kept within efficient limits it means that air must be supplied in carefully regulated quantities. The air and oil vapor must be vigorously mixed to get a rapid and complete chemical reaction. The better the mixing, the less excess air that will be needed. The combustion must take place in a space that maintains the temperatures high so the reaction will not be stopped before completion. When equipped with a means of igniting the oil and safety devices to guard against mishaps, the oil burner possesses all of the elements to be efficient and automatic.

The number of combinations of the characteristic elements of domestic oil burners is rather large and accounts for the variety of burners found in actual practice. Domestic oil burners may be classified as follows

1. AIR SUPPLY FOR COMBUSTION
   - *a. Atmospheric*—by natural chimney draft
   - *b. Mechanical*—electric-motor-driven fan or blower.
   - *c. Combination of (a) and (b)*—primary air supply by fan or blower and secondary air supply by natural chimney draft.
2. METHOD OF OIL PREPARATION
   - *a. Vaporizing*—oil distills on hot surface or in hot cracking chamber.
   - *b. Atomizing*—oil broken up into minute globules.
     - (1) Centrifugal—by means of rotating cup or disc.
     - (2) Pressure—by means of forcing oil under pressure through a small nozzle or orifice.
     - (3) Air or steam—by high velocity air or steam jet in a special type of nozzle.
     - (4) Combination air and pressure—by air entrained with oil under pressure and forced through a nozzle.
   - *c. Combination of (a) and (b).*
3. TYPE OF FLAME
   - *a. Luminous*—a relatively bright flame. An orange-colored flame is usually best if no smoke is present.
   - *b. Non-luminous*—Bunsen-type flame (*i e*, blue flame).
4. METHODS OF IGNITION
   - *a. Electric.*
     - (1) Spark—by transformer producing high-voltage sparks. Usually shielded to avoid radio interference. May take place continuously while the burner is operating or just at the beginning of operation.
     - (2) Resistance—by means of hot wires or plates.
   - *b. Gas.*
     - (1) Continuous—pilot light of constant size.
     - (2) Expanding—size of pilot light expanded temporarily at the beginning of burner operation.
   - *c. Combination*—electric sparks light the gas and the gas flame ignites the oil.
   - *d. Manual*—by manually-operated gas torch for continuously operating burners.
5. MANNER OF OPERATION
   - *a. On and off*—burner operates only a portion of the time (intermittent).
   - *b. High and low*—burner operates continuously but varies from a high to a low flame.
   - *c. Graduated*—burner operates continuously but flame is graduated according to needs by regulating both air and oil supply.

A trade classification of oil burners consists of the following general types: (*a*) gun, (*b*) rotary and (*c*) pot.

The gun type is characterized by an air tube, usually horizontal, with oil supply pipe centrally located in the tube and so arranged that a spray of atomized oil is introduced and mixed in the firepot with the air stream emerging from the air tube. A variety of patented shapes are employed at the end of the air tube to influence the direction and speed of the air

and thus the effectiveness of the mixing process. The most distinguishing feature of the rotary type is that the oil is discharged to the furnace or firepot by a rotating element of special design. The pot type can be identified by the presence of a metal structure, called a pot, in which combustion takes place. While fire brick linings in the boiler are necessary with the gun and rotary type, they are not needed with the pot type.

The oil burners are operated by a small electric motor which pumps the oil and some or all of the air required. The smallest sizes can generally burn not much less than 1½ gal of oil per hour. The grade of oil burned ranges from No. 1 to No. 4 (see pp. 491 to 493). No. 4 oil is the heaviest and most viscous of the various grades mentioned. An oil burner satisfactory for No. 4 oil can burn any of the lighter grades easily but an oil burner recommended for No. 2 oil should never be supplied with the heavier grades. It has been found that while the heavier grades of oil have a smaller heat value per pound, they have, due to greater density, a larger heat value per gallon. The relative economy of the various grades must be based upon price and the amount of excess air required for clean and efficient combustion.

The Combustion Process

Efficient combustion as previously indicated must produce a clean flame and must use relatively small excess of air (*i.e.*, between 25 and 50 per cent). This can be done only by vaporizing the oil quickly, completely, and mixing it vigorously with air in a firepot hot enough to support the combustion. A vaporizing burner (*i.e.*, pot type) prepares the oil vapor before it mixes with air to any extent. If air and oil vapor temperatures are high and the firepot hot, a clear blue flame is produced. There may be a deficiency of air as shown by the presence of carbon monoxide (CO) or an excessive supply of air, depending upon burner adjustment, without altering the clean, blue appearance of the flame. An atomizing burner (*i.e.*, gun and rotary types) is so named because the oil in one way or another is mechanically separated into very fine particles so the surface exposure of the liquid to the radiant heat of the firepot is vastly increased and vaporization proceeds quickly. Since the air enters the firepot with the liquid fuel particles, it follows that mixing, vaporization and burning are all occurring at once in the same space. This produces a luminous instead of a blue or non-luminous flame. In this case a deficient amount of air is indicated by a dull red or dark orange flame with smoky flame tips.

An excessive supply of air may produce a brilliant white flame in some cases or, in others, a short ragged flame with incandescent sparks flashing through the combustion space. While extreme cases may be easily detected, it is generally not possible to distinguish, by the eye alone, the finer adjustments which competent installation requires.

Certain tests indicate that there is no difference in economy between a blue flame and a luminous flame if the position, shape and the per cent of excess air of both flames are about the same.

Furnace or Firepot Design

It is evident that the atomizing burner is dependent upon the surrounding heated refractory or firebrick surfaces to vaporize the oil and

support combustion. While the importance of the firepot is obvious, its design has been troublesome. Unsatisfactory combustion may be due to inadequate atomization and mixing. A firepot can only compensate for these things to a limited extent. If liquid fuel continually reaches some part of the firebrick surface, a carbon deposit will result. Fundamentally, the firepot should enclose a space having a shape similar to the flame but large enough to avoid flame contact. The nearest approach in practice is to have the bottom of the firepot flat but far enough below the nozzle to avoid flame contact; the sides tapering from the air tube at the same angle as the nozzle spray and the back wall rounded. A plan view of the firepot thus resembles in shape the outline of the flame. In this way as much firebrick as possible is close to the flame so it may be kept quite hot. This insures quick vaporization, rapid combustion and better mixing by eliminating dead or inactive spaces in the firepot. An overhanging arch at the back of the firepot is sometimes used to increase the flame travel and give more time for mixing and burning and sometimes to prevent the gases from going too directly into the boiler flues. When good atomization and vigorous mixing are achieved by the burner, firepot design becomes a less critical matter. Where secondary air is used, firepot design is quite important. Manufacturers generally provide careful directions and in some instances provide special firebrick shapes suited to their burners.

Oil and Air Adjustments

Where adjustments of oil and air have been made which give efficient combustion, the problem of maintaining the adjustments constant becomes an important one. Particularly is this true when the change causes the per cent of excess air to decrease below allowable limits of the burner. A decrease in air supply while the oil delivery remains constant or an increase in oil delivery while the air supply remains constant will make the mixture of oil and air too rich for clean combustion. The more efficient the adjustment (*i.e.*, 25 per cent excess air) the more critical it will be of variations. The oil and air supply rates must remain constant.

The following factors may influence the oil delivery rate: (*a*) changes in oil viscosity due to temperature change or variations in grade of oil delivered, (*b*) erosion of atomizing nozzle, (*c*) fluctuations in by-pass relief pressures and (*d*) possible variations in methods 2*b* (3) and 2*b* (4) listed in the previous classification table. Note that any change due to partial stoppage of oil delivery will increase the proportion of excess air. This will result in less heat, reduced economy and possibly a complete interruption of service but usually no soot will form.

The following factors may influence the air supply: (*a*) changes in firepot draft due to a variety of causes (*i.e.*, changes in chimney draft because of weather changes, seasonal changes, back drafts, failure or inadequacy of automatic draft regulator, use of chimney for other purposes, possible stoppage of the chimney and changes in draft resistance of boiler due to partial stoppage of the flues), (*b*) changes in air inlet adjustments to the fan—collection of lint and dirt on the inlet grille may be enough in some cases.

Measurement of the Efficiency of Combustion

Efficient combustion being based upon a clean flame and certain proportions of oil and air employed, it is possible to determine the results by analyzing the gases formed by the combustion process. An Orsat apparatus is a device which measures the volume of carbon dioxide (CO_2), oxygen (O_2) and carbon monoxide (CO) in the flue gases. Except in the case of a non-luminous flame it is usually sufficient to analyze only for carbon dioxide (CO_2). A showing of 10 to 12 per cent indicates the best adjustment if the flame is clean. Most of the good installations at the present time show from 8 to 10 per cent CO_2. Taking into account the potential hazard of oil or air fluctuations with low excess air (high CO_2) a setting to give 10 per cent CO_2 constitutes a reasonable standard for the majority of oil burners.

Additional Design Considerations

Efficient combustion is found to be a question of good design plus a competent installation but it is not the sole feature of a completely satisfactory job. The following items should all be considered: (*a*) clean and efficient combustion, (*b*) a setting of the oil-burning rate to give the proper gross load, (*c*) a boiler to absorb the heat efficiently, (*d*) quiet operation, and (*e*) suitable control and safety devices.

The Gross Load

See Chapter 25 for general material on allowances, etc. To the design or heat loss of the house (Chapter 7), it is customary to add an allowance for piping and pick-up. The most common value for piping is 25 per cent and for pick-up 20 per cent. To these allowances should be added an allowance where domestic hot water is heated by the boiler. (See Chapter 35 for information on Domestic Hot Water). The design load plus the allowances indicates the gross load that should be produced by the boiler. While in every case the gross load will exceed the design load if adequate heating response is to be achieved, there is, however, no object whatever in over-estimating the allowances. The only effect would be to reduce the time of pick-up by a few minutes. Otherwise, it might mean forcing the boiler unduly and increasing the cost of operation.

Setting the Rate of Oil Burning

The rate of oil burning to get the gross output depends upon the combustion (*i.e.*, per cent CO_2) and the efficiency of the boiler in absorbing heat. The oil burning rate in conversion jobs—where a burner is placed in existing equipment—is troublesome to adjust accurately because the boiler efficiency is usually a matter of conjecture rather than actual knowledge. In general, each gallon of oil burned per hour will produce a boiler output of from 300 to 450 ft of steam radiation (see Fig. 7). The number of possible burners, adjustments, and boilers makes each installation a separate problem. Obviously, a satisfactory setting depends upon the knowledge, experience and judgment of the individual installer. It seems wise to set the oil-burning rate on the low rather than high side. The tendency has always been the other way. The oil-burning rate can easily be increased. If properly explained to the owner, he will appreciate the situation if subsequent adjustments are necessary.

Boiler-Burner Units

Boilers especially designed for oil burners are gradually becoming available to the purchaser of this type of equipment. They are used for replacements as well as for new installations. These boilers have more heating surface than the older coal-burning designs. Flue proportions and gas travel have been changed with beneficial results. All questions of firepot design, capacities, efficiencies, etc., have been determined. The selection of the proper size of unit should be a simple process.

Controls

Oil burner controls may be divided into two parts: (*a*) devices to regulate burner operation so the desired house heating result may be obtained and (*b*) devices for the safety and protection of the boiler and

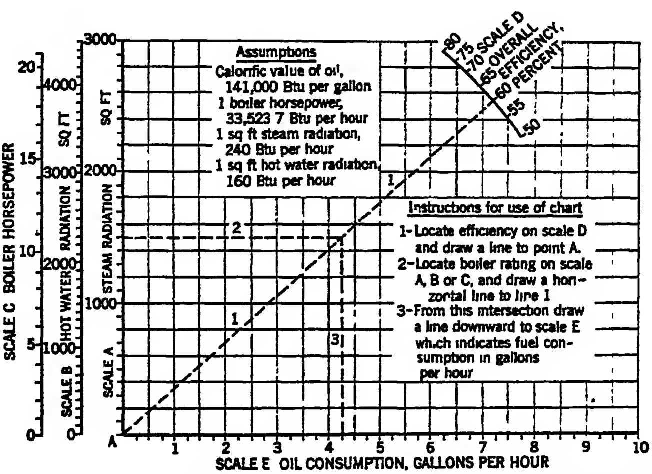


FIG. 7. FULL LOAD RATE OF OIL CONSUMPTION FOR HEATING BOILERS

burner. For control devices generally consult Chapter 14. The room thermostat has recently been improved to provide more frequent burner operation and greater uniformity of room temperature. Class (*b*) controls comprises a device to shut off the burner if the oil fails to ignite or if the flame should cease due to lack of oil; a device actuated by steam boiler pressure to shut off the burner when the pressure reaches some predetermined value; a device on the boiler to shut off the burner if the water level acts too low for safety or one which automatically feeds additional water to the boiler; a device on warm air furnaces to shut off the burner if the air temperature gets too high; a valve in the oil supply line which automatically closes in the event of fire in or near the cellar; and a device to keep the temperature of the boiler water within certain limits when it is being used to heat domestic hot water. These devices are all tested and approved by the Underwriters' Laboratory of Chicago, Ill., before they are offered to the purchaser. The selection of class (*b*) devices is made by the oil burner manufacturer.

Domestic Hot Water Supply

Provisions for heating domestic hot water in connection with automatic fuel-burning devices through heat exchangers attached to the boiler are fully described in Chapter 35.

COMMERCIAL OIL BURNERS

Liquid fuels are used for heating apartment buildings, hotels, public and office buildings, schools, churches, hospitals, department stores, as well as industrial plants of all kinds. Contrary to domestic heating, convenience seldom is a dominating factor, the actual net cost of heat production usually controlling the selection of fuel. Some of the largest office buildings have been using oil for many years. Many department stores have found that floor space in basements and sub-basements can be used to better advantage for merchandising wares, and credit the heat producing department with this saving.

Wherever possible, the boiler plant should be so arranged that either oil or solid fuel can be used at will, permitting the management to take advantage of changes in fuel costs if any occur. Each case should be considered solely in the light of local conditions and prices.

Burners for commercial heating may be either large models of types used in domestic heating, or special types developed to meet the conditions imposed by the boilers involved. Generally speaking, such burners are of the mechanical or pressure atomizing types, the former using rotating cups producing a horizontal torch-like flame. As much as 350 gal of oil per hour can be burned in these units, and frequently they are arranged in multiple on the boiler face, from two to five burners to each boiler.

The larger installations are nearly always started with a hand torch, and are manually controlled, but the use of automatic control is increasing, and completely automatic burners are now available to burn the two heaviest grades of oil. Nearly all of the smaller installations, in schools, churches, apartment houses and the like, are fully automatic.

Because of the viscosity of the heavier oils, it is customary to heat them before transferring by truck tank. It also has been common practice to preheat the oil between the storage tank and the burner, as an aid to movement of the oil as well as to atomization. This heating is accomplished by heat-transfer coils, using water or steam from the heating boiler, and heating the oil to within 30 deg of its flash point.

Unlike the domestic burner, units for large commercial applications frequently consist of atomizing nozzles or cups mounted on the boiler front with the necessary air regulators, the pumps for handling the oil and the blowers for air supply being mounted in sets adjacent to the boilers. In such cases, one pump set can serve several burner units, and common prudence dictates the installation of spare or reserve pump sets. Pre-heaters and other essential auxiliary equipment also should be installed in duplicate.

Boiler Settings

As the volume of space available for combustion is the determining factor in oil consumption, it is general practice to remove grates and

extend the combustion chamber downward to include or even exceed the ash-pit volume; in new installations the boiler should be raised to make added volume available. Approximately 1 cu ft of combustion volume should be provided for every developed boiler horsepower, and in this volume from 1.5 to 2 lb of oil can properly be burned. This corresponds to a maximum liberation of about 38,000 Btu per cubic foot per hour. There are indications that at times much higher fuel rates may be satisfactory. This in turn suggests that the value of 38,000 Btu per cubic foot per hour might be adjusted according to good engineering judgment. For best results, care should be taken to keep the gas velocity below 40 ft per second. Where checkerwork of brick is used to provide secondary air, good practice calls for about 1 sq in. of opening for each pound of oil fired per hour. Such checkerwork is best adapted to flat flames, or to conical flames that can be spread over the floor of the combustion chamber. The proper bricking of a large or even medium sized boiler for oil firing is important and frequently it is advisable to consult an authority on this subject. The essential in combustion chamber design is to provide against flame impingement upon either metallic or fire-brick surfaces. Manufacturers of oil burners usually have available detailed plans for adapting their burners to various types of boilers, and such information should be utilized.

GAS-FIRED APPLIANCES

The increased use of gas for house heating purposes has resulted in the production of such a large number of different types of gas-heating systems and appliances that today there is probably a greater variety of them than there is for any other kind of fuel.

Gas-fired heating systems may be classified as follows:

- I. Gas-Designed Heating Systems.
  - A. Central Heating Plants.
    - 1. Steam, hot water, and vapor boilers.
    - 2. Warm air furnaces.
  - B. Unit Heating Systems.
    - 1. Warm air floor furnaces.
    - 2. Industrial unit heaters.
    - 3. Space heaters.
    - 4. Garage heaters.
- II. Conversion Heating Systems.
  - A. Central Heating Plants.
    - 1. Steam, hot water and vapor boilers.
    - 2. Warm air basement furnaces.

The majority of these systems are supplied with either automatic or manual control. Central heating plants, for example, whether gas designed or conversion systems, may be equipped with room temperature control, push button control, or manual control.

Although no exact rules can be prescribed as to the field best covered by each of the foregoing systems, each installation will have problems pointing more or less directly to some particular type of heating equipment.

Gas-Fired Boilers

Information on gas-fired boilers will be found in Chapter 25.

Either snap action or throttling control is available for gas boiler operation. This is especially advantageous in straight steam systems because steam pressures can be maintained at desired points, while at the same time complete cut-off of gas is possible when the thermostat calls for it.

Warm Air Furnaces

There are two general classes of gas-fired warm air furnaces, the gravity furnace which depends upon the natural tendency of heated air to rise, providing the proper circulation of heated air into the room, and the mechanical circulation furnace by which the air to be heated is forced through or drawn through the furnace by means of a fan.

Warm air furnaces are variously constructed of cast iron, sheet metal and combinations of the two materials. If sheet metal is used, it must be of such a character that it will have the maximum resistance to the corrosive effect of the products of combustion. With some varieties of manufactured gases, this effect is quite pronounced. Warm air furnaces are obtainable in sizes from those sufficient to heat the largest residence down to sizes applicable to a single room. The practice of installing a number of separate furnaces to heat individual rooms is peculiar to mild climates, such as that of Southern California. Small furnaces, frequently controlled by electrical valves actuated by push-buttons in the room above, are often installed to heat rooms where heat may be desired for an hour or so each day. These furnaces are used also for heating groups of rooms in larger residences. In a system of this type each furnace should supply a group of rooms in which the heating requirements for each room in the group are similar as far as the period of heating and temperature to be maintained are concerned. Bedrooms, living rooms, and dining rooms often present excellent possibilities for this type of furnace.

The same fundamental principle of design that is followed in the construction of boilers, that is, breaking the hot gas up into fine streams so that all particles are brought as close as possible to the heating surface, is equally applicable to the design of warm air furnaces. The desirability of using an appliance designed for gas, when gas is to be the fuel, applies even more strongly to furnaces than to boilers.

Codes for proportioning warm air heating plants, such as that formulated by the *National Warm Air Heating and Air Conditioning Association* (see note p. 419), are equally applicable to gas furnaces and coal furnaces. Recirculation should always be practiced with gas-fired warm air furnaces. It not only aids in heating, but is essential to economy. Where fans are used in connection with warm air furnaces for residence heating, it is well to have the control of the fan and of the gas so coordinated that there will be sufficient delay between the turning on of the gas and the starting of the fan to prevent blasts of cold air being blown into the heated rooms. An additional thermostat in the air duct easily may be arranged to accomplish this.

Floor Furnaces

Warm air floor furnaces are well adapted for heating first floors, or

where heat is required in only one or two rooms. A number may be used to provide heat for the entire building where all rooms are on the ground floor, thus giving the heating system flexibility as any number of rooms may be heated without heating the others. With the usual type the register is installed in the floor, the heating element and gas piping being suspended below. Air is taken downward between the two sheets of the double casing and discharged upward over the heating surfaces and into the room. The appliance is controlled from the room to be heated by means of a control lever located near the edge of the register. The handle of the control is removable as a precaution against accidental turning on or off of the gas to the furnace.

Space heaters are generally used for auxiliary heating, but may be, and are in many cases, installed for furnishing heat to entire buildings. Space heaters are quite extensively used for house heating in milder climates such as exist in the South and Southwest. With the exception of wall heaters, they are portable, and can be easily removed and stored during the summer season. Although they should be connected with solid piping it is sometimes desirable to connect them with flexible gas tubing in which case a gas shut-off on the heater is not permitted, and only *A.G.A.* approved tubing should be used.

Space Heaters

Parlor furnaces or *circulators* are usually constructed to resemble a cabinet radio. They heat the room entirely by convection, *i.e.*, the cold air of the room is drawn in near the base and passes up inside the jacket around a drum or heating section, and out of the heater at or near the top. These heaters cause a continuous circulation of the air in the room during the time they are in operation. The burner or burners are located in the base at the bottom of an enclosed combustion chamber. The products of combustion pass up around baffles within the heating element or drum, and out the flue at the back near the top. They are well adapted not only for residence room heating but also for stores and offices.

Radiant heaters make admirable auxiliary heating appliances to be used during the occasional cool days at the beginning and end of the heating season when heat is desired in some particular room for an hour or two. The radiant heater gives off a considerable portion of its heat in the form of radiant energy emitted by an incandescent refractory that is heated by a Bunsen flame. They are made in numerous shapes and designs and in sizes ranging from two to fourteen or more radiants. Some have sheet-iron bodies finished in enamel or brass while others have cast-iron or brass frames with heavy fire clay bodies. An atmospheric burner is supported near the center of the base, usually by set screws at each end. Others have a group of small atmospheric burners supported on a manifold attached to the base. Most radiant heaters are supported on legs and are portable; however, there are also types which are encased in a jacket which fits into the wall with a grilled front, similar to the ordinary wall register. Others are encased in frames which fit into fireplaces.

Gas-fired steam and hot water radiators are popular types of room heating appliances. They provide a form of heating apparatus for intermittently heated spaces such as stores, small churches and some types of offices and apartments. They are made in a large variety of shapes and sizes and are

similar in appearance to the ordinary steam or hot water radiator connected to a basement boiler. A separate combustion chamber is provided in the base of each radiator and is usually fitted with a one-piece burner. They may be secured in either the vented or unvented types, and with steam pressure, thermostatic or room temperature controls.

Warm air radiators are similar in appearance to the steam or hot water radiators. They are usually constructed of pressed steel or sheet metal hollow sections. The hot products of combustion circulate through the sections and are discharged out a flue or into the room, depending upon whether the radiator is of the vented or unvented type.

Garage heaters are usually similar in construction to the cabinet circulator space heaters, except that safety screens are provided over all openings into the combustion chamber to prevent any possibility of explosion from gasoline fumes or other gases which might be ignited by an open flame. They are usually provided with automatic room temperature controls and are well suited for heating either residence or commercial garages.

Conversion Burners

Residence heating with gas through the use of conversion burners installed in coal-designed boilers and furnaces represents a common type of gas-fired house heating system, especially in natural gas territories. In many conversion burners radiants or refractories are employed to convert some of the energy in the gas to radiant heat. Others are of the blast type with luminous flames, operating without refractories. In each case an attempt is made to transfer the majority of the heat from the gas to the medium to be heated within the firepot itself because of the low heat transfer that takes place in the flue passages.

Many conversion units are equipped with sheet metal secondary air ducts which are inserted through the ash-pit door. The duct is equipped with automatic air controls which open when the burners are operating and close when the gas supply is turned off. This prevents a large part of the circulation of cold air through the combustion space of the appliance when not in operation. By means of this duct the air necessary for proper combustion is supplied directly to the burner, thereby making it possible to reduce the amount of excess air passing through the combustion chamber.

Conversion units are made in many sizes both round and rectangular to fit different types and makes of boilers and furnaces. They may be secured with manual, push button, or room temperature control.

Sizing Gas-Fired Heating Plants

While gas-burning equipment can be and usually is so installed as to be completely automatic, maintaining the temperature of rooms at a predetermined and set figure, there are in use installations which are manually controlled. Experience has shown that in order to effectively overcome the starting load and losses in piping, a manually-controlled gas boiler should have an output as much as 100 per cent greater than the equivalent standard cast-iron column radiation which it is expected to serve.

Boilers under thermostatic control, however, are not subject to such severe pick-up or starting loads. Consequently, it is possible to use a

much lower selection, or safety factor. A gas-fired boiler under thermostatic control is so sensitive to variations in room temperatures that in most cases a factor of 25 per cent is sufficient for pick-up load.

The factor to be allowed for loss of heat from piping, however, must vary somewhat, the proportionate amount of piping installed being considerably greater for small installations than for large ones. Consequently, a selection factor for thermostatically controlled boilers must be variable. Table 2 gives liberal selection factors to be added to the installed steam radiation under thermostatic control. They have been established by experience and are recommended by the *American Gas Association*.

The same factors may be used in determining the gas demand for which conversion burners installed in steam or hot water boilers should be set. Multiplying the equivalent direct heating surface (radiation) by 240 and adding the appropriate percentage from Table 2, and then dividing by the heat value of the gas and by the heating efficiency (see discussion of heating efficiencies in Chapter 29), gives the proper hourly rate of gas consumption. However, inadequate boiler heating surface for gas burning, often encountered in coal-designed boilers converted to gas, may necessitate operation at a lesser demand, resulting in much slower pick-up and less margin of safety for piping loss.

Table 2. Selection Factors for Gas Boilers

Cast-Iron Steam Radiation (Equivalent Square Feet)	Selection Factor (Per Cent)
500	56.0
800	54.0
1,200	51.0
1,600	48.0
2,000	45.0
3,000	42 5
4,000 and over	40 0

Appliances used for heating with gas should bear the approval seal of the *American Gas Association* Testing Laboratory. Installations should be made in accordance with the recommendations shown in the publications of that association.

Ratings for Gas Appliances

Since a gas appliance has a heat-generating capacity that can be predicted accurately to within 1 or 2 per cent, and since this capacity is not affected by such things as condition of fuel bed and soot accumulation, makers of these appliances have an opportunity to rate their product in exact terms. Consequently all makers give their product an hourly Btu output rating. This is the amount of heat that is available at the outlet of a boiler in the form of steam or hot water, or at the bonnet of the furnace in the form of warm air. The output rating is in turn based upon the Btu input rating which has been approved by the *American Gas Association* Testing Laboratory and upon an average efficiency which has been assigned by that association.

In the case of boilers, the rating can be put in terms of square feet of equivalent direct radiation by dividing it by 240 for steam, and 150[2] for water. This gives what is called the *American Gas Association* rating, and is the manner in which all appliances approved by the *American Gas Association* Laboratory are rated. To use these ratings it is only necessary to increase the calculated heat loss or the equivalent direct radiation load by an appropriate amount for starting and piping, and to select the boiler or furnace with the proper rating.

The rating given by the *American Gas Association* Laboratory is not only a conservative rating when considered from the standpoint of capacity and efficiency, but is also a safe rating when considered from the standpoint of physical safety to the owner or caretaker. The rating that is placed upon an appliance is limited by the amount of gas that can be burned without the production of harmful amounts of carbon monoxide. This same limitation applies to all classes of gas-consuming heating appliances that are tested and approved by the Laboratory. Gas boilers are available with ratings up to 14,000 sq ft of steam, while furnaces with ratings up to about 500,000 Btu per hour are available. (See Chapter 24.)

Installation Features

One feature of the piping installation that adds to the satisfactory service rendered by gas boilers is provision for adequate and rapid venting of the air from steam heating systems. If air leaks into the steam distribution system during the period that the gas is turned off, and then vents out slowly when the thermostat calls for heat, the result will be a further cooling of the premises between the time that the thermostat calls for heat and the time that steam reaches the radiators. A freely venting steam or vapor system gives maximum economy and minimum temperature variation. When gas boilers are attached to existing heating plants, it is good practice to check the effectiveness of the venting devices and if necessary to replace them with more effective ones that will prevent the return of air into the heating system, and also to check the tightness of the piping.

Frequently when a coal boiler is already installed in a home, it is expedient to leave the coal boiler in place, and to cross-connect the gas boiler with it. Where gas heating is new to the community, it produces a more secure feeling in the customer's mind when putting in gas-fired house-heating equipment, if he knows that he can burn coal at any time he may desire. For steam or vapor installations, it is desirable to have the water line in both boilers at the same level.

REFERENCES

Stoker Information, Bulletin No. 5, Committee of Ten, 307 N. Michigan Avenue, Chicago, Ill

Study of Performance Characteristics of Oil Burners and Low-Pressure Heating Boilers, by L. E Seeley and E. J. Tavanlar (A.S.H.V.E. TRANSACTIONS, Vol. 37, 1931).

Intermittent Operation of Oil Burners, by L. E Seeley and J. H. Powers (A S.H.V.E. TRANSACTIONS, Vol. 38, 1932).

[2]A value of 160 for the heat emission of hot water radiators is used by many engineers. The actual heat emission, however, depends on the temperature of the water and of the surrounding air. See Chapters 30 and 33

Comparison of Oil and Gas Firing in a Heating Boiler, by L. E. Seeley and E. J. Tavanlar (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, October, 1933).

Air Supply and Its Effect on Performance of Oil Burners and Heating Boilers, by L. E. Seeley, J. H. Powers, and E. J. Tavanlar (A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933).

Fuel Burning Rates and Power Requirements of Oil Burners in Relation to Excess Air, by L. E. Seeley and E. J. Tavanlar (A.S.H.V.E. TRANSACTIONS Vol. 40, 1934).

Oil Burning in Residences, by D. W. Nelson (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, July, 1935).

PROBLEMS IN PRACTICE

1 ● What features of furnace design are essential for the proper burning of the volatile coal gases above the fuel bed?

Adequate provisions should be made so that the furnace volume is sufficiently liberal and that the grates are a sufficient distance from the heating surfaces to permit the proper combustion of gases.

2 ● Classify stokers as to construction and operation.

a. Overfeed flat grate.

b. Overfeed inclined grate.

c. Underfeed side cleaning type

d. Underfeed rear cleaning type.

3 ● What classification may be made of stokers as to their use?

Class 1. For residences (Capacity less than 60 lb of coal per hour).

Class 2. For apartment houses and small commercial heating jobs (Capacity 60 to 500 lb of coal per hour).

Class 3. For general commercial heating and small high pressure steam plants (Capacity 500 to 1200 lb of coal per hour).

Class 4. For large commercial and high pressure steam plants (Capacity over 1200 lb of coal per hour and over 36 sq ft grate area).

4 ● What main parts are found in an underfeed residential stoker?

A *hopper* is supplied to hold coal which is fed by a *screw* or *plunger* into a *retort* provided with air openings called *tuyeres*. A *blower* supplies air under pressure for combustion, and a *gear case* provides for changes in coal feeding rates

5 ● What is a dead-plate?

A dead-plate is a flat surface without air supply openings upon which the fuel rests while combustion of the fixed carbon is completed. Generally the ash is removed from the dead-plate.

6 ● What rate of coal burning is usually recommended for small underfeed stokers?

For continuous operation, 25 lb per square foot of grate surface is recommended; for short duration peaks, 30 lb.

7 ● What methods of oil atomization are used?

1. Throwing the oil from a rotating cup or disc.
2. Forcing the oil under high pressure through a nozzle.
3. Propelling the oil with a high velocity jet of air or steam.
4. Forcing an oil and air mixture through a nozzle

8 ● What is the purpose of atomization?

Atomization is used to increase the surface area of the oil in order to facilitate putting it into a vaporous state so it may burn.

9 ● Is the furnace of much importance in oil burning?

In most cases it is very important. It is the function of the oil burner to supply the air and fuel in correct proportions, the furnace must provide heated space for proper mixing and combustion.

10 ● Which flame is considered better, the luminous or the non-luminous?

Laboratory tests show that they are equally efficient in the usual installation.

11 ● What main precaution is necessary in choosing a boiler for an oil burner?

Since the burner output is usually varied through a wide range under control of the thermostat, a boiler should be provided with enough indirect heating surface to absorb the heat as it is released. The combustion space must be large enough, and have correct proportions for mixing fuel and air at high temperatures. If oil is used inefficiently high heating costs will result

12 ● How should oil burner adjustments be made?

Adjustments should be made by an experienced man who uses a gas analysis apparatus to determine the CO_2 content

13 ● What CO_2 content should be attained in oil burning?

Ten per cent CO_2 is considered good practice, for it indicates the supplying of 50 per cent excess air.

14 ● What maximum heat release is considered good practice in oil burning?

A heat release of 38,000 Btu per cubic foot per hour is considered to be the maximum for average large installations. This figure has been greatly exceeded in some cases. The design of the combustion chamber, as to impingement of flame and as to proper mixing at high temperatures, has much to do with the attainable heat release.

15 ● Name five types of gas-fired space heaters.

a. Parlor furnaces or circulators
b. Radiant heaters.
c. Gas-fired steam or hot water radiators.
d. Warm air radiators.
e. Garage heaters

16 ● How are gas heating units rated?

Gas-fired units are rated on the basis of output in Btu per hour.

17 ● What safety consideration is noted in establishing the ratings of gas-fired units?

The rating is limited by the amount of gas that can be burned without the liberation of harmful amounts of carbon monoxide.

18 ● List some factors which might account for possible economies of stoker firing over hand firing.

a. The regular feed of coal instead of the intermittent feed.
b. The use of cheaper sizes and grades of fuel.
c. The absence of door openings for firing purposes, which avoids the admission of cold excess air.
d. The avoidance of overheating because the stoker responds quickly to automatic equipment controlled by the heat demand.

Chapter 29

HEAT AND FUEL UTILIZATION

Total Heat Loss Requirements, Utilization Factors, Degree-Day Methods, Base Temperature Determinations, Steam Consumption of Buildings, Fuel Consumption, Maximum Demands, Load Factors

THE hourly heat loss (H) is equal to the sum of the transmission losses (H_t) and the infiltration losses (H_i) of the rooms or spaces to be heated. The total equivalent heating surface required is equal to $\frac{H}{240}$ sq ft.

In estimating the fuel consumption of a building of more than one room divided by walls or partitions, it is not correct to use the calculated heat loss of the building without making the proper allowances for the fact that the heating load at any time does not involve the sum of the infiltration losses of all the heated spaces of the building but only part of the infiltration losses. This is explained in Chapter 6.

It is sufficiently accurate in most cases to consider only half of the total infiltration losses of a building having interior walls and partitions. The value of H in Equation 1 would, under these conditions, be equal to $H_t + \frac{H_i}{2}$. In some cases, where the building has no interior walls or partitions, the infiltration losses are calculated by using only half of the total crack. In this case the entire infiltration loss should be considered.

The heat required to warm the cold building and contents is a factor to be considered. Under certain conditions the cooling of the structure and contents will, to some extent, compensate for the heat required to rewarm the building. For example, if the building is under thermostatic control and the day and night temperatures are, 70 F and 50 F respectively, there will be a period during which no heat will be added while the building is cooling to 50 F, and the saving resulting therefrom will correspond to the additional heat required to bring the building and contents back to the daytime temperature.

ESTIMATING FUEL CONSUMPTION

There are two methods in use for estimating heat or fuel consumption. One method is theoretical, based on a calculated heat loss and assuming absolute constant temperatures for very definite hours each day throughout the entire heating season. It does not take into account factors which are difficult to evaluate such as opening of windows, abnormal heating of the building, sun effect, poor heating systems, etc.

The second method is based on steam consumption data which have been taken from a group of buildings in operation, and the results computed on a degree-day basis. While this method may not be as theoretically correct as the first mentioned method, it is of more value for practical use. Calculations of heat consumption made by the second method will invariably be higher than calculations made by the first method.

Theoretical Estimation Method

To predict the amount of fuel likely to be consumed in heating a building during a normal heating season, it is necessary to know the total heat requirements of the building and the utilization factor of the fuel. The accuracy of the estimate will depend on the ability to select these values and on the care taken in making allowances for other variable factors.

Heat requirements are given by the following general formula:

$$M = \frac{H\,(t - t_a)\,N}{t_d - t_o} \tag{1}$$

Steam requirements are determined by dividing the above by 1000, thus:

$$S = \frac{H\,(t - t_a)\,N}{(t_d - t_o)\,1000} \tag{2}$$

Fuel requirements may be determined by the following formula:

$$F = \frac{M}{C \times E} \tag{3}$$

where

t = inside temperature, degrees Fahrenheit.
t_d = inside design temperature, degrees Fahrenheit.
t_a = average outside temperature, degrees Fahrenheit (Table 2, Chapter 7).
t_o = outside design temperature, degrees Fahrenheit.
H = calculated heat loss of building based on outside temperature (t_o), Btu per hour.
N = number of heating hours per season, 5088 from October 1 to May 1[1]
M = heat loss, Btu per season.
S = steam required to supply M Btu of heat loss.
F = quantity of fuel required per heating season.
C = calorific value of one unit of fuel, the unit being the same as that on which F is based.
E = efficiency of utilization of the fuel, per cent.

Example 1. A small factory building located in Philadelphia is to be heated to 60 F between the hours of 7 A.M. and 7 P.M., and to 50 F during the remaining hours. The calculated hourly heat loss based on a design temperature of −6 F is 500,000 Btu. If coal having a calorific value of 12,500 Btu is fired and the overall heating efficiency is assumed to be 60 per cent, how many pounds of steam would be required for a normal heating season?

[1]This is the period for which t_a (Table 2, Chapter 7) is calculated If the heating season is different than this period, the corrected values may be substituted for N and t_a

Solution Since there are no partitions in the building, the entire heat loss is considered. From Table 2, Chapter 7, the average outside temperature (t_a) during the heating season is 41 9 F; N for the period for which t_a is taken (October 1 to May 1) is 5088; H = 500,000, t_o = −6 F; t = 50 F and 60 F; t_d = 60 F; E = 60 per cent average for heating season, C = 12,500

The average daily temperature for the 24 hours is·

$$\frac{50 \times 12 + 60 \times 12}{24} = 55 \text{ F}$$

Substituting in Equation 1:

$$M = \frac{500{,}000 \times (55 - 41.9) \times 5088}{60 - (-6)} = 504{,}912{,}000$$

S = 504,942 lb of steam.

$$F = \frac{504{,}942{,}000}{0\ 60 \times 12{,}500} = 67{,}325 \text{ lb of coal} = 33.7 \text{ tons.}$$

Practical or Degree-Day Method

The amount of heat required by a building depends upon the outdoor temperature, if other variables are eliminated. Theoretically it is proportional to the difference between the outdoor and indoor temperatures. Some years ago the *American Gas Association*[2] determined from experiment in the heating of residences that the gas consumption varied directly as the difference between 65 F and the outside temperature. In other words, on a day when the temperature was 20 deg below 65 F, twice as much gas was consumed as on a day when the temperature was 10 deg below 65 F. The degree day is defined in Chapter 44. Degree-days for various cities in the United States and Canada are given in Table 1.

Establishing the Base Inside Temperature. Recently the *National District Heating Association* has studied the metered steam consumption of 163 buildings[3] in 22 different cities and has published data substantiating the fact that the 65 F base originally chosen by the gas industry is approximately correct.

The steam consumption of each building by months was divided by the number of days in each month, thus giving the average daily steam consumption by months. The average steam consumption was then plotted against the average monthly temperature, as shown in Fig. 1, and the temperature at which a line drawn through the points crossed the base line indicated the temperature corresponding to zero steam consumption, or the base temperature. The composite results from 163 buildings calculated in this manner are shown in Table 2.

The resultant average of 66.0 F is close to the *A.G.A.* figure of 65 F. It will be noted that the base temperature calculated for hotels, apartments and residences is consistently higher than those for such buildings as garages, auto sales buildings, and manufacturing buildings. This, of course, would be expected in view of the higher inside temperatures carried in the former group; in fact, an even greater difference would be

[2]See *Industrial Gas Series, House Heating (third edition)* published by the *American Gas Association*
[3]These buildings are all served with steam from a district heating company.

TABLE 1. DEGREE-DAYS FOR CITIES IN THE UNITED STATES AND CANADA[a]

State	City	Jan	Feb	Mar	Apr	May	Sept	Oct	Nov	Dec	Total
Ala	Birmingham	589	577	260	69				318	595	2408
	Mobile	428	311	152					186	394	1471
Ariz	Flagstaff	1153	969	896	654	636[b]	292[d]	577	840	1128	7145
	Tucson	459	325	257	87				252	465	1845
Ark	Hot Springs										2665
	Little Rock	719	582	353	78			47	381	651	2811
Calif	Los Angeles	326	266	239	159	90			123	301	1504
	San Francisco	465	356	354	294	458[b]	502[c]	146	261	428	3264
Col	Colorado Springs	1085	993	884	612	459[b]	162	502	789	1067	6553
	Denver	1079	918	799	534	267	72	428	759	1017	5873
Conn	New Haven	1110	1011	899	543	223	39	360	693	1017	5895
D. C	Washington										4626
Fla	Jacksonville	285	207	56					75	267	890
Ga.	Atlanta	682	558	388	132			96	396	639	2891
	Savannah	409	316	167					201	397	1490
Idaho	Boise	1098	848	651	435	235	108	434	738	1011	4558
	Lewiston										4924
Ill	Chicago	1262	1095	909	549	248	30	353	756	1113	6315
	Springfield	1180	1008	760	365	56		282	681	1038	5370
Ind	Evansville	949	854	640	276			155	528	862	4164
	Indianapolis	1128	969	756	384	58		298	687	1017	5297
Iowa	Des Moines	1392	1173	890	429	118		357	798	1216	6373
	Sioux City	1434	1386	967	489	164	33	415	870	1265	7023
Kans	Dodge City	1116	890	688	342	46		276	672	1004	5034
	Topeka	1221	980	741	339			270	699	1051	5301
Ky.	Lexington	974	867	648	342	25		245	612	903	4616
	Louisville	939	801	589	264			186	552	849	4180
La.	New Orleans	332	230	58					102	301	1023
Me	Eastport	1380	1232	1110	786	843[b]	566[c]	543	843	1228	8531
	Portland	1321	1168	1017	642	368	120	443	780	1153	7012
Md	Baltimore	955	843	700	348	22		223	567	875	4533
Mass.	Boston	1150	1042	908	570	245	48	363	693	1026	6045
	Springfield										6464
Mich.	Detroit	1253	1134	976	573	226	42	400	777	1113	6494
	Marquette	1501	1360	1249	804	682[b]	268[d]	567	960	1301	8692
Minn.	Duluth	1727	1473	1277	810	722[b]	298[d]	620	1062	1491	9480
	Minneapolis	1609	1400	1095	570	235	93	481	963	1405	7851
Miss	Vicksburg	520	384	195					252	471	1822
Mo.	Kansas City	1201	987	750	321	15		285	605	1038	5202
	St. Louis	1060	854	657	276			205	597	936	4585
Mont.	Billings	1316	1120	955	534	376[b]	189	524	909	1192	7115
	Havre	1624	1450	1168	630	513	270	620	1041	1383	8699
Nebr.	Lincoln										6231
	Omaha	1355	1125	868	414	84		328	780	1174	6128
Nev.	Reno	1041	823	753	534	456[b]	144	452	714	974	5891
N H.	Concord	1349	1240	1011	669	351[b]	168	484	846	1234	6852
N J.	Atlantic City	992	903	806	519	220		254	588	893	5175
	Trenton	1014	942	735	402	81		242	588	930	4934
N M.	Santa Fe	1110	902	775	543	301[b]	120	459	780	1073	6063
N. Y.	Albany	1286	1142	980	549	493	72	446	774	1147	6889
	Buffalo	1240	1156	1032	675	347	75	418	774	1104	6821
	New York	1061	960	837	486	155		276	618	955	5348
	Utica	1242	1181	991	587	244	187	426	787	1140	6785
N. C.	Raleigh	722	630	446	183			130	429	694	3234
	Wilmington	555	468	322	108			19	303	527	2302
N. Dak.	Bismarck										8498

[a]Heating and Ventilating Degree-Day Handbook
[b]Including June
[c]Including July and August.
[d]Including August

Table 1 Degree-Days for Cities in the United States and Canada[a] Continued

State	City	Jan	Feb	Mar	Apr	May	Sept	Oct	Nov	Dec	Total
Ohio	Cincinnati	1076	910	747	378	73		290	675	980	5129
	Cleveland	1180	1075	950	564	220	27	366	732	1060	6154
	Columbus	1113	980	703	420	87		313	690	1017	5323
Okla	Oklahoma City	865	742	465	162			105	459	815	3613
Oreg	Portland	806	644	558	402	335[b]	105	332	558	728	4468
	Salem										4629
Pa	Philadelphia	1001	895	756	402	68		242	588	903	4855
	Pittsburgh	1054	944	787	423	78		313	669	967	5235
R I	Providence	1116	1069	890	558	251	63	348	693	1026	6014
S C	Charleston	487	372	242	36				207	425	1769
	Spartanburg	725	688	431	147			121	429	716	3257
S Dak	Sioux Falls										7683
Tenn	Memphis	744	599	384	96			62	402	663	2950
	Nashville	812	747	476	180			136	483	744	3578
Texas	Austin										1578
	Dallas										2455
	Houston	366	277	65					114	335	1157
	San Antonio	381	274	74					126	347	1202
Utah	Logan	1260	1072	893	525	376	114	468	819	1218	6735
	Salt Lake City	1110	885	722	453	234	18	388	723	1020	5553
Vt	Burlington	1535	1294	1089	654	276[b]	144	481	861	1286	7620
Va	Fredericksburg	887	820	583	303			223	549	878	4243
	Norfolk	738	650	520	246			99	411	685	3349
	Richmond	825	702	552	240			158	483	765	3725
Wash	Seattle	775	653	623	465	487[b]	276[c]	403	570	716	4968
	Spokane	1171	952	778	504	366	192	514	819	1057	6353
W. Va	Morgantown	1026	944	713	414			294	648	977	5016
	Parkersburgh										4884
Wis	Fond du Lac	1507	1321	1046	603	276	117	493	921	1328	7612
	Green Bay	1538	1358	1125	600	322	132	505	921	1322	7823
	LaCrosse	1535	1265	1032	528	183	96	462	909	1280	7290
	Milwaukee	1383	1328	1023	648	389[b]	84	449	846	1222	7372
Wyo	Cheyenne	1215	1075	995	720	569	240	605	900	1143	7462

Province	City	Jan.	Feb	Mar	Apr.	May	Sept	Oct.	Nov.	Dec	Total
B. C.	Victoria										5777
	Vancouver										5976
	Kamloops										6724
Alb.	Medicine Hat										8152
Sask.	Qu'Appelle										11,261
Man.	Winnipeg										11,166
Ont	Port Arthur										10,803
	Toronto										7732
Que.	Montreal	1615	1409	1219	720	309	190	372	961	1422	8417
	Quebec										8628
N. B	Fredericton										9099
N. S.	Yarmouth										7694
P. E. I.	Charlottetown										8485

[a] Heating and Ventilating Degree-Day Handbook.

[b] Including June

[c] Including July and August

TABLE 2 BASE TEMPERATURE FOR THE DEGREE-DAY[a]

TYPE OF BUILDING	NO OF BUILDINGS ANALYZED	TEMPERATURE F CORRESPONDS TO ZERO STEAM CONSUMPTION
Office	60	66 2
Office and Bank	4	65 8
Bank	3	66 2
Office and Telephone Exchange	2	65.5
Office and Stores	6	67 4
Stores	11	64.0
Department Stores	12	64 3
Hotels	7	66 5
Apartments	14	68.8
Residences	8	66 9
Clubs	4	65.5
Lodges	5	64 9
Theatres	3	67 6
Churches	2	65 8
Garage	2	64.8
Auto Sales and Service	4	61.2
Newspaper and Printing	3	67.7
Warehouse and Loft	3	67 7
Office and Loft	2	65 2
Manufacturing	8	65.4
Average for 163 Buildings		66 0 F

[a]Report of *Commercial Relations Committee, 1932 Proceedings, National District Heating Association*

TABLE 3. STEAM CONSUMPTION FOR VARIOUS CLASSES OF BUILDINGS[a]

(*Heating Season Only*)

BUILDING CLASSIFICATION	NO OF BUILDINGS LISTED	STEAM CONSUMPTION POUNDS PER DEGREE-DAY—65 F BASIS[d]		
		Per M Cu Ft of Heated Space	Per M Sq Ft of Radiator[c] Surface	Per M Btu per Hr of Heat Loss[b]
Apartments	16	1.78	97 5	0 359
Hotels	10	1.46	80.6	0 371
Residences	12	1.32	64.2	....
Printing	7	1.25	105.5	....
Clubs and Lodges	10	0.96	77.0	....
Retail Stores	18	0.90	80.6	0 268
Theatres	6	0.90	75.0	0 498
Loft and Mfg	16	0.89	72.3	0.283
Banks	7	0.88	45.2	....
Auto Sales and Service	8	0.83	62.2	....
Churches	6	0.58	49.4	....
Department Stores	14	0 57	60.7	0 238
Garages (Storage)[e]	6	0 42	72.3	....
Offices (Total)	35	1.09	70 0	0.283
Offices (Heating only)	35	0 975	65.4	0.256

[a]Includes steam for heating domestic water for heating season only
[b]Heat loss calculated for maximum design condition (in most cases 70 F inside, zero outside).
[c]Equivalent steam radiator surface
[d]The figures are a numerical—not a weighted—average for the several buildings in each class
[e]Based on zero consumption at 55 F

expected. For an average figure, the *A.G.A.* base of 65 F may therefore be safely used, and if greater refinement is desired, the figure for the type of building under consideration can be taken from Table 2.

Table 3[4] gives the steam consumption per degree-day, expressed in three different ways, for 196 buildings in 14 different classifications. These buildings are divided among 21 different cities in the United States. The steam used for heating the domestic water is included in these figures, but in the case of office buildings, the steam for heating only is also shown. The data are placed on a comparable basis by expressing the steam consumption in terms of pounds per degree-day per thousand square feet of equivalent installed radiator surface, per thousand cubic feet of heated space, and per thousand Btu of calculated heat loss.

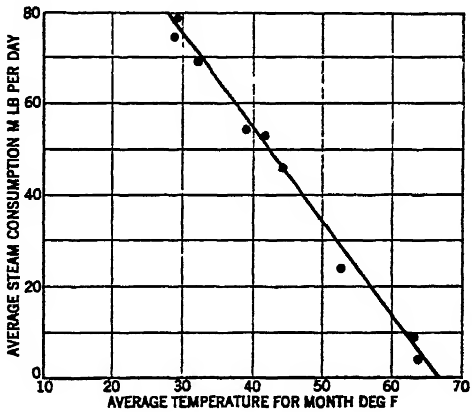


FIG. 1. METHOD OF DETERMINING BASE TEMPERATURE FOR DEGREE-DAY CALCULATIONS[5]

The choice of these units of comparison require some explanation. The use of *heated* space in preference to the gross cubage used by architects is obviously more accurate for this purpose. The architect's cubage includes the outer walls and certain percentages of attic and basement space which are usually unheated. The net heated space is usually about 80 per cent of the gross cubage and can be calculated from the latter if it cannot be measured. The cubical content is somewhat inaccurate as a basis of comparison due to differences in types of construction, exposure, and ratio of exposed area to cubical contents.

The use of radiator surface as the basis of comparison has two objections. One is that the amount of radiator surface in a building is often either excessive or deficient, and figures for steam consumption based on it are therefore likely to be in error. Another reason is that it is difficult to convert fan coil surface into equivalent direct radiator surface with accuracy. On the whole, the use of radiator surface as the basis of comparison is the least satisfactory of the three methods.

[4]The Heat Requirements of Buildings, by J H Walker and G H Tuttle (A S H V E JOURNAL SECTION, *Heating, Piping and Air Conditioning*, December, 1934)

[5]Report of *Commercial Relations Committee, 1932 Proceedings, National District Heating Association*

It should be noted that the figures in Table 3 are for the heating season only and include steam for heating domestic water.

Example 1 solved by the degree-day method and using values taken from Table 3 would show a higher steam consumption.

Example 2 Factor for steam consumption for a manufacturing building per M Btu per hour heat loss per degree-day = 0 283, total number of degree-days per year (Table 1) = 4855, heat loss = 500,000 Btu per hour

Solution 0 283 × 4855 × 500 = 686,982 lb of steam per year. This calculation results in an estimate 36 per cent higher than the previous calculation and one which would be more nearly correct for actual practice.

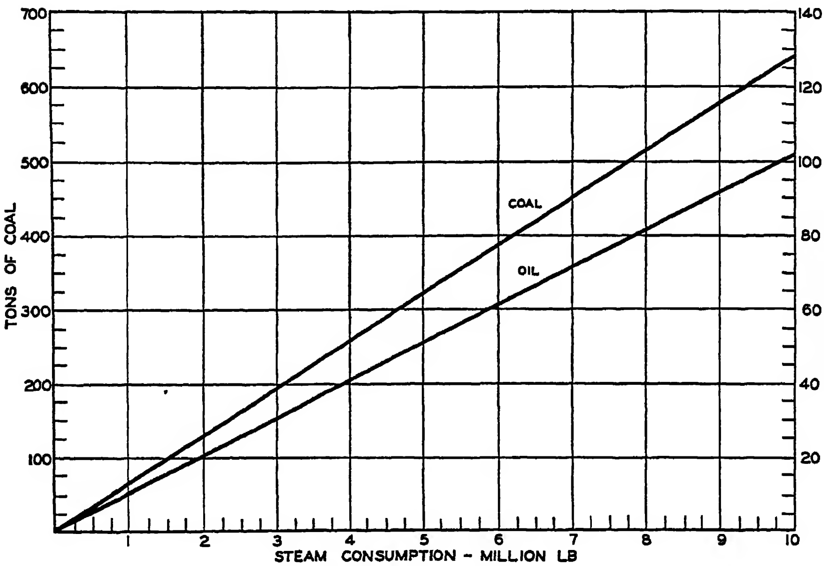


Fig. 2. Curve for Estimating Fuel Consumption for Various Known Steam Consumption[a]

[a]This curve is based on heating efficiencies of 60 to 70 per cent for coal and oil, respectively, a calorific value of coal of 13,000 Btu per pound, a calorific value of oil of 140,000 Btu per gallon

In case the heat loss figure is not known this method of estimating heat or steam consumption can also be applied if the net heated space figure is available.

CALCULATION OF FUEL CONSUMPTION

After the heat and steam consumption of the building have been calculated, the corresponding fuel requirements may also be estimated by assuming the correct boiler and furnace efficiencies. If the building is to be supplied with steam from a district heating company, the steam consumptions as calculated by the two Methods are generally assumed to be correct. However, if the steam is to be supplied from an individual boiler

unit, the consumption should be assumed as from 10 to 20 per cent greater. One reason for this difference in steam consumption is that district steam is a metered service, and building managers are therefore more conscious of their heating costs, which generally results in better maintained heating systems. Also, the district steam service is usually installed with thermostatic control which reduces overheating to a minimum.

Fig. 2 shows the amount of coal or oil that may be estimated when the steam consumption is known. Assuming the steam consumption that

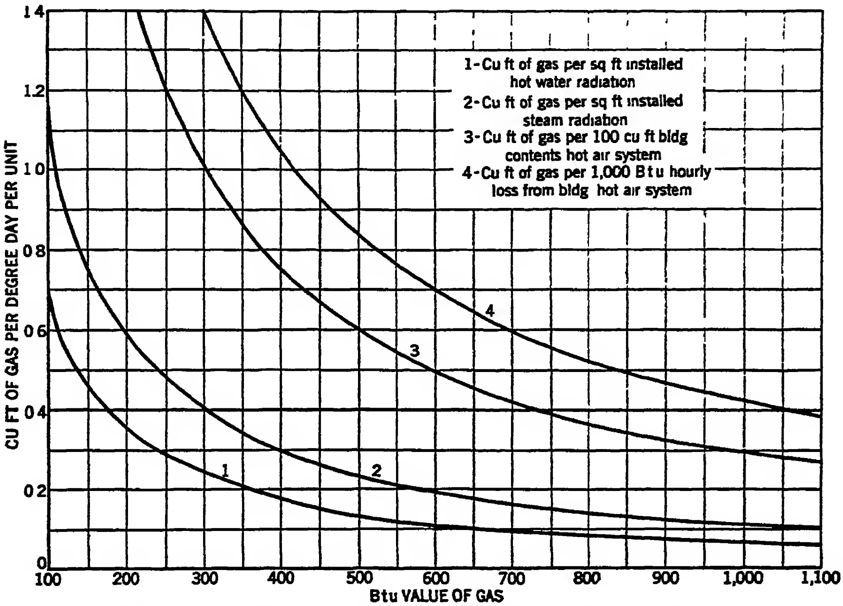


FIG. 3. CHART GIVING GAS REQUIREMENTS PER DEGREE-DAY FOR VARIOUS CALORIFIC VALUES OF GAS AND FOR DIFFERENT HEATING SYSTEMS[a]

[a]This chart is based on an inside temperature of 70 F and an outside temperature of zero. If the *radiation* is installed on the basis of any other temperature difference, multiply the result obtained from this chart by 70, and divide by the actual temperature difference. From *Industrial Gas Series House Heating (third edition)* published by the *American Gas Association*.

was calculated in Example 2, 686,982 lb, the corresponding coal consumption, from the curve, is 44 tons and the oil consumption 6000 gal.

Fig. 3 indicates the average gas consumption per degree-day for various heat contents. While the fuel consumption in individual cases may vary somewhat from the curve values, these average values are sufficiently accurate for estimating purposes and give satisfactory results.

The value generally used in the manufactured gas industry for residences is 0.21 cu ft per degree-day per square foot of equivalent steam *radiation* (240 Btu) based on the theoretical requirements. A correction for warmer climates is necessary and it is customary to gradually increase the relative fuel consumption below 3000 degree-days to about 20 per cent more at 1000 degree-days.

For hot water or warm air heat the fuel consumption is about 0.19 cu ft per degree-day per square foot of equivalent steam *radiation*, that is, per 240 Btu per hour. The actual requirements likewise relatively increase with hot water or warm air systems as the number of degree-days decreases below 3000. For larger installations, that is 1000 sq ft of theoretical *radiation* and above, there is an increase in efficiency, and a consequent decrease in the fuel consumption per degree-day per square foot of heating surface.

The approximate quantities of steam required in New York City per square foot of heating surface for various classes of buildings are given in Chapter 37.

The preceding discussion on fuel consumption has dealt with the heating requirements of the building irrespective of any air that may be introduced for ventilation purposes other than the normal infiltration of outside air. The heat required for warming air brought into the building for ventilation may be estimated from data given in Chapters 3 and 9.

MAXIMUM DEMANDS AND LOAD FACTORS

In one form of district heating rates, a portion of the charge is based upon the maximum demand of the building. The maximum demand may be measured in several different ways. It may be taken as the instantaneous peak or as the rate of use during any specified interval. One method is to take the average of the three highest hours during the winter. These figures are available for a number of buildings in Detroit, as shown in Table 4.

These maximum demands were measured by an attachment on the condensation meter and therefore represent the amounts of condensation passed through the meter in the highest hours, rather than the true rate at which steam is supplied. There might be slight differences in these two quantities due to time lag and to storage of condensate in the system, but wherever this has been investigated it has been found to be negligible.

Table 4. Building Load Factors and Demands of Some Detroit Buildings[a]

Building Classification	Load Factor	Lb of Demand per Hr per Sq Ft of Equivalent Installed Radiator Surface
Clubs and Lodges	0 318	0.184
Hotels	0.316	0.207
Printing	0.287	0.217
Offices	0.263	0.209
Apartments	0.255	0 225
Retail Stores	0.238	0.182
Auto Sales and Service	0.223	0.248
Banks	0.203	0.158
Churches	0.158	0 152
Department Stores	0.138	0.145
Theatres	0.126	0.151

[a]Loc. Cit Note 5

The load factor of a building is the ratio of the average load to the maximum load and is an index of the utilization habits. Thus, in Table 4,

the theatres, operating for short hours, have a load factor of 0.126 as compared with the figure of 0.318 for clubs and lodges.

PROBLEMS IN PRACTICE

1 ● Is it correct to use the total calculated heat loss of a building for estimating fuel consumption?

No. The heating load does not generally involve the sum of the separate infiltration losses of the several heated spaces. When a building has interior walls and partitions it is sufficiently accurate to consider only half of the total calculated infiltration losses

2 ● What will be the cost per year of heating a building with gas, assuming that the calculated hourly heat loss is 92,000 Btu based on 0 F, which includes 26,000 Btu for infiltration? The design temperatures are 0 F and 72 F. The normal heating season is 210 days, and the average outside temperature during the heating season is 36.4 F. The heating efficiency will be 75 per cent. The heating plant will be thermostatically controlled, and a temperature of 55 F will be maintained from 11 p.m. to 7 a.m. Assume that the price of gas is 7 cents per 100,000 Btu of fuel consumption, and disregard the loss of heat through open windows and doors.

The average hourly temperature is

$$\frac{(72 \times 16) + (55 \times 8)}{24} = 66.3 \text{ F.}$$

The maximum hourly heat loss will be

$$92{,}000 - \frac{26{,}000}{2} = 79{,}000 \text{ Btu} = H.$$

$$M = \frac{79{,}000\ (66.3 - 36.4) \times 24 \times 210}{100{,}000 \times 0.75 \times (72 - 0)} = 2204.6 \text{ hundred thousand Btu.}$$

$$2204.6 \times 0.07 = \$154.34 = \text{cost per year of heating the building.}$$

3 ● What factors should be taken into consideration when determining the efficiency at which a fuel will be burned?

Manufacturers' catalogs usually give equipment efficiencies obtained under test conditions. These values do not allow for poor attendance, defects in installation, or poor draft. Such efficiencies do not consider heat radiated from the outside of the equipment, but in many cases this heat is utilized

4 ● If 20 tons of coal having a calorific value of 13,000 Btu per pound are burned in a warm air furnace and produce 286,000,000 Btu at the bonnet, what is the efficiency of the furnace?

$$\frac{\text{Number of Btu at bonnet}}{\text{Number of tons} \times \text{calorific value} \times \text{number of pounds in one ton}} = \text{efficiency.}$$

$$\frac{286{,}000{,}000 \times 100}{20 \times 13{,}000 \times 2000} = 55 \text{ per cent.}$$

5 ● In making degree-day calculations, why is the base of 65 F used for an inside temperature of 70 F?

This base was chosen because data collected from numerous installations show that heat is seldom supplied to a residence when the outdoor temperature is greater than 65 F. It was also found that the amount of fuel consumed varied in almost direct proportion with the difference between 65 F and the outside temperature.

6 ● Use the degree-day method of computing the amount of coal required to heat an office building located in Cleveland, Ohio, assuming that the net heated space is 30,000 cu ft.

The steam consumption factor for office buildings is 0 975 pounds per M cu ft per degree-day (Table 3) Cleveland has 6154 degree-days per year (Table 1)

$$0.975 \times 6154 \times 30 = 180{,}000 \text{ lb of steam}$$

From Fig 2, this is equivalent to 13 tons of coal

7 ● Make a rough approximation of the gas required to heat a building located in Chicago, Ill., assuming that the calculated heating surface requirements are 1000 sq ft of hot water radiation based on design temperatures of 0 F and 70 F. Chicago has 800-Btu mixed gas, and 6315 degree-days.

Using Fig 2, the fuel consumption for a design temperature of 0 F with 800-Btu gas is found to be 0.08 cu ft of gas per degree-day per square foot of hot water radiation

$$0\,08 \times 6315 \times 1000 = 505{,}200 \text{ cu ft.}$$

8 ● A certain building has a maximum heat loss of 250,000 Btu per hour in −15 F weather. How many tons of fuel will be required to maintain a temperature of 70 F during a 260-day heating season in which the average temperature is 39 F? The heating value of the fuel is 13,200 Btu per pound and the efficiency of combustion is 60 per cent.

$$\frac{250{,}000\ (70 - 39)\ 260 \times 24}{(70 + 15)\ 13{,}200 \times 0\,60 \times 2000} = 35\,9 \text{ tons}$$

9 ● Which item may be determined more closely, the heating value of a fuel or the efficiency of its combustion?

The heating values of oil, gas, and solid fuels are closely determinable, whereas the efficiency of burning depends on the particular equipment chosen and the skill used in handling it.

10 ● In an office building, the thermostats are set to maintain 70 F from 7 a.m. to 5 p.m. and 50 F during the rest of the time. When the outside temperature is 30 F, how much saving might be expected because the temperatures are lowered? Under the above conditions the building becomes 50 F by 11 p.m. and warms up to 70 F by 8 a.m.

A temperature of 70 F is maintained during 9 hours, and one of 50 F during 8 hours, the temperature would average about 60 F during the 7 hours required for cooling down and warming up. The average is 60 4 for the 24 hours. (The average temperature calculated would have been 58.3 F, had the warming and cooling periods been neglected.)

The saving is $\left(\frac{70 - 60\,4}{70 - 30}\right) \times 100 = \frac{9\,6}{4} \times 100 = 24$ per cent.

11 ● How does the heat capacity of a structure influence the saving made by carrying lower temperatures during the night?

The heat storage capacity of the walls prevents rapid dropping of temperatures at night-time and delays the warming up process in the morning In an extreme case, the building would not reach the lowered temperature by the time the higher temperature is called for in the morning. But under any conditions, the saving made by lowering the temperature can be correctly estimated by using the average temperature observed over the 24-hour period as a factor, as in Question 10.

12 ● What are some of the miscellaneous factors that may cause actual fuel consumption to vary from the theoretical fuel requirements as calculated by the use of heat losses, temperature difference, and fuel burning efficiency?

The opening of windows, abnormally high or low inside temperatures; other sources of heat, such as machinery or lights; sun effect; and unusual winds.

Chapter 30

RADIATORS AND GRAVITY CONVECTORS

Heat Emission of Radiators and Convectors, Types of Radiators, Output of Radiators, Heating Effect, Heating Up the Radiator, Enclosed Radiators, Convectors, Selection, Code Tests, Gravity-Indirect Heating Systems

THE accepted terms for heating units are: (1) *radiators*, for direct surface heating units, either exposed, enclosed, or shielded, which emit a large percentage of their heat by radiation; and (2) *convectors*, for heating units having a large percentage of extended fin surface and which emit heat principally by convection. Convectors are dependent upon enclosures to provide the circulation by gravity of large volumes of air.

HEAT EMISSION OF RADIATORS AND CONVECTORS

All heating units emit heat by *radiation* and *convection*. The resultant heat from these processes depends upon whether or not the heating unit is exposed or enclosed and upon the contour and surface characteristics of the material in the units.

An exposed radiator emits less than half of its heat by radiation, the amount depending upon the size and number of sections. When the radiator is enclosed or shielded, radiation is further reduced. The balance of the emission is by conduction to the air in contact with the heating surface, and the resulting circulation of the air warms by convection.

A convector emits practically all of its heat by conduction to the air surrounding it and this heated air is in turn transmitted by convection to the rooms or spaces to be warmed, the heat emitted by radiation being negligible.

TYPES OF RADIATORS

Present day radiators may be classified as tubular, wall, or window types, and are generally made of cast iron. Catalogs showing the many designs and patterns available now include a junior size which is more compact than the standard unit.

Pipe Coils

Pipe coils are assemblies of standard pipe or tubing (1 in. to 2 in.) which are used as radiators. In older practice these coils were commonly used in factory buildings, but now wall type radiators are most frequently used for this service. When coils are used, the miter type assembly is to be

preferred as it best cares for expansion in the pipe. Cast manifolds or headers, known as branch tees, are available for this construction.

OUTPUT OF RADIATORS

The output of a radiator can be measured only by the heat it emits. The old standard of comparison used to be square feet of *actual surface*, but since the advance in radiator design and proportions, the surface area alone is not a true index of output. (The engineering unit of output is the *Mb* or 1000 Btu.) However, during the period of transition from the old to the new, radiators may be referred to in terms of *equivalent square feet*. For steam service this is based on an emission of 240 Btu per hour per square foot.

Table 1. Variation in Dimensions and Catalog Ratings of 10-Section Tubular Radiators

No of Tubes	3	4	5	6	7
Width of Radiator ... Inches	4 6–5 1	6 0–7 0	8 0–8 9	9 1–10 4	11 4–12 8
Length per Section ... Inches	2 5	2 5	2 5	2 5	2 5–3 0
Height with Legs—Inches	Heat Emission—Equivalent Square Feet				
13–14	....	....	... .	20	25 0–32 5
16–18	... ...	. ...	28.5	.. ..	30 0–38 3
20–21	15 0–17.5	20 0–22.5	25 0–31.2	30	36 7–45 0
22–23	20 0–21 3	25	30.0–33 9	35	40 0–45 2
25–26	20 0–26 7	25 0–27.5	32 5–39.8	37.5–40 0	50 0–53.5
30–32	25 0–30.9	33 3–35 0	40 0–48 6	50	63 3–62 5
36–38	30 0–36 7	40 0–42 5	50.0–56.5	60	70 0–75.4

Output of Tubular Radiators

Table 1 illustrates the difficulty in tabulating tubular radiator outputs since there is so much variation in design between the products of the different manufacturers. Only on the four-tube and six-tube sizes is there any practical agreement in output value. The heat emission values appear as square feet but are entirely empirical, being based on the heat emission of the radiator and not on the measured surface.

Output of Wall Radiators

An average value of 300 Btu per actual square foot of surface area per hour has been found for wall radiators one section high placed with their bars vertical. Several recent tests[1] show that this value will be reduced from 5 to 10 per cent if the radiator is placed near the ceiling with the bars horizontal and in an air temperature exceeding 70 F. When radiators are placed near the ceiling, there is usually so noticeable a difference in temperature between the floor level and the ceiling that it becomes difficult to heat the living zone of a room satisfactorily.

[1]University of Illinois, *Engineering Experiment Station Bulletin No* 223, p 30

Output of Pipe Coils

The heat emission of pipe coils placed vertically on a wall with the pipes horizontal is given in Table 2. This has been developed from available data and does not represent definite results of tests. For such coils the heat emission varies as the height of the coil. The heat emission of each pipe of ceiling coils, placed horizontally, is about 126 Btu, 156 Btu, and 175 Btu per linear foot of pipe, respectively, for 1-in., 1¼-in., and 1½-in. coils.

TABLE 2. HEAT EMISSION OF PIPE COILS PLACED VERTICALLY ON A WALL (PIPES HORIZONTAL) CONTAINING STEAM AT 215 F AND SURROUNDED WITH AIR AT 70 F

Btu per linear foot of coil per hour (not linear feet of pipe)

SIZE OF PIPE	1 IN.	1¼ IN.	1½ IN.
Single row	132	162	185
Two	252	312	348
Four	440	545	616
Six	567	702	793
Eight	651	796	907
Ten	732	907	1020
Twelve	812	1005	1135

Effect of Paint

The prime coat of paint on a radiator has little effect on the heat output, but the finishing coat of paint does influence the radiation emission. Since this is a surface effect, there is no noticeable change in the convection loss. Thus, the larger the proportion of direct radiating surface, the greater will be the effect of painting on the radiation. Available tests are on old-style column type radiators which gave results shown in Table 3.

TABLE 3. EFFECT OF PAINTING 32-IN. THREE COLUMN, SIX-SECTION CAST-IRON RADIATOR[a]

RADIATOR No.	FINISH	AREA SQ FT	COEFFICIENT OF HEAT TRANS. BTU	RELATIVE HEATING VALUE PER CENT
1	Bare iron, foundry finish	27	1.77	100 5
2	One coat of aluminum bronze	27	1 60	90.8
3	Gray paint dipped	27	1.78	101.1
4	One coat dull black Pecora paint	27	1.76	100.0

[a]Comparative Tests of Radiator Finishes, by W. H. Severns (A S H.V E. TRANSACTIONS, Vol. 33, 1927).

Effect of Superheated Steam

Available research data indicates that there is probably a decrease in heat transfer rate for a radiator or gravity convector with superheated steam in comparison with saturated steam at the same temperature. The decrease is probably small for low temperatures of superheats and additional tests are necessary with varying degrees of superheat to establish accurate comparisons for all types of radiators and convectors[2].

[2]Tests of Radiators with Superheated Steam, by R. C. Carpenter (A.S.H.V E. TRANSACTIONS, Vol. 7, 1901, p. 206).

HEATING EFFECT

For several years the *heating effect* of radiators has been considered by engineers in order to use it for the rating of radiators and in the design of heating systems. Heating effect is the *useful output* of a radiator, in the comfort zone of a room, as related to the total input of the radiator[3].

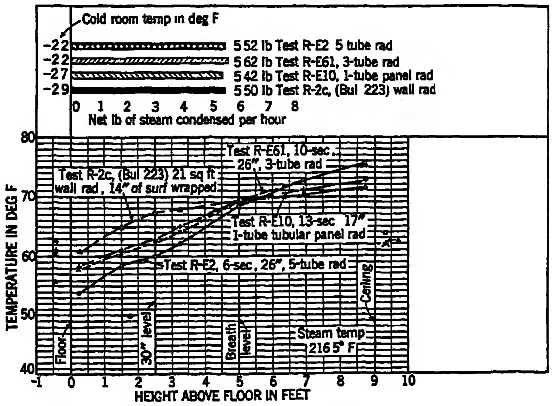


FIG. 1. ROOM TEMPERATURE GRADIENTS AND STEAM CONDENSING RATES FOR FOUR TYPES OF CAST-IRON RADIATORS WITH A COMMON TEMPERATURE AT THE 60-IN LEVEL

Note that the steam condensations are practically the same for all four radiators *when the same air temperature of 69 F is maintained at the 60-in level.*

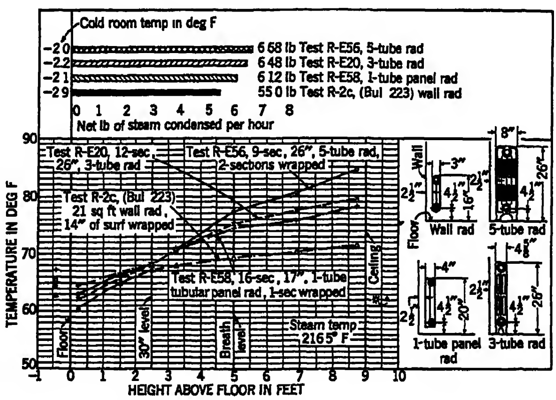


FIG. 2. ROOM TEMPERATURE GRADIENTS AND STEAM CONDENSING RATES FOR FOUR TYPES OF CAST-IRON RADIATORS WITH A COMMON TEMPERATURE AT THE 30-IN. LEVEL

Note that the steam condensations are different for all four radiators *when the same air temperature of 68 F is maintained at the 30-in level*

The results of tests conducted at the University of Illinois are shown in Figs. 1 and 2[4]. For the four types of radiators shown, the following conclusions are given:

---

[3]The Heating Effect of Radiators, by Dr. Charles Brabbée (A S H V E TRANSACTIONS, Vol 33, 1927 p 33) The Application of the Eupatheoscope for Measuring the Performance of Direct Radiators and Convectors in Terms of Equivalent Temperature, by A C Willard, A P Kratz and M K. Fahnestock (A S H V E TRANSACTIONS, Vol 39, 1933)

[4]Steam Condensation an Inverse Index of Heating Effect, by A P Kratz and M. K. Fahnestock (A S H V E TRANSACTIONS, Vol 37, 1931)

1. The heating effect of a radiator cannot be judged solely by the amount of steam condensed within the radiator

2 Smaller floor-to-ceiling temperature differentials can be maintained with long, low, thin, direct radiators, than is possible with high, direct radiators.

3 The larger portion of the floor-to-ceiling temperature differential in a room of average ceiling height heated with direct radiators occurs between the floor and the breathing level.

4. The comfort level (approximately 2 ft-6 in above floor is below the breathing line level (approximately 5 ft-0 in. above floor), and temperatures taken at the breathing line may not be indicative of the actual heating effect of a radiator in the room The comfort-indicating temperature should be taken below the breathing line level.

5. High column radiators placed at the sides of window openings do not produce as comfortable heating effects as long, low, direct radiators placed beneath window openings[5].

HEATING UP THE RADIATOR

The maximum condensation occurs in a heating unit when the steam is first turned on. Fig. 3 shows a typical curve for the condensation rate in pounds per hour for the time elapsing after steam is turned into a cast-iron radiator. The data are from tests on old style column type radiators.

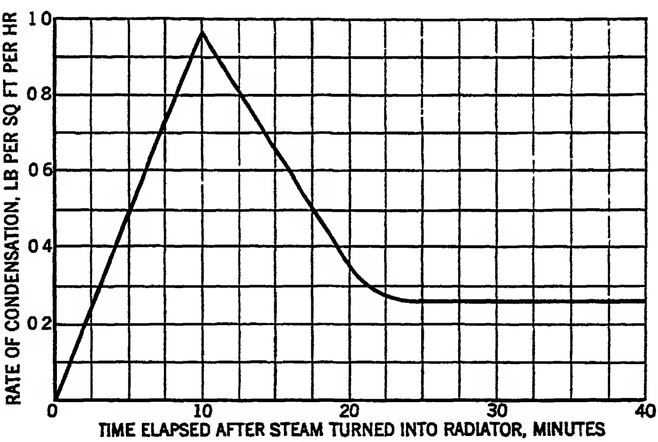


FIG. 3. CHART SHOWING THE STEAM DEMAND RATE FOR HEATING UP A CAST-IRON RADIATOR WITH FREE AIR VENTING AND AMPLE STEAM SUPPLY

In practice the rate of steam supply to the heating unit while heating up is frequently retarded by controlled elimination of air through air valves or traps. Automatic control valves may also retard the supply of steam.

ENCLOSED RADIATORS

The general effect of an enclosure placed about a direct radiator is to restrict the air flow, diminish the radiation and, when properly designed, improve the heating effect. Recent investigations[6] indicate that in the design of the enclosure three things should be considered:

---

[5]Effect of Two Types of Cast Iron Steam Radiators in Room Heating, by A. C. Willard and M. K Fahnestock (*Heating, Piping and Air Conditioning*, March, 1930)

[6]University of Illinois *Engineering Experiment Station Bulletins No* 192 and 223, and Investigation of Heating Rooms with Direct Steam Radiators Equipped with Enclosures and Shields, by A. C. Willard, A. P Kratz, M K. Fahnestock and S Konzo (A S H.V E TRANSACTIONS, Vol 35, 1929).

1. There should be better distribution of the heat below the breathing line level to produce greater heating comfort and lowered ceiling temperatures
2. The lessened steam consumption may not materially change the radiator heating performance.
3. The enclosed radiator may inadequately heat the space

A comparison between a bare or exposed radiator (*A*) and the same radiator with a well-designed enclosure (*B*), with a poorly-designed enclosure (*C*), and with a cloth cover (*D*) will illustrate the relative heating effects. In Fig. 4 the curve (*B*) reveals that the enclosed radiator used less steam than the exposed radiator, but gave a satisfactory heating performance. A well-designed shield placed over a radiator gives about the same heating effect. Curve (*C*) shows the unsatisfactory effects produced by improperly designed enclosures. Curve (*D*) shows that the

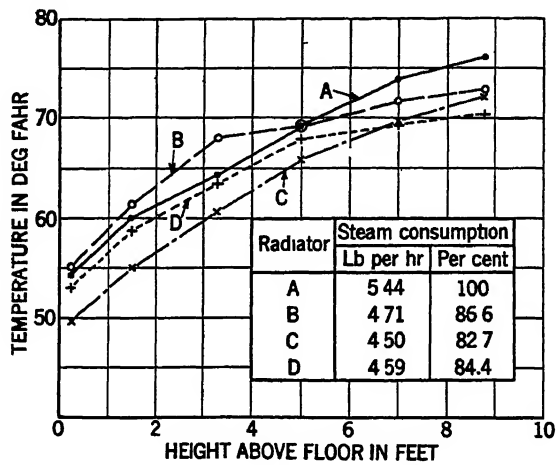


Fig 4. Steam Consumption of Exposed and Concealed Radiators

effect of a cloth cover extending downward 6 in. from the top of the radiator was to make the performance unsatisfactory and inadequate.

Practically all commercial enclosures and shields for use on direct radiators are equipped with water pans for the purpose of adding moisture to the air in the room. Tests[7] show that an average evaporative rate of about 0.235 lb per square foot of water surface per hour may be obtained from such pans, when the radiator is steam hot and the relative humidity in the room is between 25 and 40 per cent. This source of supply of moisture alone is not adequate to maintain a relative humidity above 25 per cent on a zero day.

CONVECTORS OR CONCEALED HEATERS

Although any standard radiator may be concealed in a cabinet or other enclosure so that the greater percentage of heat is conveyed to the

[7]University of Illinois *Engineering Experiment Station Bulletin No* 230, p 20

room by convection thereby resulting in a form of gravity convector. generally better results are obtained with specially designed units which permit a free circulation of a larger volume of air at moderate temperatures. Since air stratifies according to temperature, moderate delivery temperatures at the outlet of the enclosure reduce the temperature differential between the floor and ceiling and accordingly accomplish the desired heating effect in the living zone.

Fig. 5 shows a typical built-in convector. The heating element consisting of a large percentage of fin surface is usually shallow in depth and

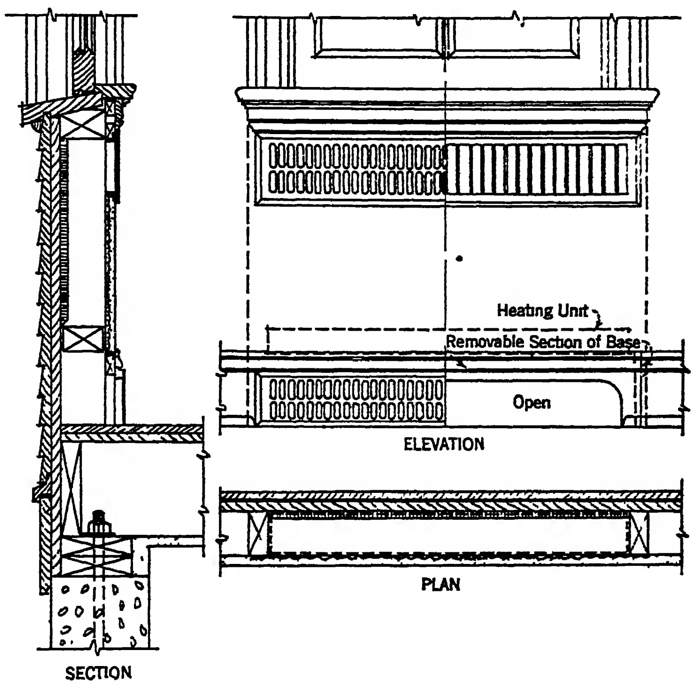


Fig. 5. Typical Concealed Convector Using Specially Designed Heating Unit

placed low in the enclosure in order to produce maximum chimney effect in the enclosure. The air enters the enclosure near the floor line just below the heating element, is moderately heated in passing through the core and delivered to the room through an opening near the top of enclosure. Since the air can only enter the enclosure at the floor line, the cooler air in the room which always lies at this level, is constantly being withdrawn and replaced by the warmer air. This air movement accomplishes the desired reduction in temperature differentials and assures maximum comfort in the living zone.

The *Convector Manufacturers Association* has adopted the A.S.H.V.E. Standard[8] in the formulation of its ratings and has compiled a tentative

[8]A.S.H.V.E Standard Code for Testing and Rating Concealed Gravity Type Radiation (Steam), (A S H.V.E. Transactions, Vol 37, 1931); (Hot Water), (A.S H V E Transactions, Vol. 39, 1933).

standard of heating effect allowances for various enclosure heights to be included in the ratings by its members.

All published ratings bearing the title *C.M.C.* Ratings (*Convector Manufacturers Certified Ratings*) indicate that the convectors have been tested in accordance with the A.S.H.V.E. code by an impartial and disinterested laboratory and that the ratings have been approved by the Standardization Committee of the *Convector Manufacturers Association.*

Concealed heaters or convectors are generally sold as completely built-in units. The enclosing cabinet should be designed with suitable air inlet and outlet grilles to give the heating element its best performance. Tables of capacities are catalogued for various lengths, depths and heights, and combinations are available in several styles for installations, such as the wall-hung type, free-standing floor type, recess type set flush with wall or offset, and the completely concealed type. Most of these types may be arranged with a top outlet grille in a plane parallel with the floor, although the front outlet is practically standard. In cases where enclosures are to be used but are not furnished by the heater manufacturer, it is important that the proportions of the cabinet and the grilles be so designed that they will not impair the performance of the assembled convector. It is important that the enclosure or housing for the convector fit as snugly as possible so that the air to be heated must pass through the convector and cannot be by-passed in the enclosure.

The output of a convector, for any given length and depth, is a variable of the height. Published ratings are generally given in terms of equivalent square feet, corrected for heating effect. However, an extended surface heating unit is entirely different structurally and physically from a direct radiator and, since it has no area measurement corresponding to the heating surface of a radiator, many engineers believe that the performance of convectors should be stated in Btu's. For steam convectors, as for radiators, 240 Btu per hour may be taken as an equivalent square foot of radiation.

RADIATOR AND CONVECTOR SELECTION

Since the capacity of a radiator varies as the 1.3 power and a convector[9] as the 1.5 power of the temperature difference between the inside of radiator and surrounding air it is obvious that for other than 70 F room temperatures the heat emission will be other than 240 Btu per square foot of rating. Therefore in selecting the size of radiator or convector to be used it is necessary to correct for this difference. Table 4 shows factors by which radiation requirements, as determined by dividing heat load by 240, shall be multiplied to obtain proper radiator or convector sizes from published rating tables for room temperatures ranging between 50 and 80 F as well as for steam or water temperatures from 150 to 300 F. For other room and heating medium temperatures the factor is determined by the following formulae:

For radiators:

$$C_s = \left(\frac{215 - 70}{t_s - t_r}\right)^{1.3}$$

[9]Factors Affecting the Heat Output of Convectors, by A P Kratz, M K Fahnestock, and E. L. Broderick (A.S.H.V.E. TRANSACTIONS, Vol 40, 1934)

For convectors·

$$C_s = \left(\frac{215 - 65}{t_s - t_i}\right)^{1.5}$$

where

C_s = correction factor.
t_s = steam temperature, degrees Fahrenheit.
t_r = room temperature, degrees Fahrenheit
t_i = average inlet air temperature, degrees Fahrenheit

TABLE 4. CORRECTION FACTORS FOR DIRECT CAST IRON RADIATORS AND CONVECTOR HEATERS[a]

	Steam Press Approx		Steam or Water Temp F	Factors for Direct Cast Iron Radiators							Factors for Convectors						
				Room Temperature F							Inlet Air Temperature F						
	Gage	Lb Abs		80	75	70	65	60	55	50	80	75	70	65	60	55	50
Vacuum In Hg	22.4	3.7	150	2.58	2.36	2.17	2.00	1.86	1.73	1.62	3.14	2.83	2.57	2.35	2.15	1.98	1.84
	20.3	4.7	160	2.17	2.00	1.86	1.73	1.62	1.52	1.44	2.57	2.35	2.15	1.98	1.84	1.71	1.59
	17.7	6.0	170	1.86	1.73	1.62	1.52	1.44	1.35	1.28	2.15	1.98	1.84	1.71	1.59	1.49	1.40
	14.6	7.5	180	1.62	1.52	1.44	1.35	1.28	1.21	1.15	1.84	1.71	1.59	1.49	1.40	1.32	1.24
	10.9	9.3	190	1.44	1.35	1.28	1.21	1.15	1.10	1.05	1.59	1.49	1.40	1.32	1.24	1.17	1.11
	6.5	11.5	200	1.28	1.21	1.15	1.10	1.05	1.00	0.96	1.40	1.32	1.24	1.17	1.11	1.05	1.00
Lb per Sq In	1	15.6	215	1.10	1.05	1.00	0.95	0.92	0.88	0.85	1.17	1.11	1.05	1.00	0.95	0.91	0.87
	6	21	230	0.96	0.92	0.88	0.85	0.81	0.78	0.75	1.00	0.95	0.91	0.87	0.83	0.79	0.76
	15	30	250	0.81	0.78	0.76	0.73	0.70	0.68	0.66	0.83	0.79	0.76	0.73	0.70	0.68	0.65
	27	42	270	0.70	0.68	0.66	0.64	0.62	0.60	0.58	0.70	0.68	0.65	0.63	0.60	0.58	0.56
	52	67	300	0.58	0.57	0.55	0.53	0.52	0.51	0.49	0.56	0.54	0.53	0.51	0.49	0.48	0.47

[a]To determine the heater size for a given space divide the heat loss in Btu per hour by 240 and multiply the result by the proper factor from the above table

To determine the heating capacity of a heater at other than standard conditions, divide the heating capacity at standard conditions by the proper factor from the above table

CODE TEST FOR RADIATORS AND CONVECTORS

As previously indicated, the output of radiators and convectors is still designated by the terms of older practice, but this is gradually giving place to an engineering method of designating heat emission. The A.S.H.V.E. has adopted the following standards: Code for Testing Radiators (1927); Codes for Testing and Rating Concealed Gravity Type Radiation (Steam, 1932, and Hot Water, 1933).

For steam services the actual condensation weight is taken without any allowance for heating effect; for hot water services the weight of circulated water is used without allowance for heating effect. In all cases the total heat transmission varies as the 1.3 power for radiators[10] and the 1.5 power for convectors[11] of the temperature difference between that inside the radiator and the air in the room, and is expressed in Btu or *Mb* per hour.

Standard test conditions specify either a steam pressure of 1 lb gage (215 F), or hot water at 170 F and a room temperature of 70 F for radiators, or an inlet air temperature of 65 F for convectors. The heating capacity of a *steam radiator* or *steam convector* is determined as follows:

$$H_t = W_s h_{fg} \quad (1)$$

[10]Loc Cit Note 8.
[11]Loc. Cit Notes 8 and 9

where

H_t = Btu per hour under test conditions
W_s = condensation in pounds per hour
h_{fg} = latent heat in Btu per pound

H_t may be converted to standard conditions of code ratings by using the proper correction factor from the following formulae.

For radiators:

$$C_s = \left(\frac{215 - 70}{T_s - T_r}\right)^{1.3} = \left(\frac{145}{T_s - T_r}\right)^{1.3} \tag{2}$$

For convectors:

$$C_s = \left(\frac{215 - 65}{T_s - T_i}\right)^{1.5} = \left(\frac{150}{T_s - T_i}\right)^{1.5} \tag{3}$$

The output under standard conditions will be:

$$H_s = C_s H_t \tag{4}$$

where

C_s = correction factor.
T_s = steam temperature during test, degrees Fahrenheit.
T_r = room temperature during test, degrees Fahrenheit
T_i = inlet air temperature during test, degrees Fahrenheit
H_s = heat emission rating under standard conditions, Btu per hour.

Similarly, for *hot water convectors*, the output under test conditions may be determined as follows:

$$H = W(\theta_1 - \theta_2)\frac{3600}{t} \tag{5}$$

where

H = Btu per hour under test conditions.
W = pounds of water handled during test
θ_1 = average temperature of inlet water, degrees Fahrenheit.
θ_2 = average temperature of outlet water degrees Fahrenheit
t = duration of test, seconds

To convert test results to standard conditions, the following correction factor is used:

$$C = \left(\frac{170 - 65}{\frac{\theta_1 + \theta_2}{2} - T_i}\right)^{1.5} = \left(\frac{105}{\frac{\theta_1 + \theta_2}{2} - T_i}\right)^{1.5} \tag{6}$$

It has been shown that when the exponent 1.5 is used the range of error is less than 3 per cent[12] for convectors.

GRAVITY-INDIRECT HEATING SYSTEMS [13]

The heating units for this system are usually of the extended surface type for steam or hot water, and are installed about as shown in Fig. 6. The temperature and volume of the air leaving the register must be great

[12]Loc. Cit. Note 9.

[13]For further information on this subject see A S H.V.E. Code of Minimum Requirements for the Heating and Ventilation of Buildings (edition of 1929) and *Mechanical Equipment of Buildings*, by Harding and Willard, Vol I, second edition, 1929

enough so that in cooling to room temperature the heat available will just equal the heat loss during the same time. In cases where ventilation is a requirement, the air volume needed may become so large that the entering air temperature will be but slightly above the room temperature. To establish and maintain a constant heat flow, provision must be made for removing the air in the room, after it has cooled to the desired room temperature, by a system of vent flues or ducts. As the air flow is maintained

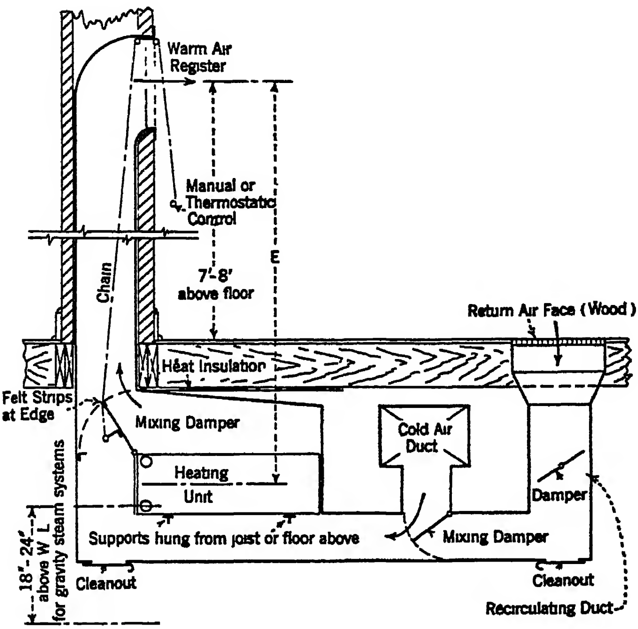


FIG 6. GRAVITY-INDIRECT HEATING SYSTEM[a]

[a]See *Mechanical Equipment of Buildings*, by Harding and Willard, Vol. I, second edition, 1929.

by natural draft and this gravity head is very slight, it is necessary to make all ducts as short as possible, especially the runs from the heating units to the base of the vertical warm air flues. Gravity-indirect arrangements, such as illustrated in Fig. 6, are not to be generally recommended for hot water systems unless the water temperature can be maintained at a reasonably high temperature and rapid circulation of the water can be had.

PROBLEMS IN PRACTICE

1 ● What is the effect on the heat output of a wall radiator when installed on the ceiling of a room?

Because the temperature differential is increased between the floor level and the ceiling when a wall radiator is placed near the ceiling, the heat output may be decreased from 5 to 10 per cent. Under such circumstances it becomes difficult to heat the living zone of a room satisfactorily

2 ● What are the principal differences between a radiator and a convector?

A radiator is commonly thought of as a commercial heating unit having a maximum amount of direct heating surface, whereas a convector is a heating device in which the extended or secondary surface may be several times that of the prime surface and which is specially designed to utilize to the fullest extent the convection principal of heating The radiator ordinarily has vertical tubular chambers for the heating medium but most convectors have horizontal tubular chambers to which fins are attached so as to form vertical flues for the passage of air While radiators are either exposed, enclosed, or shielded, convectors are concealed by means of a tight-fitting enclosure Radiators are commonly made of cast-iron but convectors may be made of a combination of metals, such as copper and brass, or copper and aluminum, as well as entirely of cast iron.

3 ● How did the term heating effect come into use?

It has been found that a room requiring a radiator of a certain determined capacity could under certain conditions be properly heated, with less temperature gradient between floor and ceiling and with less steam condensation, by the same radiator or by one of a different design having the same commercially rated capacity. This resulted in the use of the term *heating effect* to apply to the useful heat output of a radiator, in the comfort zone of a room, as related to the total input to the radiator.

4 ● Is it necessary to make any allowance for the performance of a convector because it is enclosed?

No. The commercial ratings of convectors have been determined by testing the convectors in proper enclosures with grilles in place just as they should be installed for ordinary service.

5 ● On what basis are the capacities of convectors published?

Published ratings of convectors are expressed in equivalent square feet of direct cast iron radiation Some manufacturers have increased their ratings by as much as 30 per cent to allow for a supposed improved heating effect. Tests indicate that the credit to be given heating effect is, in all cases, probably less than 10 per cent, and in many cases negligible.

6 ● How are fins of convectors attached to the tubes or prime surface?

Tubes or a solid core may be forced through piercings in the fins under pressure, or the tubes may be expanded into the holes through the fins. In addition a metallic bonding agent is sometimes used to insure permanent contact.

7 ● What is the procedure in selecting a convector when the required amount of radiation is known?

First the limiting factor or factors of the enclosure must be determined so the available size of the wall recess can be found Manufacturers' catalogs show capacities of convectors of each standard length and depth with varying enclosure heights From these capacity tables, the proper convector of the required capacity can be selected for the available wall recess. If all three dimensions of the wall recess are insufficient to accommodate a convector of the required capacity, the available height and length can be maintained, but greater depth can be obtained by using a partially recessed enclosure

8 ● Given a room to be heated to 80 F with outside temperature at 0 F, assume the heat loss under these conditions to be 10,000 Btu per hour. Determine the size of the steam radiator to be installed.

A square foot of radiation is equivalent to a heat emission of 240 Btu per hour under standard conditions of steam at one pound gage pressure (215 F) and surrounding air at 70 F. With surrounding air at 80 F, the heat emission from a radiator will be less Under these conditions, the heat emission will not be 240 Btu per square foot of catalog rating per hour, but 240 C_s.

$$C_s = \left(\frac{t_s - t_i}{215 - 70}\right)^{1.3} = \left(\frac{215 - 80}{215 - 70}\right)^{1.3} = 0.912,$$

and 240 C_s = 240 × 0.912 = 218.5 Btu. Therefore, the size of the radiator to be selected shall have a catalog rating of 10,000 divided by 218 5 or 45 8 sq ft.

Chapter 31

STEAM HEATING SYSTEMS

Gravity and Mechanical Return, Gravity One-Pipe Air-Vent System, Gravity Two-Pipe Air-Vent System, One-Pipe Vapor System, Two-Pipe Vapor System, Atmospheric System, Vacuum System, Sub-Atmospheric System, Orifice System, Zone Control, Auxiliary Conditioning Unit, Condensation Return Pumps, Vacuum Pumps, Traps

THE essential features of the common type of steam heating systems are described in this chapter. They may be classified according to the piping arrangement, the accessories used, the method of returning the condensate to the boiler, the method of expelling air from the system, or the type of control employed. Information concerning the design and layout of steam heating systems will be found in Chapter 32.

GRAVITY AND MECHANICAL RETURN

In *gravity systems* the condensate is returned to the boiler by gravity due to the static head of water in the return mains. The elevation of the boiler water line must consequently be sufficiently below the lowest heating units and steam main and dry return mains to permit the return of condensate by gravity. The *water line difference*[1] must be sufficient to overcome the maximum pressure drop in the system and, when radiator and drip traps are used as in two-pipe vapor systems, the operating pressure of the boiler. This applies only to closed circuit systems, where the condensation is returned to the boiler. If the condensation is wasted, no water line difference is required.

In *mechanical systems* the condensate flows to a receiver and is then forced into the boiler against the boiler pressure. The lowest parts of the supply side of the system must be kept sufficiently above the water line of the receiver to insure adequate drainage of water from the system, but the relative elevation of the boiler water line is unimportant in such cases except that the head on the pump or trap discharge becomes greater as the height of the boiler water line above the trap or pump increases.

There are three general types of mechanical returns in common use, namely, (1) the mechanical return trap, (2) the condensation return pump, and (3) the vacuum return pump. Further information on pumps and traps will be presented later in this chapter.

GRAVITY ONE-PIPE AIR-VENT SYSTEM

In the gravity one-pipe air-vent system each radiator has but a single connection through which steam must enter and condensation must

[1]The *water line difference* is the distance between the water line of the boiler and the level of the water in the dry or wet return main (See Fig 4)

return in the opposite direction Each radiator has an individual air valve.

Up-Feed Gravity One-Pipe Air-Vent System

This system is the most common of all methods of steam heating, especially for small size installations, due largely to its low cost of installation and its simplicity. Where the size of the system is moderate or large, it cannot be assumed that these systems will be lower in cost than two-pipe systems using steam traps. In some instances it has been found that the cost of one-pipe systems under these conditions is greater owing to the higher cost of labor and materials due to the larger pipe sizes. As will be seen from Fig. 1, the steam piping rises to a point as high as possible at the boiler and pitches downward from this location until the far end of

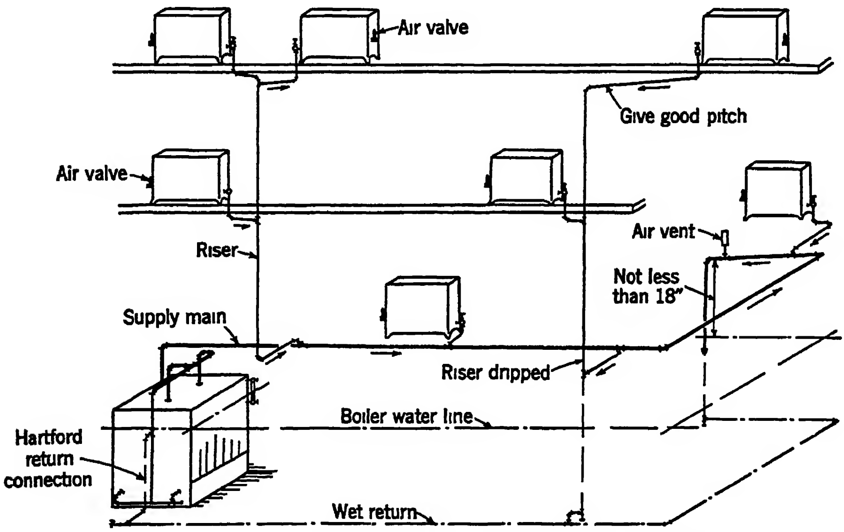


FIG 1. TYPICAL UP-FEED GRAVITY ONE-PIPE AIR-VENT SYSTEM

the main or mains is reached. At the far ends drips are taken off at the low points of the steam mains, are water-sealed below the boiler water line, and then brought back to the boiler in a wet return. Single pipe risers are branched off the main or mains to feed the radiators, the steam passing up the riser and the condensation flowing down it The steam and condensation flow in opposite directions in the riser but after the condensation enters the steam main it flows in the same direction as the steam and is disposed of through the drip connection at the end of the main. In buildings of several stories, it is customary to drip the heel of each riser separately, whereas in one- or two-story buildings this is not necessary. Both types of branches and risers are shown in Fig. 1.

Rapid elimination of air and condensation from the steam piping is essential to the successful operation of this system. It is therefore desirable that the venting and dripping of the steam main in long runs be

made at several intermediate points where the steam main may again be brought to a higher elevation.

It is desirable to install the air vent valves on the steam main about a foot ahead of the drips, as is indicated in Fig. 1 to prevent possible damage to the mechanism of the air vent valve by water, in case the valves are installed directly above the drips.

Horizontal branches to radiators and risers should be pitched at least ½ in. in 10 ft downward toward the riser or vertical pipe, and the horizontal branches from the steam main should be graded at least this amount toward the main, except where the heel of the riser is dripped, in which case the branch should pitch down toward the riser drip (Figs. 2 and 3). The return line, if wet, may be run without pitch or may be pitched in either direction, but if it is necessary to carry the return main overhead for any distance before dropping, the return should slope downward with the flow. It is desirable to install the wet return pipe with a pitch so that the system may be drained to prevent freezing in case the building remains unoccupied for a considerable length of time.

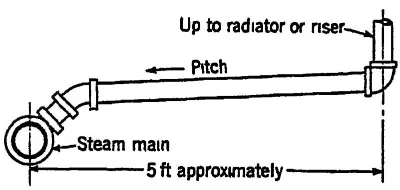


FIG. 2. TYPICAL STEAM RUNOUT WHERE RISERS ARE NOT DRIPPED

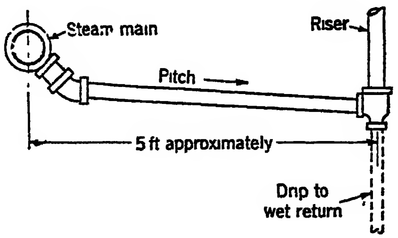


FIG. 3. TYPICAL STEAM RUNOUT WHERE RISERS ARE DRIPPED

The radiator valves may be of the angle-globe or gate type. They should not be of the straight-globe type because the damming effect of the raised valve seat interferes with the flow of condensation through the valve. Graduated valves cannot be used, as the steam valves on this system must be fully open or closed to prevent the radiators filling with water. Air valves may be manual or automatic, with or without a check to prevent the re-entrance of expelled air. Usually the automatic type is installed. An objection to one-pipe steam systems is that the heat is all on or all off, with no intermediate position possible. However, intelligent use of the on-and-off method of manual control gives reasonably satisfactory results. Improved systems and devices are now available which make it possible to obtain a modulating effect from one-pipe gravity heating systems.

It is important that the lowest points of the steam mains and heating units be kept sufficiently above the water line of the boiler to prevent flooding. Usually 18 in. is sufficient but construction limitations frequently make shorter distances necessary. The distance may be checked in the following manner:

Referring to Fig. 4 it will be seen that the water in the wet return is really in an inverted siphon, or U-shaped container, with the boiler steam pressure on the top of the

water at one end and the steam main pressure on the top of the water at the other end. The difference between these two pressures is the *pressure drop* in the system, *i.e.*, the friction of the steam in passing from the boiler to the far end of the main. The water in the far end will rise sufficiently to overcome this difference in order to balance the pressures, and it will rise enough farther to produce a flow through the return into the boiler (usually about 3 in. unless the pipes are small or full of sediment), and it will rise still farther if a check valve is installed in the return so as to obtain sufficient head to lift the tongue of the check (usually 4 in. will be necessary).

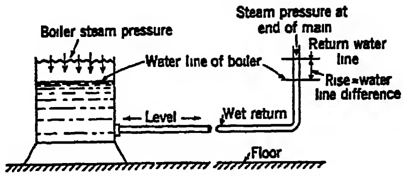


Fig. 4 Difference in Steam Pressure on Water in Boiler and at End of Steam Main

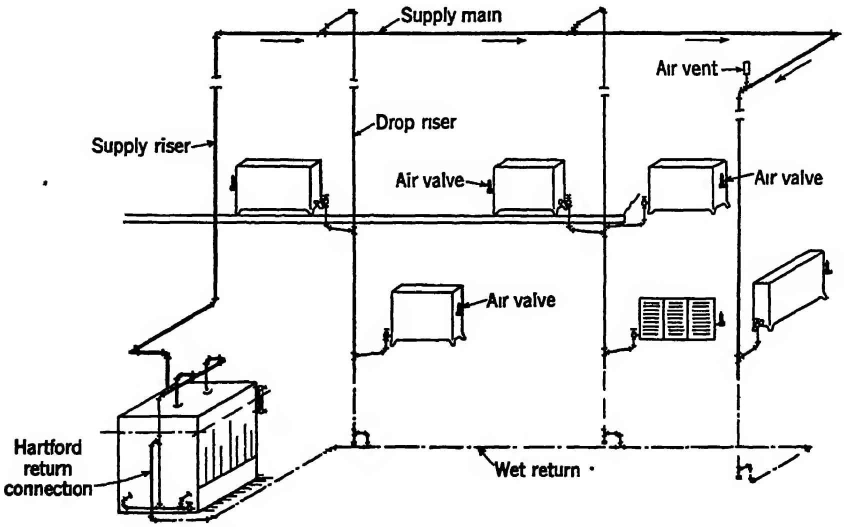


Fig. 5 Typical Down-Feed Gravity One-Pipe Air-Vent System

If a one-pipe steam system is designed, for example, for a total pressure drop of ⅛ lb, and utilizes an Underwriters' Loop[2] instead of a check valve on the return, the rise in the water level at the far end of the return due to the difference in steam pressure would be ⅛ of 28 in , or 3½ in. Adding 3 in. to this for the flow through the return main and 6 in. as a factor of safety gives 12½ in. as the distance the bottom of the lowest part of the steam main and all heating units must be above the boiler water line The same system, however, installed and sized for a total pressure drop of ½ lb, and with a check in the return, would require ½ of 28 in., or 14 in , for the difference in steam pressure, 3 in. for

[2]See discussion of piping details in Chapter 32.

the flow through the return, 4 in. to operate the check, and 6 in. for a factor of safety, making a total of 27 in. as the required distance Higher pressure drops would increase the distance accordingly

Down-Feed Gravity One-Pipe Air-Vent System

In the overhead down-feed gravity one-pipe air-vent system there is no change over the *up-feed system* in the radiators, the radiator valves, the air valves, or the radiator runouts as far back as the risers Beyond this point there are basic differences. The steam is taken from the boiler and carried to the top of the building as near the boiler as possible (Fig. 5). If the run to the main riser is long, or if the riser extends several stories in order to reach the top, the bottom of the riser should be dripped into the wet return. The horizontal main is taken off the top of the riser and grades down from the riser toward all of the drops, each drop taking its share of the main condensation (Fig. 6), or all of the drops except the last may be taken from the top of the main (Fig. 7), the last drop being from the bottom and serving as a drain for the entire main. As the overhead

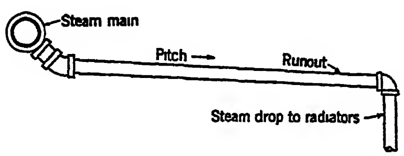


FIG. 6. STEAM RUNOUTS DRIPPING MAIN

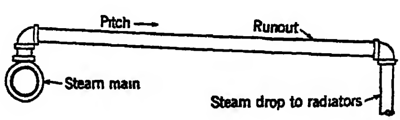


FIG 7. STEAM RUNOUTS WITH MAIN DRIPPED AT END ONLY

main does not carry any condensation from the radiators it is immaterial which method is used. The air vent shown on the main just before the last drop (Fig. 5) may be placed at this point or it may be located at the bottom of the drop under the last radiator connection and sufficiently above the water line of the boiler to prevent flooding.

GRAVITY TWO-PIPE AIR-VENT SYSTEM

The gravity two-pipe system is now considered obsolete although many of these systems are still in use in older buildings. Separate supply and return mains and connections are required for each heating unit; air valves are installed on the heating units and mains; hand valves are installed on the returns.

Up-Feed Gravity Two-Pipe System

This system (Fig. 8) has a steam and a return connection to each radiator. The radiator valves for steam, return, and air are the same as those described for the gravity one-pipe air-vent system. The steam main is run and pitched in the same manner as in the one-pipe system, but the returns from each radiator are connected into a separate return line system which has its risers carried down and joined to a wet return line under the boiler water line level. Where the return has to be kept high to function as a dry return, it is advisable to connect the return

risers to the dry return main through water seals about 36 in. deep, as shown in Fig. 9, to prevent steam from one riser entering another and closing the air valves on the nearest radiators.

Down-Feed Gravity Two-Pipe System

The steam main in the down-feed system is carried to the top of the building, and the piping of the steam side is arranged practically as in the down-feed one-pipe gravity system. The drips at the bottoms of the steam drops and the runouts to the radiators are similar to those shown in Fig. 8 for the up-feed gravity two-pipe system. On the return side of the system, the piping is arranged in exactly the same manner as the up-feed gravity two-pipe system.

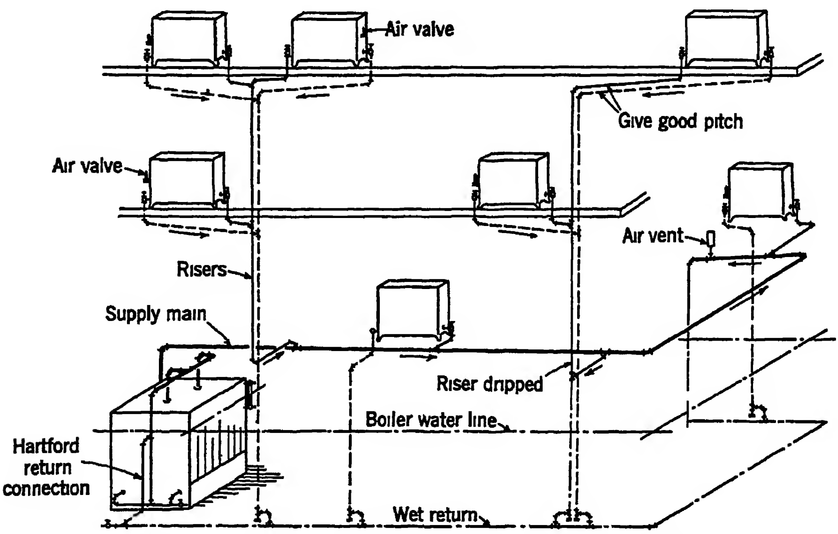


FIG. 8. TYPICAL UP-FEED GRAVITY TWO-PIPE AIR-VENT SYSTEM

ONE-PIPE VAPOR SYSTEM

A vapor system is one which operates under pressures at or near atmospheric and which returns the condensation to the boiler by gravity. The piping arrangement of a one-pipe vapor system is similar to that of the gravity one-pipe steam system; in fact, one-pipe gravity installations may readily be changed to one-pipe vapor systems by making a few simple alterations. The steam radiator valve is a plug cock which when opened gives a free and unobstructed passageway for water The automatic air valve is of special design to permit the ready release of air from the radiator and to prevent the return of the air after it is expelled. The air valves on the main are a quick relief type, and the whole system is designed to operate on a few ounces of pressure.

TWO-PIPE VAPOR SYSTEM

Two-pipe vapor systems may be classified as (1) *closed systems* consisting of those which have a device to prevent the return of air after it is once expelled from the system, and which can operate at sub-atmospheric pressures for a period of four to eight hours depending upon the tightness of the system and rate of firing, and (2) *open systems* consisting of those which have the return line constantly open to the atmosphere without a check or other device to prevent the return of air, and which operate at a few ounces above atmospheric pressure. The open systems have the disadvantage of not holding heat when the rate of steam generation is diminishing.

Under the first classification the essentials are packless graduated valves on the radiators, thermostatic return traps on the returns, and traps on all drips unless they are water sealed. Such a system, illustrated in Fig. 10, should be equipped with an automatic return trap to prevent the water from backing out of the boiler. In this up-feed arrangement the supply piping is carried to a high point directly at the boiler and is

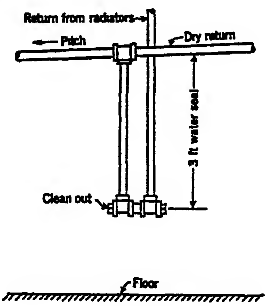


FIG. 9. METHOD OF CONNECTING TWO-PIPE GRAVITY RETURNS TO DRY RETURN MAIN

graded down toward the end or ends of the supply main, each supply main being dripped at the end into the wet return or carried back to a point near the boiler where it drops down below the boiler water line and becomes a wet return. From this main, runouts are branched off to feed risers or radiators above, these being graded back toward the steam main if they are not dripped at the bottom of the riser, or toward the riser if the riser heel is dripped. Both conditions are illustrated in Figs. 2 and 3.

Return risers are connected to each radiator on its return end through thermostatic traps. Their bottoms are connected to the return main through runouts which slope toward the main. The return main itself is sloped back toward the boiler if it is carried overhead; if run wet, the slope may be neglected, although it is desirable to slope the pipe so that the system may be drained. An air vent is installed at the point at which the return main drops below the water line. In the simplest cases this vent consists of a ¾-in. pipe with a check valve opening outward, but certain systems employ special patented forms of vent valves, designed to allow the air readily to pass out of the system and to prevent its return.

A check valve is inserted in the return main at a point near the boiler and a vertical pipe is run up into the bottom of the return trap, which usually is located with the bottom about 18 in. above the boiler water line. Some traps are constructed so that they will operate when they are installed with their bottom as close as 8 in. above the boiler water line. On the other side of this connection a second check valve is installed in the main return just before it enters the boiler (Fig. 11).

Down-Feed Two-Pipe Vapor System

In the down-feed two-pipe vapor system the steam is carried to the top of the building, the top of the vertical riser constituting the high point of

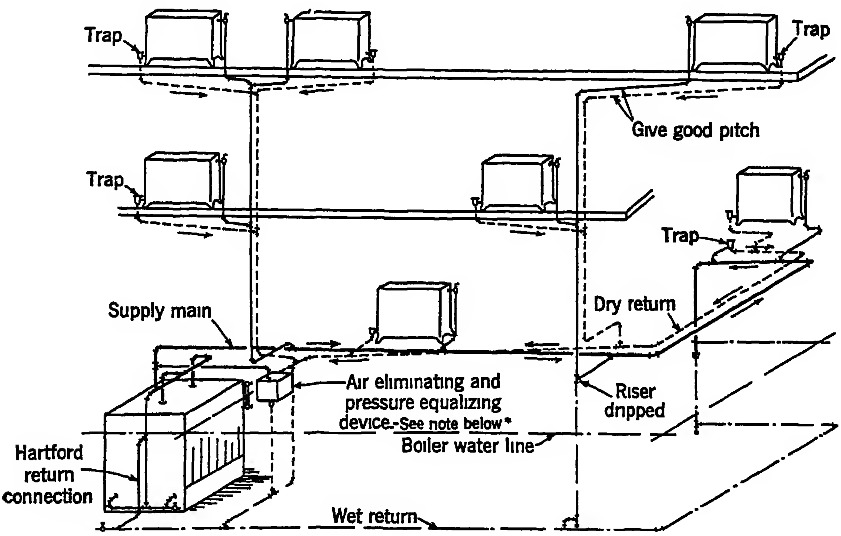


FIG. 10. TYPICAL UP-FEED VAPOR SYSTEM WITH AUTOMATIC RETURN TRAP[a]

[a]Proper piping connections are essential with special appliances for pressure equalizing and air elimination

the system, and the horizontal supply main is sloped down from this location to the far ends of each branch. The branches are taken off the main from the bottom or at a 45-deg angle downward, with the runouts sloped toward the drops (Fig. 6). Thus each branch from the main forms a drip and no accumulation of water is carried down any one drop. Another method of running the steam main, which is not considered as satisfactory but which is practical, is to take the branches off the top of the main (Fig. 7) and to drip the end of the main through the last riser, as illustrated in the down-feed one-pipe system detail shown in Fig. 6. If this is done, the pipe drop at the end or ends of the mains should be enlarged one pipe size to provide capacity for this concentration of the main drip.

The steam drops are carried down through the building with suitable reductions as the various radiator connections are taken off until the lowest radiator runout is reached. If the drop is only two or three stories high, the portion feeding the bottom radiator should be increased one

pipe size to provide for draining the riser, and if the drop is over three stories high it is well to increase the portion feeding the two lowest radiators one or two pipe sizes, especially if the two lowest radiators are small and the normal size of drop required is 1 in. or less. The bottom of the steam drops should terminate with a dirt pocket above which a drip trap connection is located, as shown in Fig. 12. The returns on a down-feed vapor system are the same as on an up-feed system except that every steam drop must have a drip at the bottom connected either into the return through a trap or into a separate water-sealed drip line below the boiler water line, as illustrated in Fig. 10, in which case the thermostatic

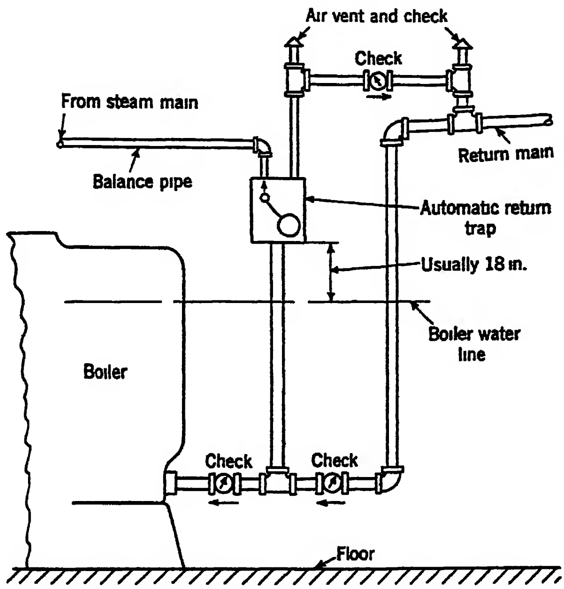


FIG 11. TYPICAL CONNECTIONS FOR AUTOMATIC RETURN TRAP

traps may be omitted. The runouts to the radiators and the radiator connections of the down-feed system are the same as those of the up-feed system already described.

ATMOSPHERIC SYSTEM

The distinguishing features of the atmospheric system are gravity return to the boiler or to waste, graduated or ordinary radiator valves, no automatic air valves on the radiators, thermostatic traps on the radiator returns, and the venting of all air from the system by means of pipes open to the atmosphere. The returns are open to the atmosphere at all times, usually by extending the return risers to the top of the building where they are either connected together in groups and carried through the roof or extended through the roof individually. Atmospheric systems, either up-feed or down-feed, are often used where the condensation is not returned to the boiler, as in heating systems supplied by high pressure

steam through pressure-reducing valves at locations far from the boilers. The returns may be delivered back to the boiler, if desired, by condensation return pumps which are vented to the atmosphere. The return lines in such systems are simply gravity waste lines in which the condensation flows entirely by gravity and is not aided by any pressure difference.

Atmospheric systems contemplate maintaining a practically constant pressure in the steam pipe and atmospheric pressure in the return pipe. When graduated steam valves are provided, they enable the occupant of a room to vary the flow area to the radiator so as to obtain a greater or lesser heating effect.

The steam side may be run as that for either up-feed or down-feed two-pipe vapor systems, as the conditions require, and the radiator connections are the same as for vapor systems in that they have graduated

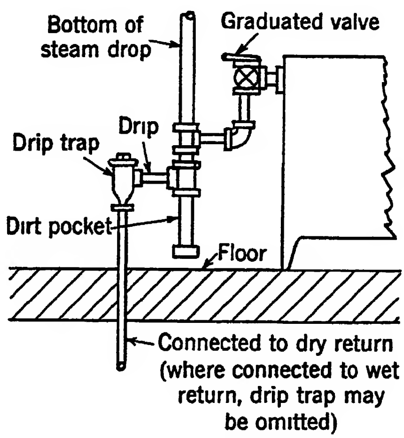


Fig. 12. Detail of Drip Connections at Bottom of Down-Feed Steam Drop

valves on the radiator supply ends and thermostatic traps on the radiator return ends. All drips from the supply main and the steam side of the system must pass through thermostatic drip traps before entering the return system where only atmospheric pressure exists. Fig. 13 illustrates a typical scheme of piping used on atmospheric systems.

VACUUM SYSTEM

In the vacuum system, a vacuum is maintained in the return line practically at all times but no vacuum is carried on the steam side, and the usual accessories include graduated valves on the radiator supply and thermostatic traps on the radiator return. The air is expelled from the system by a vacuum pump and all drips must pass through thermostatic traps before connecting to the return side of the system.

These systems are often fed from high pressure steam mains through pressure-reducing valves but they may be fed direct from a low-pressure steam heating boiler as shown in Fig. 14, in which a typical up-feed

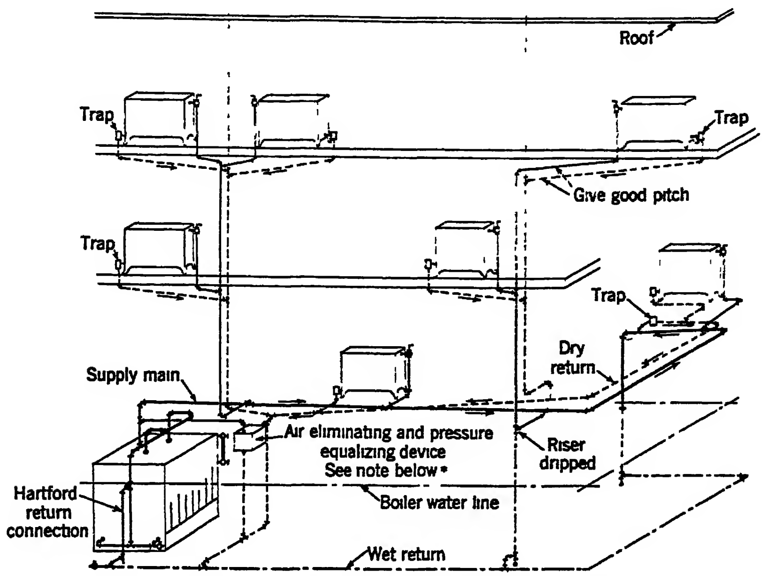


FIG. 13. TYPICAL ATMOSPHERIC SYSTEM WITH AUTOMATIC RETURN TRAP[a]

[a]Proper piping connections are essential with special appliances for pressure equalizing and air elimination

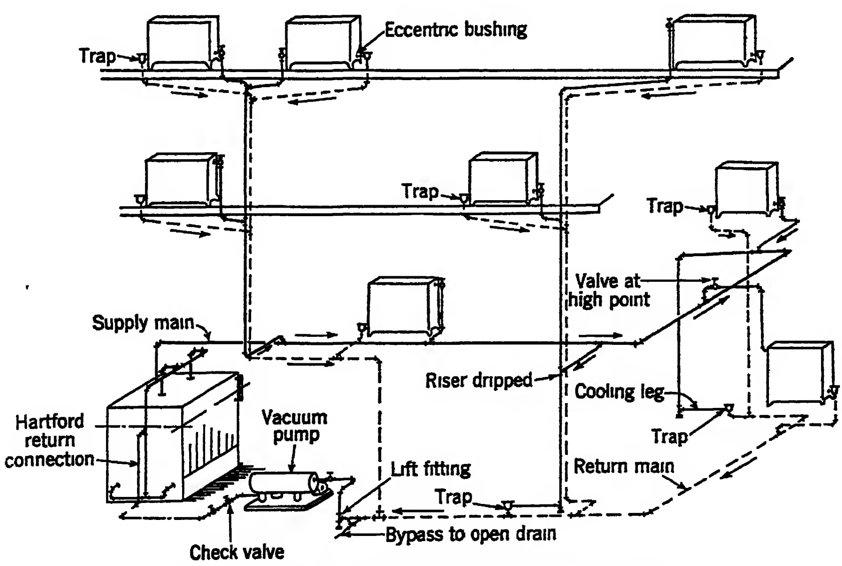


FIG. 14. TYPICAL UP-FEED VACUUM PUMP SYSTEM

vacuum system is illustrated. The supply main slopes down in the direction of flow; the runouts pitch down toward the riser if the riser is dripped (Fig. 3) or up toward the riser if the riser is not dripped (Fig. 2); both conditions are indicated in Fig. 14. The matter of dripping the risers depends largely on the height of the riser and the judgment of the designer. Ordinarily risers less than three stories high are not dripped and those more than four stories high are dripped, but there is no set rule for this. When risers are dripped the runouts from the steam main may be taken from the bottom if desired and each runout then serves as a drip for the main.

The risers are carried up to the highest radiator connection and are connected to the radiator through runouts sloping back toward the riser. The radiators usually have graduated valves on the supply end, although

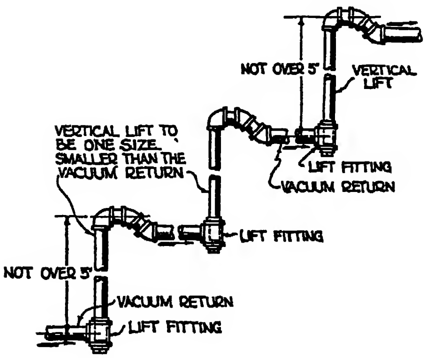


Fig. 15. Method of Making Lifts on Vacuum Systems when Distance is Over 5 ft

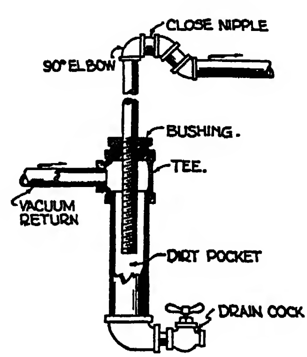


Fig. 16. Detail of Main Return Lift at Vacuum Pump

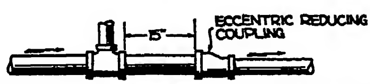


Fig. 17 Method of Changing Size of Steam Main when Runouts are Taken from Top

this is not absolutely necessary. Angle-globe valves and gate valves may be used where graduated manual control is not desirable. The return valves must be of the thermostatic type which will pass air and water but which will close against the passage of steam.

The return risers are connected in the basement into a common return line, which slopes downward toward the vacuum pump. The vacuum pump discharges the air from the system and pumps the water back to the boiler, or other receiver, which may be a feed-water tank or a hot well. It is essential on these systems that no connection from the supply side to the return side be made at any point except through a trap.

While the best practice demands a return flowing to the vacuum pump in an uninterrupted downward slope, in some cases limitations make it necessary to drop the return below the level of the vacuum pump inlet before the pump can be reached. In such event one of the advantages of

the vacuum system is that the return can be raised by the suction of the vacuum pump to a considerable height, depending on the amount of vacuum maintained, by means of a lift fitting inserted in the return. Best practice dictates that the lift should be limited to a single lift connection at the entrance to the vacuum pump and that lifts scattered throughout the system be avoided When the lift is considerable, several lift fittings are used in steps (Fig. 15), more successful operation being obtained by this method than when the lift is made in one step. If the lift occurs close to the vacuum pump, a special arrangement is used as shown in Fig. 16. It is desirable that means be provided for draining manually the low points of the lift fittings to eliminate from the return piping all water in danger of freezing in case the system is shut down for a considerable length of time.

Down-Feed Vacuum System

The piping arrangement for the down-feed vacuum system is similar on the supply side to the down-feed vapor system in that it has similar runouts, radiator valves, drips on the bottom of the steam drops, and enlargement of the drops for the lower radiator connections. The return side of the system is exactly the same as the up-feed system except that the steam riser drips at the bottom are connected into the return line through thermostatic traps. It is preferable to take the runouts for the risers from the bottom or at a 45-deg angle down from the steam main (Fig. 6) so that they may serve as steam main drips. When this is done it is practical to run the steam main level if a runout is located at every change in pipe size, or if eccentric fittings are used (Fig. 17). A slight pitch in the steam main, however, should be used when possible. An overhead vacuum down-feed system is shown diagrammatically in Fig. 18.

SUB-ATMOSPHERIC SYSTEMS

Sub-atmospheric systems are similar to vacuum systems, but in contrast provide temperature control by variation of the heat output from the radiators both by varying the pressure at which steam is circulated in the radiation and the amount of steam. The steam supply is continuous at varying rates. A vacuum pump capable of operating at high partial vacua is preferable since the higher the vacuum the greater is the accuracy in the distribution of steam through the system, particularly in mild weather. A pump capable of producing up to 25 in. of vacuum on the system is used in such cases. A controller is placed on the pump so that the vacuum or absolute pressure carried in the returns can be maintained at a certain amount below that existing in the line to insure circulation.

The traps are designed to operate in high vacuum. It is apparent that this system differs from the ordinary vacuum system by having a vacuum on both sides of the system, instead of only on the return side, in order to secure control of the heat emission from the radiators and thus to control the temperature in the building. These systems permit the heat output from the steam mains and risers to be diminished as the weather becomes milder, thus giving control to this portion of a heating system. The system can be operated in the same manner as the ordinary vacuum system when desired.

In the vacuum system, steam pressure above that of the atmosphere exists in the supply mains and radiators practically at all times. In the sub-atmospheric system, steam pressure exists in the steam main and radiators only during the most severe weather, while under average winter temperatures the steam is under a partial vacuum which in mild weather may reach as high as 25 in. after which further reduction in heat output is obtained by partially filling the radiation with steam.

This vacuum is partially self-induced by the condensation of the steam in the system due to the supply of steam being furnished through the control which admits it, and it being proportioned to balance the existing heat loss. In the sub-atmospheric system, a control valve is inserted on

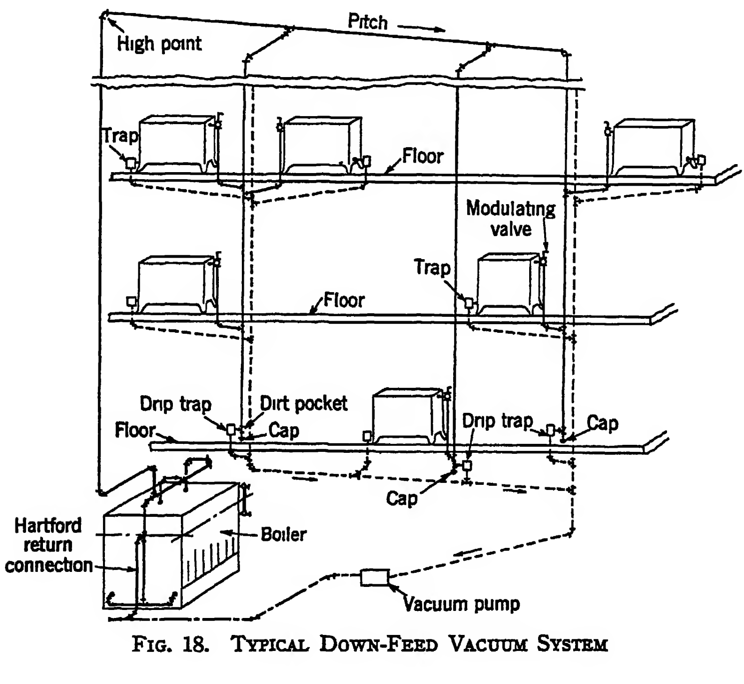


FIG. 18. TYPICAL DOWN-FEED VACUUM SYSTEM

the steam main of an ordinary vacuum system near the boiler or the boiler is automatically controlled, a high-vacuum pump is substituted for the ordinary type and is supplied with a pressure-difference control, and traps are placed on the radiators and drips which will operate satisfactorily at any pressure from 5 lb gage to 26 in. of vacuum.

The control valve is either a special pressure-reducing valve which may be controlled manually, or a control valve or combustion equipment which may be operated thermostatically from points selected in the building. The vacuum pump regulator is simply a diaphragm so arranged that, when the vacuum in the return line is insufficient to hold the desired difference in pressure between the steam and return sides of the system, the vacuum pump is automatically started and the vacuum increased to the necessary amount. The actual pressure difference main-

tained between the two sides of the system is only enough to secure adequate circulation and is often about 2 in. of mercury. This fixed pressure difference between the supply and return sides of the system results in practically constant circulation under all pressure conditions.

In order to distribute the steam equally when the system is being warmed up and also to reduce the amount of steam delivered to the radiators on mild days, orifice plates are used in the graduated radiator control valves. The heat emitted from the radiators in mild weather and under conditions of high vacuum is not only reduced in proportion to the difference in the steam temperature between that for 2 lb gage and for 25 in. of vacuum but it is reduced still further by a reduction in the amount of steam which can pass through the orifice when the steam is expanded due to the vacuum. This renders possible the control of heat emission from the radiators to a point not indicated entirely by the difference in steam temperatures, but far beyond it.

Sub-atmospheric operation has advantages even where individual thermostatic radiator control is installed. By operating the system with steam temperatures in parallel with the outside temperature requirements, a large part of the load is removed from the temperature control system, it makes fewer operations and the radiator follows an even temperature without fluctuating from extreme hot to extreme cold.

The high-vacuum pumps on this system are equipped with receivers having float control so that the pump can be placed on a receiver-return-pump basis at night if desired so no high vacuum will be carried. One radical difference between this system and the ordinary vacuum system is that no lifts can be made in the return line, except at the vacuum pump. The returns must grade downward constantly and uninterruptedly from the radiator return outlet to the inlet on the high-vacuum pump receiver.

No attempt should be made to heat service water on this system unless the steam line for water heating is taken off the boiler header back of the heating system control valve, and then only when 2 lb or more will be carried on the boiler at all times.

ORIFICE SYSTEM

Orifice systems of steam heating may have piping arrangements identical with vacuum systems but some of these systems omit both the radiator thermostatic traps and the vacuum pump in cases where the returns are wasted to a sewer or delivered to some type of receiver in which no back pressure exists. The principle on which they operate is embodied in the well-known fact that an orifice will deliver varying velocities when the ratio of the absolute pressures on the two sides of the orifice exceeds 58 per cent. If the absolute pressure on the outlet side is less than 58 per cent of the absolute pressure on the inlet side no further increase in velocity will be obtained.

As a result, if an orifice is so designed in size as to exactly fill a radiator with steam at 2-lb gage on one side and ¼-lb gage on the other, the absolute pressure relation is

$$\frac{14.7 + 0.25}{14.7 + 2.0} = 90 \text{ per cent}$$

Should the steam pressure be dropped to ¼-lb gage, the pressure on each side of the orifice would be balanced and no steam flow would take place. From this it will be seen that if an orifice of a given diameter will fill a given radiator with steam when there is a given pressure on the main, it is simply a question of dropping this main pressure provided the supply pipe pressures be controlled sufficiently closely, so as to fill any desired portion of the radiator down to the point where the main pressure equals the back pressure in the radiator, at which time no steam will be supplied at all. If orifices throughout a system are designed on a similar basis, all radiators will heat proportionately to the steam pressure within the limits for which the orifices are designed.

Some systems use orifices not only in radiator inlets but also at different points on the main, thus balancing the system to a greater extent. For example, the system may be designed for a particularly long run involving an initial pressure of 3-lb gage on the main and 2 lb at the end of the main, but each branch from the main may have an orifice for reducing the pressure at it to 2-lb gage. This is particularly useful for branches near the boiler where the drop in the main has not yet been produced.

Orifice systems using a vacuum pump operate successfully with the ordinary low vacuum type of pump producing 8 to 10 in. of vacuum. They are controlled by various means to regulate the steam pressure. One method is by a thermostat located on the roof to govern the steam pressure by a combination of outside and inside temperatures; another, useful on systems without traps and vacuum pumps, controls the steam pressure manually from temperature indication stations in the building, or automatically by a thermostatically-controlled pressure reduction valve or draft regulator on the boiler; with oil or gas firing, the on-and-off control or a boiler pressure control may be used.

ZONE CONTROL

Certain portions of a building may require more heat at times than others but if the whole building is on one general control, such as would occur with a single piping system with an on-and-off control or with the sub-atmospheric or the orifice systems, it would be necessary to supply sufficient heat to accommodate the coldest portion of the building even though some sections would be overheated. By separation of a building into zones each with its own piping system, each zone of the building may be controlled separately.

The sides of the building with different exposures should be considered first, because of the varying effects of the wind and sun. With the prevailing winter winds from the northwest, a simple zoning would place the north and west sides of the building on one system and the south and east sides on another. If the building is large enough to justify the expenditure, a better arrangement would be to place all north walls on one zone, all west walls on a second, all east walls on a third, and all south walls on a fourth.

In case of high buildings, the lowest 8 or 10 stories may be well protected from wind by surrounding buildings, the next 10 stories may have moderate exposure, and above this there may be an unobstructed exposure to gales. On still days the heat demands vertically will vary little, but on

windy days there will be a marked difference in the heat requirements for the different horizontal sections. In addition, the *chimney effect* caused by the difference in density between the warm air on the inside of a building and the colder air on the outside will give an air movement which will require zoning to correct. Where such conditions are encountered, the building should be divided horizontally as well as vertically. An arrangement of this character would give 12 zones· namely, north, east, south, and west lower zones; similar middle zones; and similar top zones. Each zone should constitute an individual and separate system of piping with its own supply steam valve (controlled by thermostats in its respective zone) and with its own return or vacuum pump, if one is used. Certain interior areas, such as basements, light well walls and other

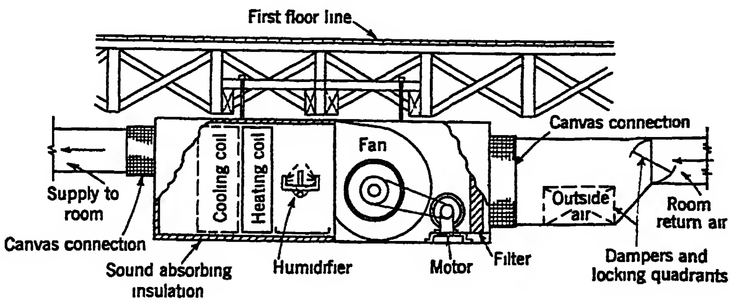


FIG 19. RESIDENTIAL CONDITIONING UNIT

locations where sun and wind do not affect the conditions, should be placed in still another zone if the most economical results are to be secured.

Zoning has advantages even where individual thermostatic radiator control is installed whether this be of pneumatic, electric, or the self-contained radiator valve type. By operating each zone to supply heat in parallel with its outside temperature and wind fluctuations, a large part of the load is taken off the thermostatic controls; they operate less frequently and the radiators follow a more even temperature instead of fluctuating from extreme hot to extreme cold.

Sub-atmospheric, orifice, and zone control systems, generally are proprietary.

AUXILIARY CONDITIONING UNIT

In connection with a residential steam or hot water system using radiator or convector heating a unit as shown in Fig. 19, is available to supplement the old or new system. The unit is arranged in a sheet metal enclosure with a filter, circulating fan, means for adding moisture to the air, heating or tempering coil and generally provisions are made for the addition of a cooling coil in case summer air circulation is desired. The unit is frequently located on the ceiling of the basement and is connected with one or more supply in return air ducts in the various rooms. In some cases, provisions are made for the introduction of a portion of the

outside air to the system and dampers are included to adjust the desired air quantities.

The heating coil of the unit may be connected to a steam or hot water boiler system and is adaptable for operation with a one-pipe, two-pipe or vacuum system. The cooling coil may be connected to a source of refrigeration, or in some cases city water is circulated through the coil when 58 F or lower temperature water is available. The amount of moisture released is adjustable depending upon the degree of humidification desired. The complete unit may be adapted to various automatic control arrangements to satisfy the comfort demands of the occupants.

CONDENSATION RETURN PUMPS

Condensation return pumps are generally required when the elevation of the boiler with respect to the heating units is such that the condensate will not return by gravity, or when the boiler pressure is greater than that supplied the heating units, as in a high-pressure boiler installation supplying steam through a reducing valve to the heating units. The condensate is commonly returned by gravity to a receiver, vented to the atmosphere, from which it flows to the pump.

Condensation return pumps are assembled with tank or receiver and arranged for either continuous operation or for automatic starting and stopping by float control. Any style of water pump may be employed for this service, the power available determining whether the mode of drive shall be steam or electric. The motor-driven, automatic, centrifugal pump and receiver has found wide acceptance for low pressure heating systems.

Fig. 20 shows a typical installation using an automatic condensation return pump and vented receiver. A float control operates the pump whenever sufficient water accumulates in the receiver. Condensation return pumps are suitable for use on systems in which the returns are under atmospheric pressure. These include atmospheric systems, orifice systems with open returns, and certain types of vapor systems which operate within a few ounces of atmospheric pressure, but ordinarily do not carry any sub-atmospheric pressure. They may also be used on one-pipe and two-pipe gravity steam systems with a proper arrangement for venting the receiver. In discharging to waste, there is no object in using a condensation pump unless the discharge must be elevated.

VACUUM PUMPS

A vacuum heating pump is employed to create a vacuum on the return side of a system to remove air and water from the return piping and to pump the condensate to the boiler or to a receiving tank. Pumps of this classification may be driven by steam or electricity; they may be continuous in operation, or automatic with float or vacuum control in one or more combinations.

For rating purposes[3], vacuum pumps are classified as *low vacuum* and *high vacuum*. Low vacuum pumps are those rated under operation at 5½-in. mercury vacuum, and high vacuum pumps are those rated at vacuums above 5½ in.

[3]See A S H V E. Standard Code for Testing and Rating Return Line Low Vacuum Heating Pumps

Return line vacuum pumps are classified in the method of their performance as follows

a. Those which perform the function of air separation under atmospheric pressure.
b Those which perform the function of air separation under a partial vacuum.

Pumps coming under the first classification will handle vacuum steam system condensation coming back by gravity at any temperature up to 205 F without either the sealing or the hurling water flashing into steam. These pumps, to operate under a combined water level and vacuum control, must be equipped with a float-control receiver between the vacuum pump and the system, but where they are intended for continuous operation, they do not require a receiver. Such pumps employ a single vacuum

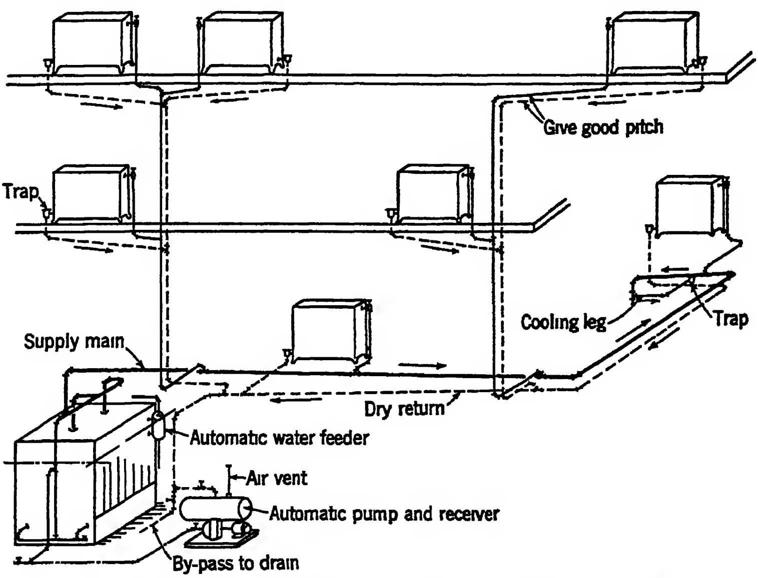


FIG. 20. TYPICAL INSTALLATION USING CONDENSATION PUMP

producer which removes the condensate and air from the system and delivers it into a separating chamber under atmospheric pressure from which the condensate is delivered to the boiler or feed water heater. They are constructed on one of the following evacuating and discharge principles:

1. Hydraulic vacuum producer with one pump impeller.
2. Hydraulic vacuum producer with two pump impellers.
3. Water displacement vacuum producer with two pump impellers.
4. Piston displacement vacuum producer with one pump piston.

The second classification of pumps will handle vacuum steam system condensation coming back by gravity at any temperature not exceeding 190 F without the flashing into steam of either the sealing or the hurling water. In order to operate under a combined water-level and vacuum control, these pumps must be equipped with a float-control receiver

between the vacuum pump and the system; where intended for continuous operation they do not require a receiver. Such pumps employ a vacuum producing impeller which removes air from the receiver or heating system under a partial vacuum and delivers it through an air separator against atmospheric pressure. The condensate is removed from the receiver under a partial vacuum by a separate impeller and is delivered to the boiler or feed water heater. For evacuating and discharge, a water displacement vacuum producer with two pump impellers is used.

Receiver Capacities for Vacuum Pumps

Where receivers are used in connection with vacuum pumps there is a definite relation between the capacity of the receiver and the capacity of the pump. The receiver should have a capacity of not less than 1½ times the volumetric quantity of condensation per minute and should not have such a capacity that the pump will empty the receiver in less than half a minute. Receivers of larger capacities will result in less frequent periods of operation.

Piston Displacement Vacuum Pumps

Piston displacement return-line vacuum heating pumps may be either power or steam driven. They should be provided with mechanical lubricators and their piston speed in feet per minute should not exceed 20 times the square root of the number of inches in their stroke. While the volumetric displacement for such pumps was formerly figured at 8 to 10 times the volumetric flow of condensation to be handled, the more efficient thermostatic traps used today in connection with vacuum heating systems make it possible to change this proportion so that the volumetric displacement of these pumps may not be less than 6 times the volume of condensation.

Vacuum Pump Controls

In the ordinary vacuum system the vacuum pump is controlled by a vacuum regulator which cuts in when the vacuum drops to the lowest point desired and which cuts out when the vacuum has been increased to the highest point. This is done largely to eliminate the constant starting and stopping of the vacuum pump which would occur if the vacuum were maintained constant. In addition to this control, a float control is included which will automatically start the pump whenever sufficient condensation accumulates in the receiver, regardless of the vacuum in the system. This arrangement makes the vacuum pump primarily a condensation pump and secondarily an air pump.

On the sub-atmospheric systems the high vacuum pump is controlled by a differential regulator which keeps the vacuum in the return line always a few inches higher than that in the steam line and in the radiators.

TRAPS

Traps are used for draining the condensate from radiators, steam piping systems, kitchen equipment, laundry equipment, hospital equipment, drying equipment and many other kinds of apparatus. The usual

functions of a trap are to allow the passage of condensate and to prevent the passage of steam. In addition to these functions, traps are frequently required to allow the passage of air as well as condensate. Traps are also required to allow the passage of air and to prevent the passage of either water or steam, or both.

In addition, traps are used for returning condensate either by gravity, by steam pressure, or by both, to a boiler or other point of disposal, and for lifting condensate from a lower to a higher elevation, or for handling condensate from a lower to a higher pressure.

The fundamental principle upon which the operation of practically all traps depends is that the pressure within the trap at the time of discharge shall be equal to, or slightly in excess of, the pressure against which the trap must discharge, including the friction head, velocity head and static head on the discharge side of the trap. If the static head is in favor of the trap discharge it is a minus quantity and may be deducted from the other factors of the discharge head.

Traps may be classified according to the principle of operation as (1) float, (2) bucket, (3) thermostatic, or (4) tilting traps.

Float Traps. A discharge valve is operated by the rise and fall of a float due to the change of water level in the trap. When the trap is empty the float is in its lowest position, and the discharge valve is closed. A gage glass indicates the height of water in the chamber.

Unless float traps are well made and proportioned there is danger of considerable steam leakage through the discharge valve due to unequal expansion of the valve and seat and the sticking of moving parts. The discharge from a float trap is usually continuous since the height of the float, and consequently the area of the outlet, is proportional to the amount of water present.

Bucket Traps. Bucket traps are of two types, the upright and inverted, and although they are both of the open float construction, their operating principle is entirely different. In the *upright bucket* trap, the water of condensation enters the trap and fills the space between the bucket and the walls of the trap. This causes the bucket to float and forces the valve against its seat, the valve and its stem usually being fastened to the bucket. When the water rises above the edges of the bucket it flows into it and causes it to sink, thereby withdrawing the valve from its seat. This permits the steam pressure acting on the surface of the water in the bucket to force the water to a discharge opening. When the bucket is emptied it rises and closes the valve and another cycle begins The discharge from this type of trap is intermittent.

In the *inverted bucket* trap, steam floats the inverted submerged bucket and closes the valve. Water entering the trap fills the bucket which sinks and through compound leverage opens the valve, and the trap discharges. It is impossible to install a water gage glass on an inverted bucket trap, but if visual inspection is necessary, a gage glass can be placed on the line leading to the trap. No air relief cocks can be used but this is unnecessary, as the elimination of air is automatically taken care of by air passing through the vent in the top of the inverted bucket regardless of temperature.

Thermostatic Traps. Thermostatic traps are of two types, those in which the discharge valve is operated by the relative expansion of metals, and those in which the action of a volatile liquid is utilized for this purpose. Thermostatic traps of large capacity for draining blast coils or very large radiators are called *blast* traps.

Tilting Traps. With this type of trap, water enters a bowl and rises until its weight overbalances that of a counter-weight, and the bowl sinks to the bottom. As the bowl sinks, a valve is opened thus admitting live steam pressure on the surface of the water and the trap then discharges. After the water is discharged, the counter-weight sinks and raises the bowl, which in turn closes the valve and the cycle begins again. Tilting traps are necessarily intermittent in operation. They are not ordinarily equipped with glass water gages, as the action of the trap shows when it is filling or emptying. The air relief of tilting traps is taken care of by the valves of the trap.

Thermostatic traps are generally used for draining radiators and heaters, except for very large capacities where bucket, float or blast-type thermostatic traps are used. Thermostatic traps for this service usually pass both condensate and air and in the case of float and upright bucket traps the air is usually relieved through an auxiliary thermostatic trap in a by-pass around the main trap. Sometimes this auxiliary air trap is an integral part of the trap. Such traps are termed float and thermostatic traps.

Blast-type thermostatic traps are sometimes used on vacuum heating systems for connecting old one- or two-pipe gravity systems in parallel with vacuum return line systems, in which case the blast-type thermostatic traps should not be provided with auxiliary air by-pass, as the action of this will allow the vacuum to draw air into the old system through its air valves, especially when the steam is wholly or partially cut off. The air from the returns of such old systems should be relieved just ahead of the traps by means of quick-venting automatic air valves,

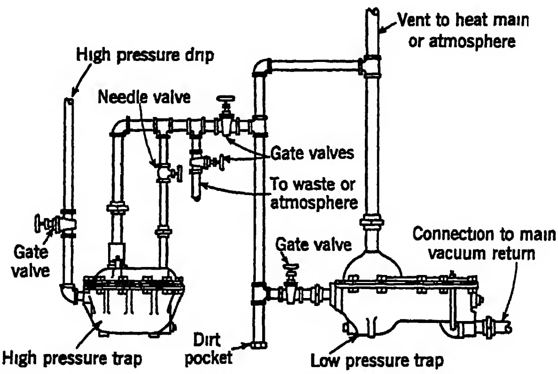


Fig 21. Method of Discharging High-Pressure Apparatus into Low-Pressure Heating Mains and Vacuum Return Mains through a Low-Pressure Trap

preferably of the non-return type, especially if the other air valves on the old system are non-return valves.

Tilting traps used for discharging to a higher or a lower pressure are provided with two or three valves operated by the action of the trap. In the case of the two-valve tilting traps, one valve closes a steam inlet and the other valve opens a vent outlet while the trap is filling, and as soon as the trap dumps, the first valve opens the steam inlet and the second valve closes the vent outlet, while the trap discharges. In this type of trap there must be a swinging check-valve on each side of the trap, in addition to the usual by-pass, to prevent the pressure in the trap, while discharging, from backing up through the inlet and the pressure in the discharge line from backing up into the trap while it is filling. This type of trap will blow steam out through the vent while filling, if the pressure on the inlet side is sufficient, and should not be used, therefore, with such pressures unless the vent is properly piped back into the return to a feed water heater, a condenser or a perforated pipe in the bottom

of the receiver to which the trap discharges in such a way as to prevent the escape of the steam that comes in with the condensate and passes through the vent. In the three-valve traps of this type there is an extra

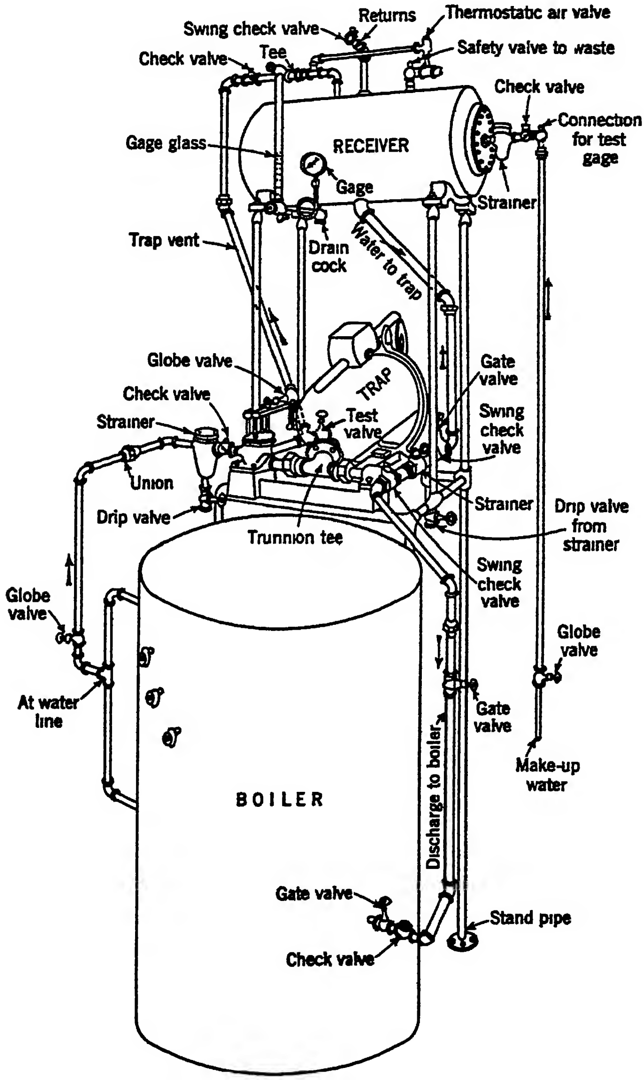


Fig. 22. Return Trap and Receiver for Automatic Boiler Feed

valve for closing the discharge while the trap is filling with condensate. High pressure traps should not discharge directly into a vacuum return because of the vapor formed by the re-evaporation of a part of the hot

condensation. Fig. 21 shows a method which may be used for disposing of the greater part of the vapor of re-evaporation. An expansion chamber often is installed between the high- and low-pressure traps.

Automatic Return Traps

In the general heating plant, where thermostatic traps are installed on the heating units, it becomes necessary to provide a means for returning the water of condensation to the boiler, if a condensation or vacuum pump is not used. When the return main can be kept sufficiently high above the boiler water line for all operating conditions, the water of condensation will flow back by gravity, and no mechanical device is required. But actually this does not work out in practice. It follows, therefore, that a direct return trap is needed for the handling of the condensation even though it may not be called into action except under some operating condition where the pressure differential exceeds the static head provided. The installation of a direct return trap assures safety for such systems, and guarantees the operation of the plant under varying conditions.

Automatic return traps, sometimes called alternating receivers, may be of the counter-balanced, tilting type, or spring actuated. These consist of a small receiver with an internal float, and when the condensate will not flow into the boiler under pressure, it will feed into the receiver of the trap, and in so doing, raise or tilt the float or mechanism which actuates a steam valve automatically. This admits steam to the receiver, at boiler pressure, and the equalizing of the pressures which follows allows the water to flow into the boiler. Fig. 22 shows a direct return tilting trap and receiver properly connected for automatically feeding a boiler from a system of returns delivering the condensate to the receiver.

PROBLEMS IN PRACTICE

1 ● What is meant by water line difference in a gravity steam heating system?

The water line difference is the distance between the level of the water in the dry or wet return and the boiler water line This difference is equivalent to the pressure required to overcome the maximum drop in the system and the operating pressure of the boiler.

2 ● How many types of common mechanical returns are there and what are they?

Three: (1) the mechanical return trap, (2) the condensation return pump, and (3) the vacuum pump.

3 ● In the ordinary vacuum system of steam heating, where does the vacuum usually exist?

On the return side of the system only, between the radiator trap and the vacuum pump If the radiator supply valve is closed off, the vacuum may extend back through the radiator as far as the supply valve, if an inadequate supply of steam is furnished to the system, some vacuum may be developed in the steam main, but neither of these can be termed *normal operation*

4 ● What is the distinction between the open and the closed vapor systems?

The open vapor system has the return line always open to the atmosphere, while the closed vapor system has an automatic device on the air vent so that air once expelled from the system through the vent cannot re-enter via this route.

5 ● On a vacuum system, what device must be placed on all drips before they enter the vacuum return line?

A thermostatic drip trap or occasionally, where large volumes of condensation are to be handled, a float trap

6 ● How does the sub-atmospheric system differ in operation from the ordinary vacuum system?

The ordinary vacuum system has pressure in the steam line, and a vacuum produced by the vacuum pump in the return line, usually varying between 5 and 10 in of mercury The sub-atmospheric system may have either a vacuum or pressure on the steam and return lines, but a constant difference in pressure is maintained between the lines regardless of what pressure or vacuum may be carried The vacuum, which is generally produced jointly by condensation and the exhausting action of the pump, in the system under conditions of throttled steam supply, will run much higher than in the ordinary vacuum system, and as high as 25 in of mercury in the radiators

7 ● What is generally understood by zoning in building steam heating systems?

Zoning is a term applied to the placing of certain sections of a building on a single temperature control instead of having either individual room control or a single temperature control governing the whole building Zones may be horizontal, such as a single story, a basement, or an attic, or vertical such as the north side, or the west side

8 ● Why does the water line in the far end of a wet return in a gravity steam system rise higher than the water line in the boiler?

The friction of the steam flowing through the steam main from the boiler to the far end of the system and the pressure reduction resulting from the condensing action of the radiators causes a drop in steam pressure at the point where the wet return is connected, consequently, the steam pressure on top of the water in the wet return is less than the steam pressure on top of the water in the boiler, so the water in the end of the wet return rises until a balanced condition is set up

9 ● On gravity one-pipe systems as indicated in Fig. 1 and Fig. 3, why is the drip on the steam runout connected to wet return?

Because if it were connected to dry return, the pressure drops to two different points would not necessarily be the same and the system would short circuit

10 ● What is the function of the automatic return trap?

To insure the return of condensate to the boiler when the operating condition is such that the boiler pressure exceeds the static head on the returns.

11 ● What advantage is there to an air valve with a check to prevent the re-entrance of expelled air?

A system equipped with such valves builds up a vacuum and holds the heat longer. With proper controls on the boiler, lower radiator temperatures can be maintained in mild weather, giving better plant efficiency.

12 ● With a one-pipe steam heating plant designed for a total pressure drop of ¼ lb with a check valve on the return, how high must the lowest part of the steam main be above the boiler water line?

Water line difference (¼ × 28)	7 in.
Flow head required	3 in.
Friction head of check valve	4 in.
Factor of safety	6 in.
Total required	20 in.

13 ● What are the essentials of a two-pipe closed vapor system?

Packless graduated valves on radiators; thermostatic return traps on return and drips; an automatic return trap to prevent water from backing out of the boiler.

14 ● Why must the automatic return trap on two-pipe vapor systems be about 18 in. above the boiler water line?

That height is necessary to overcome water line difference owing to pressure drop and friction in pipe and fittings

15 ● What is the difference between the systems illustrated in Fig. 10 and Fig. 13?

The risers and the air eliminator in Fig 13 are vented to atmosphere

16 ● What is the difference between a vacuum pump and a condensation return pump?

The vacuum pump produces and maintains a vacuum in the return lines whereas the condensation return pump returns the condensation back to the boiler The relation of the boiler to the heating units is such that the condensate will not return by gravity

17 ● What is the function of a trap?

The usual function is to allow the passage of condensate and air and to prevent the passage of steam.

18 ● Under what conditions is it advisable to use a combination float and thermostatic trap?

Where unusual capacities are required, as on large mains or blast coils

19 ● Why should the discharge from high pressure traps not go directly into a vacuum return main?

Because of its higher temperature, the high pressure condensate would immediately flash into steam in the vacuum main and cause difficulty with the vacuum pump

Chapter 32

PIPING FOR STEAM HEATING SYSTEMS

Flow of Steam in Pipes, Pipe Sizes, Tables for Pipe Sizing, One-Pipe Gravity Air Vent Systems, Two-Pipe Gravity Air Vent Systems, Two-Pipe Vapor Systems, Vacuum Systems, Atmospheric Systems, Sub-Atmospheric Systems, Orifice Systems, High Pressure Steam, Expansion in Steam and Return Lines, Piping Connections and Details, Boiler Connections, Hartford Return Connection

THE design of a steam heating system should be considered under four headings, namely, (1) the details of the heating units, (2) the arrangement of the general piping scheme, (3) the details of connections, and (4) the sizing of the lines. Items 1 and 2 are covered in Chapters 30 and 31, respectively, while this chapter considers the two latter items.

The functions of piping are to supply the heating units with steam and to remove the condensation. In some systems both the air and condensation are removed from the heating units by the return piping. To accomplish this effectively, the distribution of the steam should be efficient and equitable, without noise, and the returns should be as short as possible. When air is handled its escape should be facilitated to the utmost since an air-bound system will not heat properly. Condensation takes place in a steam system not only in the heating units, but throughout the piping system as well, and the returns also condense any steam or vapor that may be contained. At the same time part of the condensation may flash back into steam when the vacuum or pressure in the return is considerably below the steam pressure.

It is essential that steam piping systems not only distribute steam at full load but also at partial loads, as the average winter demand is less than half of the demand in most severe outside temperatures. Furthermore, in heating up rapidly the load on the steam main may exceed the maximum operating load even in extreme weather, due to the necessity of raising the temperature of the metal in the system to the steam temperature. This may require more heat than would be emitted from the system itself after it once is thoroughly heated.

STEAM FLOW

The rate of flow of dry steam or steam with a small amount of water flowing in the same direction is in accordance with the general laws of gas flow and is a function of the length and diameter of the pipe, the density of the steam, and the pressure drop through the pipe. This relationship of flow of dry steam or steam with a small amount of water has been

established by Babcock in formula 1.

$$P = 0.0000000367 \left(1 + \frac{3.6}{d}\right) \frac{W^2 L}{D d^5} \tag{1}$$

or

$$W = 5220 \sqrt{\frac{P D d^5}{\left(1 + \frac{3.6}{d}\right) L}} \tag{2}$$

where

P = loss in pressure, pounds per square inch
d = inside diameter of pipe, inches
L = length of pipe, feet.
D = weight of 1 cu ft of steam
W = weight of steam flowing per hour, pounds

Example 1 How much steam will flow per hour through 100 ft of 2-in pipe if the initial pressure is 1.3 lb per square inch and the pressure drop is 1 oz?

Solution. $P = \frac{1}{16} = 0.0625$ lb; $d = 2.067$ in. (Table 1, Chapter 34), $L = 100$ ft, $D = 0.04038$ lb (Table 8, Chapter 1). Substituting these values in Formula 2:

$$W = 5220 \sqrt{\frac{0.0625 \times 0.04038 \times 2.067^5}{\left(1 + \frac{3.6}{2.067}\right) 100}} = 97.2 \text{ lb per hour.}$$

Formula 2 does not allow for entrained water in low-pressure steam, condensation in pipe, and roughness in commercial pipe as found in practice.

The latent heat of steam (h_{fg}) at atmospheric pressure (Table 8, Chapter 1) is 970.2 Btu per pound. Inasmuch as the heat emission of an equivalent square foot of heating surface (radiation) is 240 Btu, 1 lb of steam at this pressure will supply $\frac{970.2}{240}$ or 4.04 sq ft of equivalent heating surface. This figure is usually taken as 4 even. In Example 1, the weight of steam flowing per hour would therefore supply 4 × 97.2 or 388.8 sq ft of equivalent heating surface.

PIPE SIZES

The determination of pipe sizes for steam heating depends on the following principal factors:

1. The initial pressure and the total pressure drop which may be allowed between the source of supply and the end of the return system.
2. The maximum velocity of steam allowable for quiet and dependable operation of the system.
3. The equivalent length of the run from the boiler or source of steam supply to the farthest heating unit
4. Unusual conditions in the building to be heated

Initial Pressure and Pressure Drop

Theoretically there are several factors to be considered, such as initial pressure and pressure required at the end of the line, but it is most important that (1) the total pressure drop does not exceed the initial pressure of the system; (2) the pressure drop is not so great as to cause excessive

velocities; (3) there is a constant initial pressure, except on systems specially designed for varying initial pressures, such as the sub-atmospheric which normally operate under controlled partial vacua, the orifice, and the vapor systems which at times operate under such partial vacua as may be obtained due to the condition of the fire; (4) there is sufficient difference in level, for gravity return systems, between the lowest point on the steam main, the heating units, and the dry return, when considered in relation to the boiler water line.

All systems should be designed for a low initial pressure and a reasonably small pressure drop for two reasons: *first*, the present tendency in steam heating unmistakably points toward a constant lowering of pressures even to those below atmospheric; *second*, a system designed in this manner will operate under higher pressures without difficulty. When a system designed for a relatively high initial pressure and a relatively high pressure drop is operated at a lower pressure, it is likely to be noisy and have poor circulation.

The total pressure drop should never exceed one-half of the initial pressure when condensate is flowing in the same direction as the steam Where the condensate must flow counter to the steam, the governing factor is the velocity permissible without interfering with the condensate flow. Laboratory experiments limit this to the capacities given in Tables 1 and 2 for vertical risers and in Table 3 for horizontal pipes at varying grades.

Maximum Velocity and Reaming

The capacity of a steam pipe in any part of a steam system depends upon the quantity of condensation present, the direction in which the condensate is flowing, and the pressure drop in the pipe. Where the quantity of condensate is limited and is flowing in the same direction as the steam, only the pressure drop need be considered. When the condensate must flow against the steam, even in limited quantity, the velocity of the steam must not exceed limits above which the disturbance between the steam and the counter-flowing water may produce objectionable sounds, such as water hammer, or may result in the retention of water in certain parts of the system until the steam flow is reduced sufficiently to permit the water to pass. The velocity at which such disturbances take place is a function of (1) the pipe size, whether the pipe runs horizontally or vertically, (2) the pitch of the pipe if it runs horizontally, and (3) the quantity of condensate flowing against the steam.

Two factors of uncertainty always exist in determining the capacity of any steam pipe. The first is variation in manufacture, which apparently cannot be avoided and which caused an actual difference of 20 per cent in the capacity of a 1-in. pipe in experiments carried on at the A.S.H.V.E. Research Laboratory (Table 4). The second is the reaming of the ends of the pipe after cutting, which, experiments indicate, might reduce the capacity of a 1-in. pipe as much as 28.7 per cent (Table 5). All of the capacity tables given in this chapter include a factor of safety. However, the pipe on which Table 4 is based showed no particular defects or constrictions on the inside, and the factor of safety referred to does not cover abnormal defects or constrictions *nor does it cover pipe not properly reamed.*

TABLE 1. MAXIMUM ALLOWABLE CAPACITIES OF UP-FEED RISERS FOR ONE-PIPE LOW PRESSURE STEAM

Based on A S H V. E. Research Laboratory Tests

PIPE SIZE INCHES	VELOCITY FEET PER SECOND	PRESSURE DROP OUNCES PER 100 FT	CAPACITY		
			Sq Ft Radiation	Btu per Hour	Lb Steam per Hour
A	*B*	*C*	*D*	*E*	*F*
1	14.1	0 68	45	10,961	11 3
1¼	17 6	0 66	98	23,765	24 5
1½	20 0	0 66	152	36,860	38 0
2	23.0	0 57	288	69,840	72 0
2½	26.0	0 54	464	112,520	116 0
3	29.0	0 48	799	193,600	199 8
3½	31.0	0 44	1144	277,000	286 0
4	32.0	0 39	1520	368,000	380 0

INSTRUCTIONS FOR USING TABLE 1

1 Capacities given in Table 1 should never be exceeded on one-pipe risers

2 Capacities are based on ¼-lb condensation per square foot equivalent radiation and actual diameter of standard pipe

3 All pipe should be well reamed and free from constrictions. Fittings should be up to size (See Tables 4 and 5)

TABLE 2 MAXIMUM ALLOWABLE CAPACITIES OF UP-FEED RISERS FOR TWO-PIPE LOW PRESSURE STEAM

Based on A. S. H. V. E. Research Laboratory Tests

PIPE SIZE INCHES	VELOCITY FEET PER SECOND	PRESSURE DROP OUNCES PER 100 FT	CAPACITY		
			Sq Ft Radiation	Btu per Hour	Lb Steam per Hour
A	*B*	*C*	*D*	*E*	*F*
¾	20	—	40	9550	10.0
1	23	1.78	74	17,900	18.45
1¼	27	1.57	151	36,500	37.65
1½	30	1.48	228	55,200	57 0
2	35	1.33	438	106,100	109 5
2½	38	1.16	678	164,100	169.4
3	41	0.95	1129	273,500	282.2
3½	42	0.81	1548	375,500	387 0
4	43	0.71	2042	495,000	510.5

INSTRUCTIONS FOR USING TABLE 2

1 The capacities given in this table should never be exceeded on two-pipe risers

2. Capacities are based on ¼-lb condensation per square foot equivalent radiation and actual diameter of standard pipe

3 All pipe should be well reamed and free from constrictions Fittings should be up to size. (See Tables 4 and 5)

Table 3. Comparative Capacity of Steam Lines at Various Pitches for Steam and Condensate Flowing in Opposite Directions[a]

Pitch of Pipe in Inches per 10 Ft

Pitch of Pipe	¼ in		½ in		1 in		1½ in		2 in		3 in		4 in		5 in.	
Pipe Size Inches	Sq Ft Rad Based on 240 Btu	Max Vel	Sq Ft Rad Based on 240 Btu	Max Vel	Sq Ft Rad Based on 240 Btu	Max Vel	Sq Ft Rad Based on 240 Btu	Max Vel	Sq Ft Rad Based on 240 Btu	Max Vel	Sq Ft Rad Based on 240 Btu	Max Vel	Sq Ft Rad Based on 240 Btu	Max Vel	Sq Ft Rad. Based on 240 Btu	Max Vel
¾	25.0	12	30.3	14	37.3	18	40.4	19	42.5	20	46.1	21	47.5	22	49.3	23
1	45.8	12	52.6	15	63.0	17	70.0	20	75.2	22	83.0	23	87.9	25	90.2	26
1¼	104.9	18	117.2	20	133.0	23	144.5	25	154.0	27	165.0	28	172.6	29	178.2	31
1½	142.6	18	159.0	21	181.0	23	196.5	25	209.3	27	224.0	28	234.8	30	242.6	31
2	236.0	19	263.5	20	299.5	23	325.5	25	346.5	27	371.5	28	388.4	29	401.1	30

[a]Data from American Society of Heating and Ventilating Engineers Research Laboratory

Equivalent Length of Run

All tables for the flow of steam in pipes, based on pressure drop, must allow for the friction offered by the pipe as well as for the additional resistance of the fittings and valves. These resistances generally are stated in terms of straight pipe; in other words, a certain fitting will produce a drop in pressure equivalent to so many feet of straight run of the same size of pipe. Table 6 gives the number of feet of straight pipe usually allowed for the more common types of fittings and valves. In all pipe sizing tables in this chapter the *length of run* refers to the *equivalent length of run* as distinguished from the *actual length* of pipe in feet. The length of run is not usually known at the outset; hence it is necessary to assume some pipe size at the start. Such an assumption frequently is considerably in error and a more common and practical method is to assume the length of run and to check this assumption after the pipes are sized. For this purpose the length of run usually is taken as double the actual length of pipe.

Table 4 Per Cent Difference in Capacity for Carrying Steam and Condensate Due to Variation of Pipe Size and Smoothness[a]

	Maximum Condensation, Lb per Hour			
Size of pipe	¾ In	1 In.	1¼ In.	1½ In
Minimum	14.00	24.89	45.42	70.50
Maximum	15.20	30.08	52.08	82.00
Per cent variation	8.6	20.8	14.7	16.3

[a]Data from American Society of Heating and Ventilating Engineers Research Laboratory.

Table 5 Effect of Reaming Entrance to One-Inch One-Pipe Risers[a]

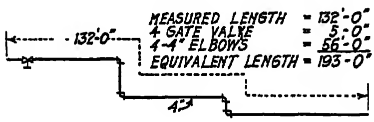
	Maximum Capacity of Riser	Per Cent Decrease
Reamed entrances	24.7 lb per hour	0.0
Rounded entrances	23.9 lb per hour	3.2
Squared entrances	22.2 lb per hour	10.1
Three wheel cutter	19.2 lb per hour	22.2
Single wheel cutter	17.6 lb per hour	28.7

[a]Data from American Society of Heating and Ventilating Engineers Research Laboratory.

TABLE 6. LENGTH IN FEET OF PIPE TO BE ADDED TO ACTUAL LENGTH OF RUN—OWING TO FITTINGS—TO OBTAIN EQUIVALENT LENGTH

SIZE OF PIPE INCHES	ST'D ELBOW	SIDE OUTLET TEE	GATE VALVE	GLOBE VALVE	ANGLE VALVE
	Length in Feet to be Added to Run				
2	5	16	2	18	9
2½	7	20	3	25	12
3	10	26	3	33	16
3½	12	31	4	39	19
4	14	35	5	45	22
5	18	44	7	57	28
6	22	50	9	70	32
7	26	55	10	82	37
8	31	63	12	94	42
9	35	69	13	105	47
10	39	76	15	118	52
12	47	90	18	140	63
14	53	105	20	160	72

Example of length in feet of pipe to be added to actual length of run.



TABLES FOR PIPE SIZING[1]

Factors determining the size of a steam pipe and its allowable limit of capacity are as follows:

1. Pipe condensate flowing with steam.
2. Pipe condensate flowing against steam.
3. Pipe and radiator condensate flowing with steam.
4. Pipe and radiator condensate flowing against steam

It is apparent that (3) and (4) are practically limited to one-pipe systems while (1) and (2) cover all other systems.

Tables 7 and 8, worked out for determining pipe sizes, have their columns lettered continuously, Columns *A* through *L* being in Table 7, and *M* through *EE* in Table 8. In the following text, reference made to columns will be by letter. The tables are based on the actual inside diameters of the pipe and the condensation of ¼ lb (4 oz) of steam per square foot of equivalent direct radiation[2] (*abbreviated EDR*) per hour. The drops indicated are drops in pressure per 100 ft of equivalent length of run. The pipe is assumed to be well reamed without unusual or noticeable defects.

[1]Pipe size tables in this chapter have been compiled in simplified and condensed form for the convenience of the user; at the same time all of the information contained in previous editions of THE GUIDE has been retained. Values of pressure drops, formerly expressed in ounces, are now expressed in fractions of a pound.

[2]As steam system design has materially changed in recent years so that 240 Btu no longer expresses the heat of condensation from a square foot of radiator surface per hour, and as present day heating units have different characteristics from older forms of radiation, it is the purpose of THE GUIDE to gradually eliminate the empirical expression *square foot of equivalent direct radiation EDR*, and to substitute a logical unit based on the Btu. The new terms to express the equivalent of 1000 Btu (Mb), and 1000 Btu per hour (Mbh), have been approved by the A S H V E.

Table 7 may be used for sizing piping for steam heating systems by determining the allowable or desired pressure drop per 100 equivalent feet of run and reading from the column for that particular pressure drop. This applies to all steam mains on both one-pipe and two-pipe systems, vapor systems, and vacuum systems. Columns *B* to *G*, inclusive, are used where the steam and condensation flow in the same direction, while Columns *H* and *I* are for cases where the steam and condensation flow in opposite directions, as in risers and runouts that are not dripped. Columns *J*, *K*, and *L* are for one-pipe systems and cover riser, radiator valve, and vertical connection sizes, and radiator and runout sizes, all of which are based on the critical velocities of the steam to permit the counter flow of condensation without noise.

Sizing of return piping may be done with the aid of Table 8 where pipe capacities for wet, dry, and vacuum return lines are shown for the pressure drops per 100 ft corresponding to the drops in Table 7. *It is customary to use the same pressure drop on both the steam and return sides of a system.*

TABLE 7. STEAM PIPE CAPACITIES

Capacity Expressed in Square Feet of Equivalent Direct Radiation

(Reference to this table will be by column letter *A* through *L*)

This table is based on pipe size data developed through the research investigations of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

	CAPACITIES OF STEAM MAINS AND RISERS								SPECIAL CAPACITIES FOR ONE-PIPE SYSTEMS ONLY		
	DIRECTION OF CONDENSATION FLOW IN PIPE LINE										
	With the Steam in One-Pipe and Two-Pipe Systems						Against the Steam Two-Pipe Only		Supply Risers Up-Feed	Radiator Valves and Vertical Connections	Radiator and Riser Run-outs
PIPE SIZE IN.	1/32 lb or ½ Oz Drop	1/24 lb or ⅔ Oz Drop	1/16 lb or 1 Oz Drop	⅛ lb or 2 Oz Drop	¼ lb or 4 Oz Drop	½ lb or 8 Oz Drop	Vertical	Horizontal			
A	*B*	*C*	*D*	*E*	*F*	*G*	*H*[a]	*I*[c]	*J*[b]	*K*	*L*[c]
¾	———	———	30	———	———	———	30	———	25	———	———
1	39	46	56	79	111	157	56	26	45	20	20
1¼	87	100	122	173	245	346	122	58	98	55	55
1½	134	155	190	269	380	538	190	95	152	81	81
2	273	315	386	546	771	1,091	386	195	288	165	165
2½	449	518	635	898	1,270	1,797	635	395	464	———	260
3	822	948	1,163	1,645	2,326	3,289	1,129	700	799	———	475
3½	1,228	1,419	1,737	2,457	3,474	4,913	1,548	1,150	1,144	———	745
4	1,738	2,011	2,457	3,475	4,914	6,950	2,042	1,700	1,520	———	1,110
5	3,214	3,712	4,546	6,429	9,092	12,858	———	3,150	———	———	2,180
6	5,276	6,094	7,462	10,553	14,924	21,105	———	———	———	———	———
8	10,983	12,682	15,533	21,967	31,066	43,934	———	———	———	———	———
10	20,043	23,144	28,345	40,085	56,689	80,171	———	———	———	———	———
12	32,168	37,145	45,492	64,336	90,985	128,672	———	———	———	———	———
16	60,506	69,671	84,849	121,012	169,698	242,024	———	———	———	———	———
	All Horizontal Mains and Down-Feed Risers						Up-Feed Risers	Mains and Undripped Run-outs	Up-Feed Risers	Radiator Connections	Run-outs Not Dripped

Note—All drops shown are in pounds per 100 ft of equivalent run—based on pipe properly reamed

[a]Do not use Column *H* for drops of 1/24 or 1/32 lb; substitute Column *C* or Column *B* as required.

[b]Do not use Column *J* for drop of 1/32 lb except on sizes 3 in. and over; below 3 in substitute Column *B*.

[c]On radiator runouts over 8 ft long increase one pipe size over that shown in Table 7

TABLE 8. RETURN PIPE CAPACITIES

Capacity Expressed in Square Feet of Equivalent Direct Radiation

(Reference to this table will be by column letter *M* through *EE*)

This table is based on pipe size data developed through the research investigations of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

CAPACITY OF RETURN MAINS AND RISERS

Pipe Size Inches	1/32 Lb or ½ Oz Drop per 100 Ft			1/24 Lb or ⅔ Oz Drop per 100 Ft			1/16 Lb or 1 Oz Drop per 100 Ft			⅛ Lb or 2 Oz Drop per 100 Ft			¼ Lb or 4 Oz Drop per 100 Ft			½ Lb or 8 Oz Drop per 100 Ft		
	Wet	Dry	Vac	Wet	Dry	Vac	Wet	Dry	Vac	Wet	Dry	Vac	Wet	Dry	Vac	Wet	Dry	Vac
M	*N*	*O*	*P*	*Q*	*R*	*S*	*T*	*U*	*V*	*W*	*X*	*Y*	*Z*	*AA*	*BB*	*CC*	*DD*	*EE*
Mains																		
¾	......	......	......	......	......	326	......	......	400	......	......	568	......	......	800	......	......	1,130
1	500	248	......	580	285	570	700	320	700	1,000	412	994	1,400	460	1,400	......	......	1,977
1¼	850	520	......	990	595	976	1,200	670	1,200	1,700	868	1,704	2,400	962	2,400	......	......	3,390
1½	1,350	822	......	1,570	943	1,547	1,900	1,058	1,900	2,700	1,362	2,696	3,800	1,512	3,800	......	......	5,370
2	2,800	1,880	......	3,240	2,140	3,256	4,000	2,300	4,000	5,600	2,960	5,680	8,000	3,300	8,000	......	......	11,300
2½	4,700	3,040	......	5,300	3,470	5,453	6,700	3,800	6,700	9,400	4,900	9,510	13,400	5,450	13,400	......	......	18,925
3	7,500	5,840	......	8,500	6,250	8,710	10,700	7,000	10,700	15,000	9,000	15,190	21,400	10,000	21,400	......	......	30,230
3½	11,000	7,880	......	13,200	8,800	13,020	16,000	10,000	16,000	22,000	12,900	22,710	32,000	14,300	32,000	......	......	45,200
4	15,500	11,700	......	18,300	13,400	17,910	22,000	15,000	22,000	31,000	19,300	31,220	44,000	21,500	44,000	......	......	62,180
5	......	......	......	......	......	31,500	......	......	38,700	......	......	54,920	......	......	77,400	......	......	109,300
6	......	......	......	......	......	50,450	......	......	62,000	......	......	88,000	......	......	124,000	......	......	175,100
Risers																		
¾	......	190	......	......	190	570	......	190	700	......	190	994	......	190	1,400	......	......	1,977
1	......	450	......	......	450	976	......	450	1,200	......	450	1,704	......	450	2,400	......	......	3,390
1¼	......	990	......	......	990	1,547	......	990	1,900	......	990	2,696	......	990	3,800	......	......	5,370
1½	......	1,500	......	......	1,500	3,256	......	1,500	4,000	......	1,500	5,680	......	1,500	8,000	......	......	11,300
2	......	3,000	......	......	3,000	5,453	......	3,000	6,700	......	3,000	9,510	......	3,000	13,400	......	......	18,925
2½	......	......	......	......	......	8,710	......	......	10,700	......	......	15,910	......	......	21,400	......	......	30,230
3	......	......	......	......	......	13,020	......	......	16,000	......	......	22,710	......	......	32,000	......	......	45,200
3½	......	......	......	......	......	17,910	......	......	22,000	......	......	31,220	......	......	44,000	......	......	62,180
4	......	......	......	......	......	31,500	......	......	38,700	......	......	54,920	......	......	77,400	......	......	109,300
5	......	......	......	......	......	50,450	......	......	62,000	......	......	88,000	......	......	124,000	......	......	175,100

Example 2. What pressure drop should be used for the steam piping of a system if the measured length of the longest run is 500 ft and the initial pressure is not to be over 2-lb gage?

Solution It will be assumed, if the measured length of the longest run is 500 ft, that when the allowance for fittings is added the equivalent length of run will not exceed 1,000 ft Then, with the pressure drop not over one half of the initial pressure, the drop could be 1 lb or less With a pressure drop of 1 lb and a length of run of 1,000 ft, the drop per 100 ft would be $\frac{1}{10}$ lb, while if the total drop were $\frac{1}{2}$ lb, the drop per 100 ft would be $\frac{1}{20}$ lb In the first instance the pipe could be sized according to Column *D* for $\frac{1}{16}$ lb per 100 ft, and in the second case, the pipe could be sized according to Column *C* for $\frac{1}{24}$ lb. On completion of the sizing, the drop could be checked by taking the longest line and actually calculating the equivalent length of run from the pipe sizes determined If the calculated drop is less than that assumed, the pipe size is all right: if it is more, it is probable that there are an unusual number of fittings involved, and either the lines must be straightened or the column for the next lower drop must be used and the lines resized Ordinarily resizing will be unnecessary

ONE-PIPE GRAVITY AIR-VENT SYSTEMS

One-pipe gravity air-vent systems in which the equivalent length of run does not exceed 200 ft should be sized as follows:

1 *For the steam main and dripped runouts to risers* where the steam and condensate flow in the same direction, use $\frac{1}{16}$-lb drop (Column *D*).

2. *Where the riser runouts are not dripped* and the steam and condensation flow in opposite directions, and *also in the radiator runouts* where the same condition occurs, use Column *L*.

3 *For up-feed steam risers* carrying condensation back from the radiators, use Column *J*.

4. *For down-feed systems the main risers* of which do not carry any radiator condensation, use Column *H*.

5. *For the radiator valve size and the stub connection*, use Column *K*.

6. For the *dry return main*, use Column *U*.

7. For the *wet return main* use Column *T*.

On systems exceeding an equivalent length of 200 ft, it is suggested that the total drop be not over $\frac{1}{4}$ lb. The return piping sizes should correspond with the drop used on the steam side of the system. Thus, where $\frac{1}{24}$-lb drop is being used, the steam main and dripped runouts would be sized from Column *C;* radiator runouts and undripped riser runouts from Column *L;* up-feed risers from Column *J*; the main riser on a down-feed system from Column *C* (it will be noted that if Column *H* is used the drop would exceed the limit of $\frac{1}{24}$ lb); the dry return from Column *R;* and the wet return from Column *Q*.

With a $\frac{1}{32}$-lb drop the sizing would be the same as for $\frac{1}{24}$ lb except that the steam main and dripped runouts would be sized from Column *B*, the main riser on a down-feed system from Column *B*, the dry return from Column *O*, and the wet return from Column *N*.

Example 3. Size the one-pipe gravity steam system shown in Fig. 1 assuming that this is all there is to the system or that the riser and run shown involve the longest run on the system.

Solution. The total length of run actually shown is 215 ft. If the equivalent length of run is taken at double this, it will amount to 430 ft, and with a total drop of $\frac{1}{4}$ lb the drop per 100 ft will be slightly less than $\frac{1}{16}$ lb. It would be well in this case to use $\frac{1}{24}$ lb, and this would result in the theoretical sizes indicated in Table 9 These theo-

retical sizes, however, should be modified by not using a wet return less than 2 in while the main supply, *g-h*, if from the uptake of a boiler, should be made the full size of the main, or 3 in. Also the portion of the main *k-m* should be made 2 in if the wet return is made 2 in.

Notes on Gravity One-Pipe Air-Vent Systems

1. Pitch of mains should be not less than ¼ in. in 10 ft.

2 Pitch of horizontal runouts to risers and radiators should not be less than ½ in. in 10 ft Where this pitch cannot be obtained runouts over 8 ft in length should be one size larger than called for in the table.

3 In general, it is not desirable to have a main less than 2 in The diameter of the far end of the supply main should be not less than half its diameter at its largest part

4. Supply mains, branches to risers, or risers, should be dripped where necessary.

Table 9. Pipe Sizes for One-Pipe Up-feed System Shown in Fig 1

Part of System	Section of Pipe	Radiation Supplied (Sq Ft)	Theoretical Pipe size (Inches)	Practical Pipe size (Inches)
Branches to radiators	....	100	2	2
Branches to radiators.	....	50	1¼	1¼
Riser	*a* to *b*	200	2	2
Riser	*b* to *c*	300	2½	2½
Riser	*c* to *d*	400	2½	2½
Riser	*d* to *e*	500	3	3
Riser	*e* to *f*	600	3	3
Branch to riser	*f* to *g*	600	3½	3½
Supply main	*g* to *h*	600	3	3
Branch to supply main	*h* to *j*	600	2½	3
Dry return main	*f* to *k*	600	1¼	2
Wet return main	*k* to *m*	600	1	2
Wet return main	*m* to *n*	600	1	2
Wet return main	*n* to *p*	600	1	2

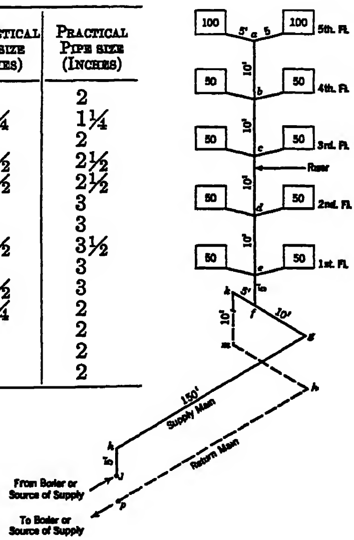


Fig. 1. Riser, Supply Main and Return Main of One-Pipe System

TWO-PIPE GRAVITY AIR-VENT SYSTEMS

The method employed in determining pipe sizes for two-pipe gravity air-vent systems is similar to that described for one-pipe systems except that the steam mains never carry radiator condensation. The drop allowable per 100 ft of equivalent run is obtained by taking the equivalent length to the farthest radiator as double the actual distance, and then dividing the allowable or desired total drop by the number of hundreds of feet in the equivalent length. Thus in a system measuring 400 ft from the boiler to the farthest radiator, the approximate equivalent length of run would be 800 ft. With a total drop of ½ lb the drop per 100 ft would be $\frac{\frac{1}{2}}{8}$ or $\frac{1}{16}$ lb; therefore, Column *D* would be used for all steam mains where the condensation and steam flow in the same direction. If a total drop of ¼ lb is desired, the drop per 100 ft would be $\frac{1}{32}$ lb

and Column *B* would be used. If the total drop were to be 1 lb, the drop per 100 ft would be ⅛ lb and Column *E* would be used.

For mains and riser runouts that are not dripped, and for radiator runouts where in all three cases the condensation and steam flow in opposite directions, Column *I* should be used, while for the steam risers Column *H* should be used unless the drop per 100 ft is 1/24 lb or 1/32 lb, when Columns *B* or *C* should be substituted so as not to exceed the drop permitted.

On an overhead down-feed system the main steam riser should be sized by reference to Column *H*, but the down-feed steam risers supplying the radiators should be sized by the appropriate Columns *B* through *G*, since the condensation flows downward with the steam through them. The riser runouts, if pitched down toward the riser as they should be, are sized the same as the steam mains, and the radiator runouts are made the same as in an up-feed system.

In either up-feed or down-feed systems the returns are sized in the same manner and on the same pressure drop basis as the steam main; the return mains are taken from Columns *O*, *R*, *U*, *X*, or *AA* according to the drop used for the steam main; and the risers are sized by reading the lower part of Table 8 under the column used for the mains. The horizontal runouts from the riser to the radiator are not usually increased on the return lines although there is nothing incorrect in this practice. The same notes apply that are given for one-pipe gravity systems.

TWO-PIPE VAPOR SYSTEMS

While many manufacturers of patented vapor heating accessories have their own schedules for pipe sizing, an inspection of these sizing tables indicates that in general as small a drop as possible is recommended. The reasons for this are: (1) to have the condensation return to the boiler by gravity, (2) to obtain a more uniform distribution of steam throughout the system, (3) because with large variation in pressure the value of graduated valves on radiators is destroyed.

For small vapor systems where the equivalent length of run does not exceed 200 ft, it is recommended that the main and any runouts to risers that may be dripped should be sized from Column *D*, while riser runouts not dripped and radiator runouts should employ Column *I*. The up-feed steam risers should be taken from Column *H*. On the returns, the risers should be sized from Column *U* (lower portion) and the mains from Column *U* (upper portion). It should again be noted that the pressure drop in the steam side of the system is kept the same as on the return side except where the flow in the riser is concerned.

On a down-feed system the main vertical riser should be sized from Column *H*, but the down-feed risers can be taken from Column *D* although it so happens that the values in Columns *D* and *H* correspond. This will not hold true in larger systems.

For vapor systems over 200 ft of equivalent length, the drop should not exceed ⅛ lb to ¼ lb, if possible. Thus, for a 400 ft equivalent run the drop per 100 ft should be not over ⅛ lb divided by 4, or 1/32 lb. In this case the steam mains would be sized from Column *B*; the radiator and

undripped riser runouts from Column *I*; the risers from Column *B*, because Column *H* gives a drop in excess of 1/32 lb. On a down-feed system, Column *B* would have to be used for both the main riser and the smaller risers feeding the radiators in order not to increase the drop over 1/32 lb. The return risers would be sized from the lower portion of Column *O* and the dry return main from the upper portion of the same column, while any wet returns would be sized from Column *N*. The same pressure drop is applied on both the steam and the return sides of the system.

Notes on Vapor Systems

1 Pitch of mains should be not less than 1/4 in. in 10 ft

2 Pitch of horizontal runouts to risers and radiators should be not less than 1/2 in. in 10 ft Where this pitch cannot be obtained runouts over 8 ft in length should be one size larger than called for in the table.

3 In general it is not desirable to have a supply main smaller than 2 in, and when the supply main is 3 in. or over at the boiler or pressure reducing valve it should be not less than 2 1/2 in. at the far end

4. When necessary, supply main, supply risers, or branches to supply risers should be dripped separately into a wet return The drip for a vapor system may be connected into the dry return through a thermostatic drip trap.

VACUUM SYSTEMS

Vacuum systems are usually employed in large installations and have total drops varying from 1/4 to 1/2 lb. Systems where the maximum equivalent length does not exceed 200 ft preferably employ the smaller pressure drop while systems over 200 ft equivalent length of run more frequently go to the higher drop, owing to the relatively greater saving in pipe sizes. For example, a system with 1200 ft longest equivalent length of run would employ a drop per 100 ft of 1/2 lb divided by 12, or 1/24 lb. In this case the steam main would be sized from Column *C*, and the risers also from Column *C* (Column *H* could be used as far as critical velocity is concerned but the drop would exceed the limit of 1/24 lb). Riser runouts, if dripped, would use Column *C* but if undripped would use Column *I*; radiator runouts, Column *I*; return risers, lower part of Column *S*; return runouts to radiators, one pipe size larger than the radiator trap connections.

Notes on Vacuum Systems

1. It is not generally considered good practice to exceed 1/8-lb drop per 100 ft of equivalent run nor to exceed 1 lb total pressure drop in any system.

2. Pitch of mains should be not less than 1/4 in. in 10 ft.

3. Pitch of horizontal runouts to risers and radiators should be not less than 1/2 in. in 10 ft. Where this pitch cannot be obtained runouts over 8 ft in length should be one size larger than called for in the table.

4. In general it is not considered desirable to have a supply main smaller than 2 in. When the supply main is 3 in or over, at the boiler or pressure reducing valve, it should be not less than 2 1/2 in at the far end.

5. When necessary, the supply main, supply riser, or branch to a supply riser should be dripped separately through a trap into the vacuum return. A connection should not be made between the steam and return sides of a vacuum system without interposing a trap to prevent the steam from entering the return line.

6. Lifts should be avoided if possible, but when they cannot be eliminated they should be made in the manner described in Chapter 31 under *Up-Feed Vacuum Systems*.

ATMOSPHERIC SYSTEMS

The sizing of the supply and return piping on atmospheric systems is practically identical with the sizing used for vacuum systems and the same notes apply, except that no lift can be made in the return line.

SUB-ATMOSPHERIC SYSTEMS

Any properly pitched, correctly sized vacuum system without a lift may be used as a sub-atmospheric system when the proper equipment is substituted for the ordinary vacuum pump, traps, and controls. On new systems manufacturers usually recommend a drop on the steam line of between ¼ and ⅜ lb for the total run, and suggest adding 25 ft to the total equivalent length of run to insure that the steam gets through to the last radiator.

The same notes apply to these systems as for vacuum systems, except that no lifts can be made in the returns.

ORIFICE SYSTEMS

The orifice systems can be operated with any piping system suitable for vacuum operation, according to experienced designers. Because these systems vary considerably in detail, it is advisable to consult the manufacturer of the particular system contemplated for recommendations.

The same notes apply to these systems as to vacuum systems, except that lifts cannot be made in the returns of orifice systems if a vacuum pump is used.

HIGH PRESSURE STEAM

When steam heating systems are supplied with steam from a high pressure plant, one or more pressure-reducing valves are used to bring the pressure down to that required by the heating system. It has been considered good practice to make the pressure reductions in steps not to exceed 50 lb in each case. For example, in reducing from 100-lb gage to 2-lb gage, two pressure reducing valves would be used, the first reducing the pressure from 100-lb gage to 50 lb and the second reducing the pressure from 50-lb gage to 2-lb gage. Valves are available that will reduce 100 lb in one step, and it is questionable whether two valves are now required for initial pressures of 150 lb or less.

The pressure-reducing valve, or pressure-regulator as it is sometimes termed, has ratings which vary 200 to 400 per cent. Some of these ratings are based on arbitrary steam velocities through the valve of 5,000 to 10,000 fpm and it is assumed that the valve when wide open has the same area as the pipe on the inlet opening of the valve. At times it is considered desirable to keep the steam velocity in the high pressure section of the piping and the low pressure section constant. The velocity through the valve port is obviously a function of the pressure drop across the valve. It is well known that steam flowing through an orifice increases its velocity until the pressure on the outlet side is reduced to 58 per cent of the absolute pressure on the inlet side, and that with further reduction of pressure on the outlet side little change in velocity will be obtained. As practically all pressure-reducing valves used for steam heating work

lower the steam pressure to less than 58 per cent of the inlet pressures, only the maximum velocity through such valves need be considered. If it is assumed that the valve, when fully open, has an area equal to that of the inlet pipe size, that the steam is flowing into a pressure less than 58 per cent of the initial pressure, that the orifice efficiency is approximately 70 per cent, and that 20 per cent more is allowed for a factor of safety, then the pressure reducing valves will have the working capacities shown in Table 10. If the valve, when fully open, does not give an orifice area equal to that of the pipe on the inlet side, then the capacities will be proportional to the percentage of opening secured, taking the pipe area as 100 per cent. More frequently, difficulty is encountered from the use of pressure reducing valves which are too large in size instead of being

Table 10. Capacities of Pressure-Reducing Valves

(100-lb Gage Down to any Pressure—52 lb or Less)

Inlet Nominal Pipe Diameter (Inches)	Pounds Steam per Hour at 100-Lb Gage	Equivalent Direct Radiation Sq Ft at ¼ Lb	Equivalent Direct Radiation Sq Ft at ⅛ Lb
½	866	3,464	2,598
¾	1,576	6,304	4,728
1	2,459	9,836	7,377
1¼	4,263	17,052	12,689
1½	5,808	23,232	17,424
2	9,564	38,256	28,692
2½	13,623	54,492	40,869
3	21,041	84,104	63,123
3½	28,213	112,852	84,039
4	36,285	145,140	108,855
5	56,971	227,884	170,913
6	82,336	329,344	247,008

Formula:

$$\frac{A \times V \times 3600 \times 50}{144 \times 3.88} = \text{pounds per hour passed by orifice}$$

where

A = area of inlet pipe, square inches
V = velocity of steam through orifice (approximately 870 fps)
50 = 70 per cent efficiency of orifice less 20 per cent for factor of safety
144 = square inches in 1 sq ft
3600 = seconds in one hour.
3 88 = cubic feet per pound at 100-lb gage

too small. Where valves are large in size, the valve tends to work close to the seat, causing it to cut out in a relatively short time, as well as being noisy in operation.

Most exact regulation of pressure on steam heating systems is secured from diaphragm-operated valves controlled by a pilot line from the low pressure pipe, taken off the low pressure main at least 15 ft from the reducing valve. The reducing valves operating on the proportional-reduction principle will give a variation of steam pressure on the low pressure side if the initial pressure varies between considerable limits. The so-called dead-end valve is used for reduced pressures where the line has not sufficient condensing capacity at all times to condense the leakage that might occur with the ordinary valve. Single-disc valves do not give as close regulation as double-disc valves, but the single disc is preferable where dead-end valves are necessary, such as on short runs to thermo-

statically controlled hot water heaters, central fan heating units and unit heaters.

The correct installation (Fig. 2) of a pressure-reducing valve includes a pressure-reducing valve with a gate valve on each side, a by-pass controlled by a globe valve, a pressure gage on the low pressure side, and a safety valve on the low pressure main at some point, usually within a reasonable distance of the pressure-reducing valve. Pressure-reducing valves should have expanded outlets for sizes greater than 2 in. Where the steam main is of still larger diameter than the expanded outlet, and in cases where straight valves are used, an increaser is placed close against the outlet of the valve to reduce the velocity immediately after passing through the valve. Strainers are recommended on the inlets of all pressure-reducing valves. A pressure gage may be located on the high-pressure line near the valve if desired.

Owing to the large variation in steam demand on the average heating system, it is generally advisable to use two pressure-reducing valves con-

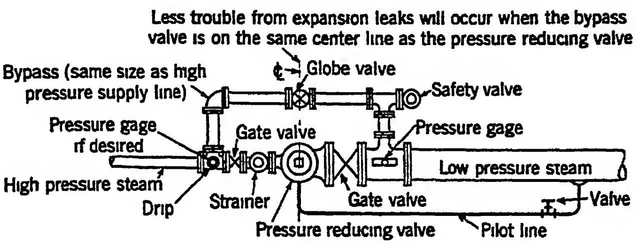


FIG 2. TYPICAL PRESSURE-REDUCING VALVE INSTALLATION

nected in parallel. One valve should be large enough for the maximum load and the other should have a diameter approximately half that of the first. The smaller valve can be used most of the time, for it will give much better regulation than the larger one on light or normal loads.

Control Valves

Gate valves are recommended in all cases where service demands that the valve be either entirely open or entirely closed, but they should never be used for throttling. Angle globe valves and straight globe valves should be used for throttling, as done on by-passes around pressure reducing valves or on by-passes around traps.

EXPANSION IN STEAM AND RETURN LINES

Because all steam and return lines expand and contract with changes in temperature, provision should be made for such movement. The expansion in steam supply pipes is normally taken at 1¼ to 1½ in. per 100 ft and in return lines at one-half or two-thirds of this amount. It may be calculated accurately if the temperature rise and fall can be determined with reasonable certainty (Page 631, Chapter 34). The temperature at the time of erection often has a greater expansion effect on piping than the temperature in the building after it has been put into service.

Expansion may be taken care of by any, or all, of three different methods, namely, (1) the spring in the pipe including offsets and expansion bends, (2) the turning of the pipe on its threads and swing joints, and (3) the use of expansion joints.

By the first scheme, which is the most popular method where space permits, the pipe is offset, or *broken*, around rooms or corners, and is hung so that the spring in the pipe at right angles to the expansion movement is sufficient to absorb the expansion. If conditions do not lend themselves to this treatment, regular expansion bends of the *U* or offset type may be used. In tight places such as pipe tunnels the expansion joint is preferable. See additional material on pipe expansion bends in Chapter 34.

On riser runouts and radiator runouts the swing joint is used almost without exception. On high vertical risers the pipes may be reversed every five to ten stories; that is, the supply is carried over to the adjacent return riser location and the return riser is run over to the former supply riser location, thus making horizontal offsets in each line. Corrugated copper expansion joints also are used on risers but must be made accessible in case future replacement becomes necessary.

PIPING CONNECTIONS AND DETAILS

Piping connections may be classified into two groups: *first*, those suitable for any system of steam heating; *second*, those devised for certain systems which cannot be satisfactorily applied to any other type. There are also various details that apply to piping on the steam side which cannot be used on the returns. An installation that is designed and sized correctly and installed with care may be rendered defective by the use of improper connections, such as runouts that do not allow for expansion, thermostatic traps unprotected from scale, pressure-reducing valves without strainers, and lack of drips at required points.

BOILER CONNECTIONS

Supply

Boiler headers and connections have the largest sizes of pipe used in a system. Cast-iron, horizontal-type, low pressure heating boilers usually have several tapped outlets in the top, the manufacturers recommending their use in order to reduce the velocity of the steam in the vertical uptakes from the boiler and to permit entrained water to return to the boiler instead of being carried over into the steam main where it must be cared for by dripping. Steel heating boilers usually are equipped with only one steam outlet but many engineers believe that better results are obtained by specifying that such boilers have two. The second outlet, usually located 3 or 4 ft back of the regular one, reduces the velocity 50 per cent in the steam uptake.

Fig. 3 shows a type of boiler connection that was used for many years and one with which some boilers are now piped. The uptakes are carried as high as possible, turned horizontally and run out to the side of the boiler and then are connected together into the main boiler runout which drops into the top of the boiler header through a boiler stop valve. No drips are provided on this type of runout except a very small one which is sometimes installed on the boiler side of the stop valve. Fig. 4 shows a

type of boiler connection which is regarded as superior to that shown in Fig. 3 and which is the type illustrated in the system diagrams in Chapter 31. This type is similar to that shown in Fig. 3 except that the horizontal branches from the uptakes are connected into the main boiler runout, and the steam is carried toward the rear of the boiler. The branch to the building or boiler header is taken off *behind* the last horizontal boiler connection. At the rear end of this main runout, a large size drip, or balance pipe, is dropped down into the boiler return, or into the top of the Hartford Loop, which is described in a following paragraph. As a result, any water carried over from the boiler follows the direction of steam flow

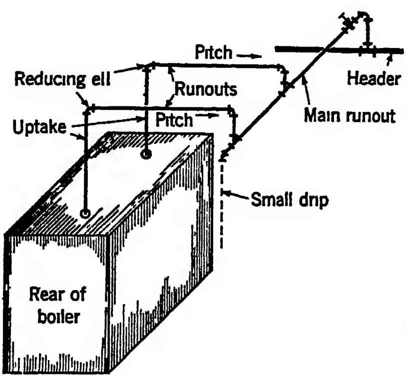


FIG. 3. OLD STYLE STANDARD BOILER CONNECTIONS

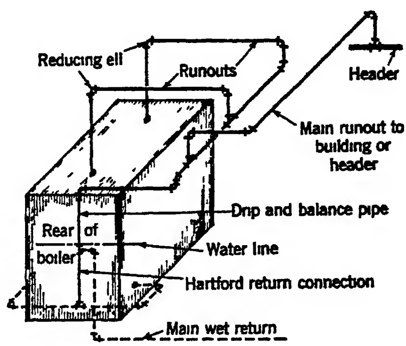


FIG. 4. APPROVED METHOD OF BOILER CONNECTIONS

toward the rear and is discharged into the rear drip, or balance pipe, without being carried over into the system.

Return

Cast-iron boilers are generally provided with return tappings on both sides, but steel boilers often are equipped with only one return tapping. A boiler with side return tappings will usually have a more effective circulation if both tappings are used. Check valves generally should not be used on the return connection to steam heating boilers from one and two pipe gravity systems because they are not always dependable inasmuch as a small piece of scale or dirt lodged on the seat will hold the tongue open and make the check useless. These valves also offer a certain amount of resistance to the returns coming back to the boiler, and in gravity systems will raise the water line in the far end of the wet return several inches[3]. However, if check valves are omitted and the steam pressure is raised with the boiler steam valve closed, the water in the boiler will be blown out into the return system with the accompanying danger of boiler damage. These objections are largely overcome with the Hartford return connection.

[3]See method of calculating height above water line for gravity one-pipe systems in Chapter 31.

Hartford Return Connection

In order to prevent the boiler from losing its water under any circumstances, the use of the Hartford Connection, or the Underwriters Loop, is recommended.

Fig. 5 shows this connection for a two-boiler installation. For a single boiler installation the connection is made as is indicated for one boiler. The essential features of construction of a Hartford Loop connection are: (1) A direct connection (made without valves) between the steam side of

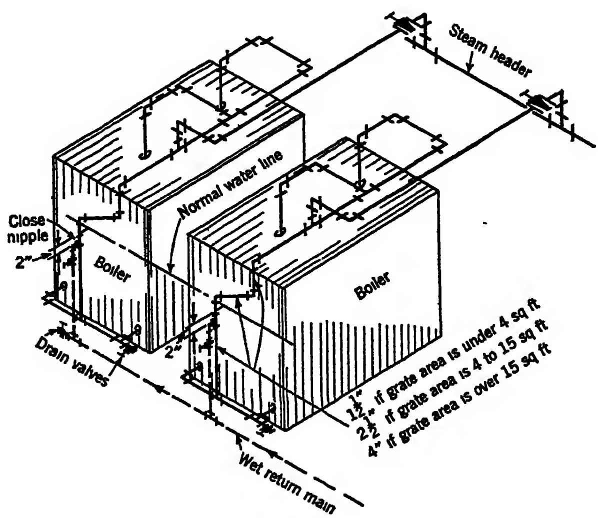


FIG. 5. THE HARTFORD RETURN CONNECTION

the boiler and the return side of the boiler, and (2) a close nipple connection about 2 in. below the normal boiler water line from the return main to the boiler steam and return balance connection.

Sizing Boiler Connections

Little authentic information is available on the sizing of boiler runouts and steam headers. Although many engineers prefer an enlarged steam header to serve as additional steam storage space, there ordinarily is no sudden demand for steam in a steam heating system except during the heating-up period, at which time a large steam header is a disadvantage rather than an advantage. The boiler header may be sized by first computing the maximum load that must be carried by any portion of the header under any conceivable method of operation, and then applying the same schedule of pipe sizing to the header as is used on the steam mains for the building. The horizontal runouts from the boiler, or boilers, may be sized by calculating the heaviest load that will be placed on the boiler at any time, and sizing the runout on the same basis as the building mains. The difference in size between the vertical uptakes from the

boiler and the horizontal main or runout is compensated for by the use of reducing ells (Figs. 3 and 4).

The following example illustrates the sizing of the boiler connections shown in Fig. 6.

Example 4. Determine the size of boiler steam header and connections (Fig. 6) if there are three boilers, two to carry 50 per cent of the load each, and the third to be used as a spare. The steam mains are based on ⅛-lb drop per 100 sq ft of equivalent direct radiation (EDR).

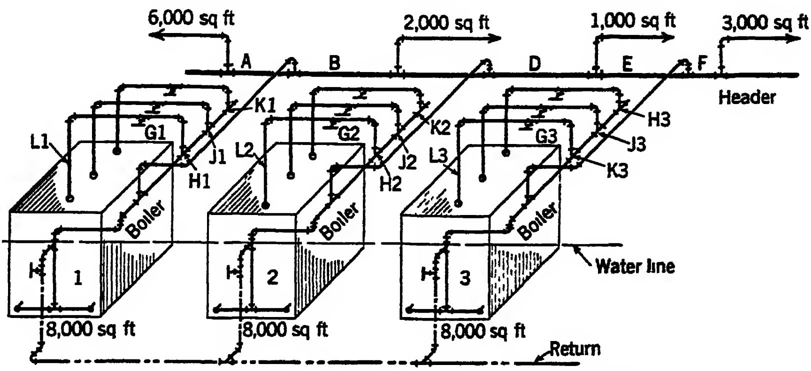


FIG. 6 BOILER STEAM HEADER AND CONNECTIONS

Solution:

Size of Boiler Header

When Operating on Boilers	Load on Various Portions of Header						Maximum Load
	A	*B*	*C*	*D*	*E*	*F*	
Nos. 1 and 2	6000	0	2000	4000	3000	3000	6000
Nos. 2 and 3	6000	6000	8000	2000	3000	3000	8000
Nos 3 and 1	6000	0	2000	2000	3000	3000	6000
Max Load	6000	6000	$\underline{\underline{8000}}$	4000	3000	3000	8000

8000 sq ft @ ⅛ lb per 100 ft = 6 in main. (See Table 7.)

Size of Boiler Runouts

The three runouts

G_1, G_2, G_3 = $\frac{8000}{3}$ = 2667 sq ft each @ ⅛ lb per 100 ft = 4 in. pipe.

H_1, H_2, H_3 = 2667 sq ft each @ ⅛ lb per 100 ft = 4 in. pipe[4] (See Table 7).

J_1, J_2, J_3 = 5333 sq ft each @ ⅛ lb per 100 ft = 5 in. pipe[4] (See Table 7)

K_1, K_2, K_3 = 8000 sq ft each @ ⅛ lb per 100 ft = 6 in. pipe[4] (See Table 7)

The uptakes from the boiler probably would be 6 in. pipe with a 6 in. × 4 in. reducing ell at top.

[4]*Note.*—As K_1, K_2, K_3 all carry 8000 sq ft and are 6 in. pipe, the whole runout including J_1, J_2 and J_3 and H_1, H_2 and H_3 and the leads from the boiler headers to the main steam header would also be made 6 in pipe.

Return connections to boilers in gravity systems are made the same size as the return main itself. Where the return is split and connected to two tappings on the same boiler, both connections are made the full size of the return line. Where two or more boilers are in use, the return to each may be sized to carry the full amount of return for the maximum load which that boiler will be required to carry. Where two boilers are used, one of them being a spare, the full size of the return main would be carried to each boiler, but if three boilers are installed, with one spare, the return line to each boiler would require only half of the capacity of the entire system, or, if the boiler capacity were more than one-half the entire system load, the return would be sized on the basis of the maximum boiler capacity. As the return piping around the boiler is usually small and short, it should not be sized to the minimum.

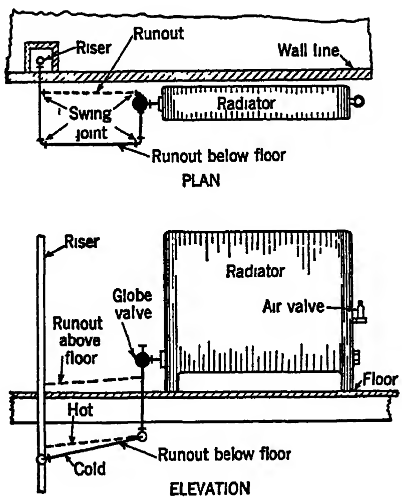


Fig. 7. One-Pipe Radiator Connections

With returns pumped from a vacuum or receiver return pump, the size of the line may be calculated from the water rate on the pump discharge when it is operating, and the line sized for a very small pressure drop, the size being obtained from the Chart for Pressure Drop for Various Rates of Flow of Water, Fig. 3, Chapter 35. The relative boiler loads should be considered, as in the case of gravity return connections.

Radiator Connections

Radiator connections are important on account of the number of repetitions which occur in every heating installation. They must be properly pitched and they must be arranged to allow not only for movement in the riser but, in frame buildings, for the shrinkage of the building. In a three story building this sometimes amounts to 1 in. or more. The simplest connection is that for the one-pipe system where only one radiator connection is necessary. Where the radiator runouts are located on the ceiling or under the floor, sufficient space usually is available to make

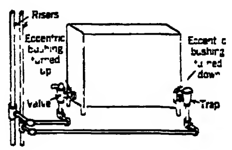


Fig. 8 Connections to Steam-Type Radiator for Two-Pipe System

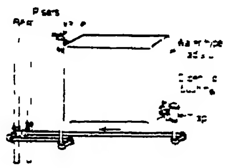


Fig 9. Top and Bottom Opposite End Radiator Connections

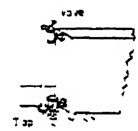


Fig 10 Top and Bottom Radiator Connections

a good swing joint with plenty of pitch, but where the runouts must come above the floor the vertical space is small and the runouts can project out into the room only a short distance. Fig. 7 illustrates two satisfactory methods of making runouts on a one-pipe gravity air vent system of either the up-feed or down-feed type, the runout below the floor being indicated in full lines and the runout above the floor in dotted lines. Sometimes it is necessary to set a radiator on pedestals, or to use high legs, in order to obtain sufficient vertical distance to accommodate above-the-floor runouts. Particular attention must be given to the riser expansion as it will raise the runout and thereby reduce the pitch.

Similar connections for a two-pipe system of the gravity air vent type are illustrated in Fig. 8 for the old steam type radiator. If the water type is used, the supply tapping is at the top instead of at the bottom, the runouts otherwise remaining as shown in Fig. 8. A satisfactory type of radiator connection for atmospheric, vapor, vacuum, sub-atmospheric, and orifice systems of both the up-feed and down-feed types is shown in Fig. 9.

While short radiators, not exceeding 8 to 10 sections, may be supplied and returned from the same end as indicated in Fig. 10, the top-and-bottom-opposite-end method is to be preferred in all cases where it can be used. On down-feed systems of the atmospheric, vapor, vacuum, sub-atmospheric, and orifice types, the bottom of the supply riser must be dripped into the return somewhat as illustrated in Fig. 11. On up-feed systems of the vapor and atmospheric types, where radiators in the basement are located below the level of the steam main, the drop to the radiator is dripped into the wet return and an air line is used to vent the return radiator connection into an overhead return line, as illustrated in Fig. 12. When the radiator stands on the floor below the main, the drip

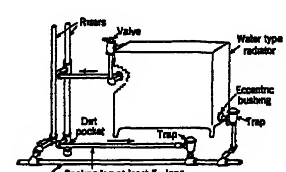


Fig. 11. Top and Bottom Opposite End Radiator Connections

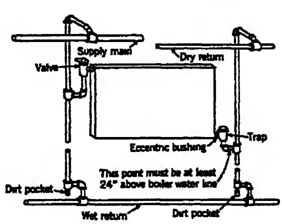


Fig. 12. Connections to Radiator Hung on Wall

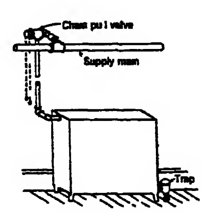


Fig. 13. Connecting Drop Riser Direct to Radiator

on the steam branch down to the radiator may be omitted if an overhead valve, as shown in Fig. 13, is used. This method is also suitable for vacuum, sub-atmospheric, and orifice systems.

Convector Connections

Convectors often are installed without control valves, a damper being used to shut off the flow of air to retard the heat transfer from the convector even though it is still supplied with steam. The piping connections for a convector with the inlet and outlet at the same end are shown in Fig. 14. There is no valve on the steam side but there is a thermostatic trap on the return. The damper for control is shown immediately above the convector. This piping is suitable for atmospheric, vapor, vacuum,

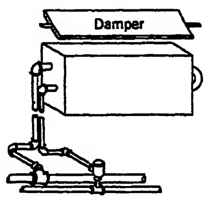


Fig. 14. Convector Connections Same End

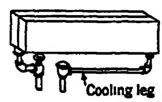


Fig 15 Horizontal Fin-Type Heating Unit

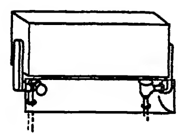


Fig. 16 Heating Unit Valves Behind Grille

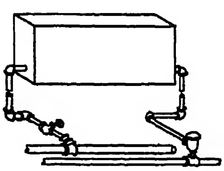


Fig 17. Heating Unit with Valves in Basement

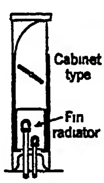


Fig 18 Fin-Type Heating Unit in Cabinet

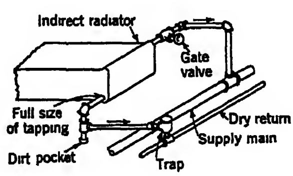


Fig. 19 Piping Connections to Indirect Radiators

sub-atmospheric, and orifice systems of the up-feed type. A similar unit with connections on opposite ends and suitable for the same systems is shown in Fig. 15. This unit has no damper but requires a valve on the steam connection for control. When valves must be located so as to be accessible from the supply air grille, the arrangement usually takes the form indicated in Fig. 16. A convector located in the basement and supplying air to a room on the floor above may be piped as pictured in Fig. 17 for all systems except gravity one-pipe or two-pipe systems. Convectors with damper control, installed in cabinets or under window sills, usually are connected as shown in Fig. 18.

Vapor systems with heating units in the basement where the returns are dry would be treated as in Fig. 19. Similar heating units where a wet return is available would be connected as shown in Fig. 20. If the dry

return were on a vacuum, atmospheric, sub-atmospheric or orifice system, the treatment would be identical.

On all heating units it is important to use a nipple the full size of the outlet and to reduce the pipe size to the normal return size required, by the use of a reducing ell, as indicated in Fig. 21.

Pipe Coil Connections

Pipe coils, unless coupled in a correct manner, often give trouble from short circuiting and poor circulation. The method of connecting shown in Fig. 22 is suitable for atmospheric, vapor, vacuum, sub-atmospheric, and orifice systems.

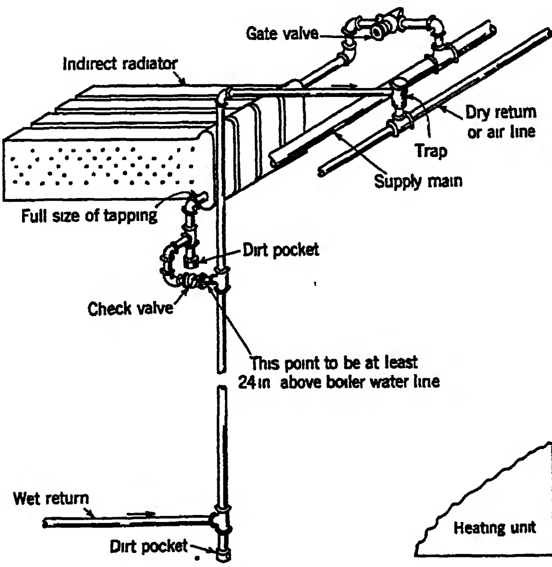


FIG. 20. TYPICAL PIPING CONNECTIONS TO CONCEALED HEATING UNITS WITH WET RETURNS

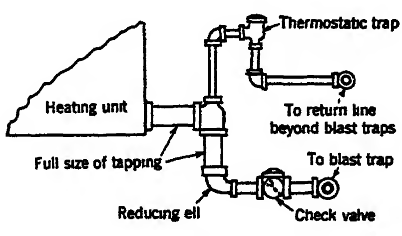


FIG. 21. HEATING UNIT RETURN CONNECTION WITH SEPARATE AIR LINE

Indirect Air Heater Connections

Heating units for central fan systems have simple connections on the steam side. The steam main is carried into the fan room and has a single branch tapped off for each row of heating units. Each of these main branches is split into as many connections as need be made to each row, governed by the number of stacks and the width of the stacks. Each stack must have at least one steam connection, and wide stacks are more evenly heated with two steam connections, one at each end.

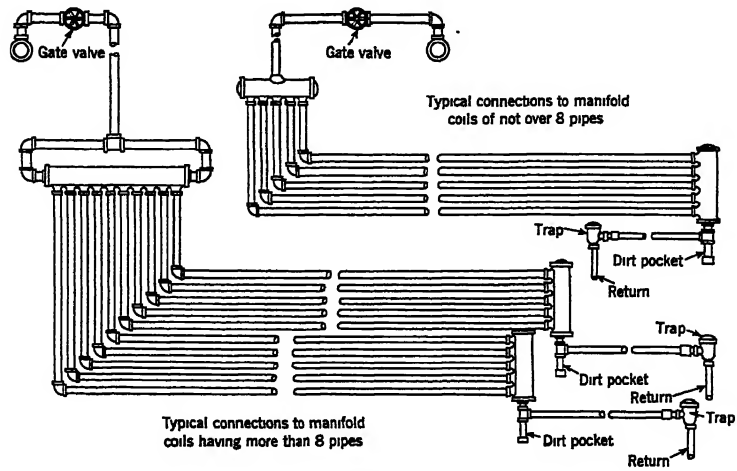


Fig. 22. Typical Pipe Coil Connections

The piping shown in Fig. 23 is for small stacks and has the steam connected at only one end. On the return side all of the returns are collected together through check valves and are passed through blast traps which are connected to the vacuum return or to an atmospheric return. The air from the stacks, in the case illustrated, passes up into a small air line and through a thermostatic trap into a line connecting into the return beyond the blast trap.

Where the stacks contain some thirteen or more sections, an auxiliary air tapping is made to the lower portion of one of the middle sections, in the manner illustrated in Fig. 24, to prevent air collecting at this point. Thermostatic control as applied to such heating units in modern practice

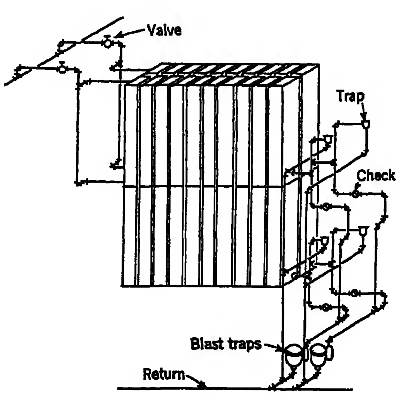


Fig 23. Supply and Return Connections for Heating Units of Central Fan Systems

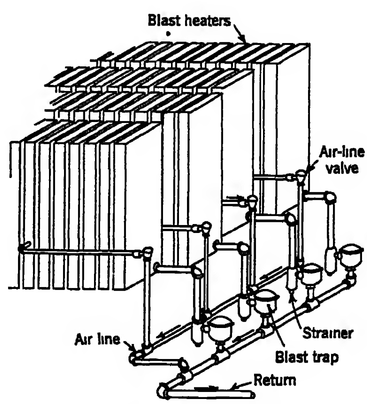


Fig. 24. Typical Connections to Central Fan System Heating Units Exceeding 12 Sections

consists of a thermostatic valve located in each main branch from the steam line so that each valve will open or close a complete row of stacks across the entire face of the heating unit. The stack closest to the outside air intake usually is not equipped with a thermostatic valve. A gate valve on the steam pipe to the first coil is operated manually to supply steam continuously in freezing weather. Good practice demands that the returns be connected in parallel with the steam supplies, with a separate steam trap for each bank of coils having a separately valved steam supply. This arrangement is illustrated in Fig. 23, for blast traps having external thermostatic by-passes and integral thermostatic by-passes, respectively.

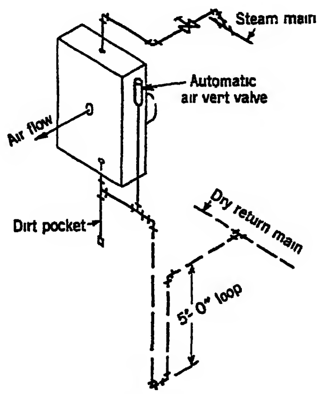


FIG. 25 UNIT HEATER CONNECTED TO ONE PIPE AIR VENT SYSTEM

A method of connecting a unit heater to a one pipe air-vent steam heating system is illustrated in Fig. 25.

PIPE SIZING FOR INDIRECT HEATING UNITS

Pipe connections and mains for indirect heating units are sized in a manner similar to radiators, but the equivalent direct radiation must be ascertained for each row of heating unit stacks and then must be divided into the number of stacks constituting that row and into the number of connections to each stack.

$$EDR = \frac{Q \times 60 \times (t_l - t_e)}{55.2 \times 240} = \frac{Q \times (t_l - t_e)}{220.8} \tag{3}$$

where

EDR = equivalent direct radiation, square feet.
Q = volume of air, cubic feet per minute.
t_e = the temperature of the air entering the row of heating units under consideration, degrees Fahrenheit.
t_l = the temperature of the air leaving the row of heating units under consideration, degrees Fahrenheit.
60 = the number of minutes in one hour.
55.2 = the number of cubic feet of air heated 1 F by 1 Btu.
240 = the number of Btu in 1 sq ft of EDR

Example 5. Assume that the heating units shown in Fig. 26 are handling 50,000 cfm of air and that the rise in the first row is from 0 to 40 F, in the second row from 40 to 65 F, and in the third row from 65 to 80 F. What is the load in EDR on each supply and return connection?

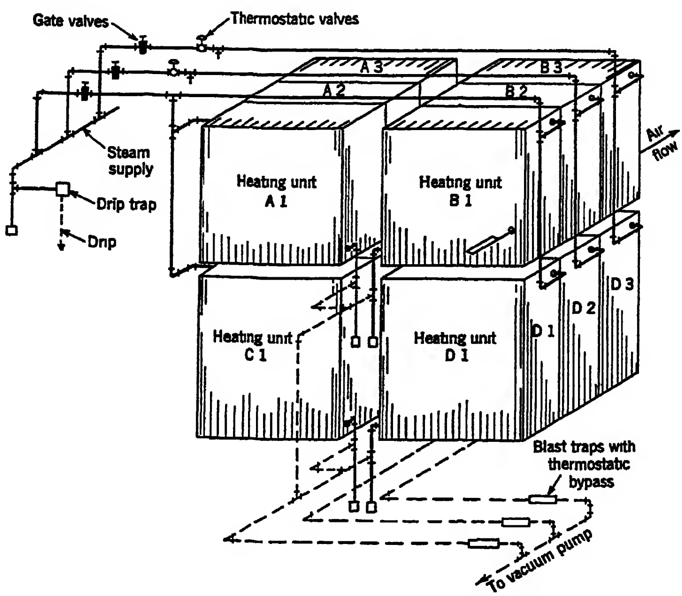


Fig 26. Typical Piping for Atmospheric and Vacuum Systems with Thermostatic Control (Central Fan System)

Solution. For row 1,

$$R = \frac{50{,}000 \times (40 - 0)}{220.8} = 9058 \text{ sq ft.}$$

For row 2,

$$R = \frac{50{,}000 \times (65 - 40)}{220.8} = 5661 \text{ sq ft.}$$

For row 3,

$$R = \frac{50{,}000 \times (80 - 65)}{220.8} = 3397 \text{ sq ft.}$$

Each row of heating units consists of four stacks and each stack has two connections so that the load on each stack and each connection of the stack is as follows:

Row	Total Load (EDR)	Stack Load[a] (EDR)	Connection Load[b] (EDR)
1	9058	2265	2265 or 1132
2	5661	1415	1415 or 708
3	3397	849	849 or 425

[a]One quarter of total row load

[b]One half of stack load if two steam connections are made; otherwise, same as stack load

The pipe sizes would then be based on the length of the run and the pressure drop desired, as in the case of radiators. It generally is considered desirable to place the indirect heating units on a separate system and not on supply or return lines connected to the general heating system

DRIPPING

Any steam main in any type of steam heating system may be dropped to a lower level without dripping if the pitch is downward with the steam flow. Any steam main in any heating system can be elevated if dripped (Fig. 27). Steam mains also may be run over obstructions without a change in level if a small pipe is carried below the obstruction to care for

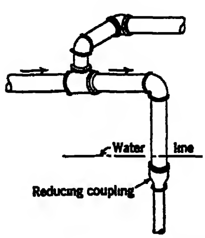


FIG. 27. DRIPPING MAIN WHERE IT RISES TO HIGHER LEVEL

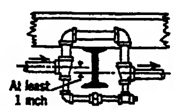


FIG 28. LOOPING MAIN AROUND BEAM

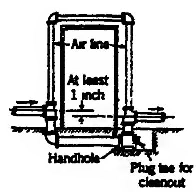


FIG. 29 LOOPING DRY RETURN MAIN AROUND OPENING

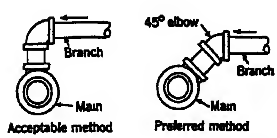


FIG. 30. METHODS OF TAKING BRANCH FROM MAIN

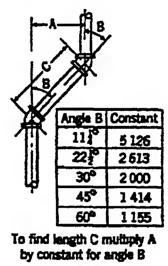


FIG. 31. CONSTANTS FOR DETERMINING LENGTH OFFSET PIPE

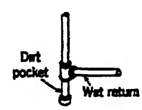


FIG. 32. DIRT POCKET CONNECTION

the condensation (Fig. 28). Return mains may be carried past doorways or other obstructions by using the scheme illustrated in Fig. 29; in vacuum systems it is well to have a gate valve in the air line.

Branches from steam mains in one-pipe gravity steam systems should use the *preferred connection* shown in Fig. 30, but where radiator condensation does not flow back into the main the *acceptable* method shown in the same figure may be used. This acceptable method has the advantage of giving a perfect swing joint when connected to the vertical riser or radiator connection, whereas the preferred connection does not give this swing without distorting the angle of the pipe. Runouts from the steam main are usually made about 5 ft long to provide flexibility for movement in the main.

Offsets in steam and return piping should preferably be made with

90-deg ells but occasionally fittings of other angles are used, and in such cases the length of the diagonal offset will be found as shown in Fig. 31.

Dirt pockets, desirable on all systems employing thermostatic traps, should be so located as to protect the traps from scale and muck which will interfere with their operation. Dirt pockets are usually made 8 in. to 12 in. deep and serve as receivers for foreign matter which otherwise would be carried into the trap. They are constructed as shown in Fig. 32.

On vapor systems where the end of the steam main is dripped down into the wet return, the air venting at the end of the main is accomplished by an air vent passing through a thermostatic trap into the dry return line as shown in Fig. 33. On vacuum systems the ends of the steam mains are dripped and vented into the return through drip traps opening into the return line. The same method may be used in atmospheric systems. A float type trap is preferable to a thermostatic trap for dripping steam mains and large risers. If thermostatic traps are used a cooling leg

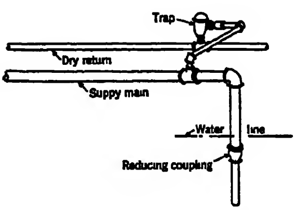


Fig. 33 Dripping End of Main into Wet Return

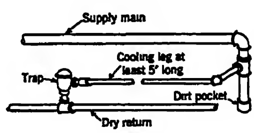


Fig 34 Dripping End of Main into Dry Return

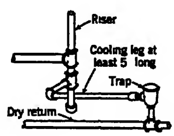


Fig 35 Dripping Heel of Riser into Dry Return

(Fig. 34) should always be provided. The cooling leg is for cooling the condensation sufficiently before it reaches the trap so the trap will not be held shut by too high a temperature. On down-feed systems of atmospheric, vapor, and vacuum types, the bottom of the steam risers are dripped in the manner shown in Fig. 35

PROBLEMS IN PRACTICE

1 ● What factors determine the size of steam piping and the allowable limit of capacity?

Factors which determine the size of steam piping are the desired initial pressure and the allowable drop in pressure which is permissable to maintain a pressure in the farthest radiator The length of run in sizing piping is important and it is generally considered as the distance along the piping from the source of steam supply to the farthest radiator, with allowances for resistance of elbows and valves expressed in terms of equivalent length.

2 ● When the size of pipe is still undetermined, what arbitrary percentage is usually added to the actual length to obtain the equivalent length?

Usually 100 per cent; in other words, the actual length is doubled to allow for the added drop produced by the valves, tees, elbows, and other fittings.

3 ● What are the major factors to be considered in determining the flow of steam in pipes?

a. The initial steam pressure available and the total pressure drop allowable between the source of steam supply and the end of the return system. The pressure drop should never exceed one half of the initial pressure.

b. The maximum steam velocity allowable When condensate is flowing against the steam, the velocity must not be so great as to produce water hammer, or hold up water in parts of the system until the steam flow is reduced sufficiently to permit the water to pass. The velocity at which disturbances take place depends upon·

1. Size of pipe
2. Whether pipe is vertical or horizontal
3. Pitch or grade of pipe
4. Quantity of water flowing against steam.

c. The equivalent length of run from the source of steam supply to the farthest heating unit, with allowance for friction in pipe fittings and valves

4 ● Name three fundamental considerations in designing the piping system for steam heating.

a. Provision for the distribution of suitable quantities of steam to the various heating units.

b. Provision for the return of condensate from the radiators and piping to the boiler.

c. Provision of means for expelling air from the radiators and piping.

5 ● Why is the proper reaming of the ends of pipe necessary?

The capacities of pipes depend upon the free area available for flow. In cutting the pipe this area may be restricted by a burr, which may decrease the capacity of a pipe more than 25 per cent in the smaller pipe sizes

6 ● a. What are the major factors to be considered when selecting a pressure reducing valve?

b. How should such valve be installed?

a. The initial pressure of the steam must be considered along with the desired reduced pressure. The connected load to be supplied must be known in square feet of equivalent direct radiation or in pounds of steam per hour. For operation with a continuous load, a semi-balanced or double-seated valve operated by a diaphragm gives good results Where the load is intermittent, as in process work or with thermostatically controlled blast heaters, a so-called dead end or single-seated valve should be used.

b. The pressure reducing valve should be installed in a horizontal line with a gate valve on each side, and with a by-pass operated by a valve The pressure balancing pipe from the diaphragm chamber should be connected into the top or side of the low pressure main not less than 15 ft from the reducing valve.

7 ● What is the usual expansion allowance and how it is compensated for in heating system supply risers?

The expansion of low pressure steam piping is normally taken as 1¼ to 1½ in. per 100 ft of pipe. With a five story building a double swing connection between the riser and the main will suffice. In buildings between 5 and 10 stories high the riser should be anchored near its center and have double swing connections to the main. For taller buildings expansion loops or riser offsets are used which are capable of handling a length of riser reaching 5 stories in either direction from the joint. The risers are anchored at each alternate 5 stories. All radiators must have double swing connections, and those connected above where the riser is anchored must be given greater pitch to insure their having proper grade when the riser is heated.

8 ● Why should all boiler steam supply tappings be used full size?

In order to operate at low steam velocities so the water in suspension can separate from the steam and remain in the boiler

9 ● What is the Underwriters Loop or the Hartford Connection?

An arrangement of piping on the returns to low pressure boilers wherein the return line is raised up nearly to the water line of the boiler and is then dropped back and connected to the boiler return inlet, the high point is connected by a balance pipe to the steam runout from the boiler on the boiler side of all stop valves With this loop no check valve is required, and water cannot be backed out of the boiler and into the return at a point lower than the invert of the pipe at the top of the loop.

10 ● What are the important factors in making radiator connections?

Connections to radiators should be made as direct as possible, of proper size, with ample pitch of piping and allowance for expansion.

11 ● Why should careful attention be given to proper dripping and drainage of steam piping?

The steam mains and risers must be quickly drained of condensate and where necessary vented of air in order to obtain a sufficient supply of steam to the radiators Proper drainage is also necessary to insure a noiseless heating system.

12 ● What is the limit of pressure drop usually recommended in a vacuum system?

Not over ⅛ lb (2 oz) per 100 ft of equivalent run, and not over 1 lb total drop.

13 ● When steam and condensation are flowing in the same direction, what is the maximum total pressure drop which should be used?

The maximum total pressure drop should not exceed one half of the initial steam pressure.

14 ● What does a proper installation of a pressure reducing valve include?

A strainer in front of the pressure reducing valve; a gate valve in front of the strainer, a gate valve after the reducing valve; a by-pass around the two gate valves, strainer, and pressure reducing valve; and a globe valve in the by-pass. Sometimes a safety valve on the low pressure side and pressure gages on both sides are installed. The high pressure line should be dripped just before the high pressure steam enters the pressure reducing valve assembly.

15 ● Will a pressure reducing valve which is reducing the steam pressure from 100 lb gage to 50 lb gage pass more or less steam than the same valve when reducing the steam pressure from 100 lb gage to 5 lb gage?

The valve will pass practically the same volume of steam in each case as the velocity of steam flowing through an orifice shows no material increase after the reduced absolute pressure has fallen to 58 per cent of the initial absolute pressure Because of its greater density, the weight of steam passed will be greater in the case of the reduction to 50 lb gage.

Chapter 33

HOT WATER HEATING SYSTEMS AND PIPING

One- and Two-Pipe Systems, Mechanical Circulation, Circulators, Pipe Sizes, Forced Circulation, Effect of Variations in Pipe Sizes, Gravity Circulation, Expansion Tanks, Relief Valves, Installation Details

A HOT water heating system is one in which water is the medium by which heat is carried through pipes from the boiler to the heating units. There are two general types, namely, *forced circulation* and *gravity circulation* systems. In the former the pressure head maintaining flow is produced mechanically, whereas in the latter the pressure head is produced by the differences in weight of the water in the flow and in the return risers.

Both forced circulation and gravity circulation systems may be further divided into high or low temperature classifications. The use of high temperature water in conjunction with forced circulation permits a material reduction in the size of piping and radiation, thereby lowering installation cost. The less bulky radiation of this type of system also lends itself to the current practice of concealing or recessing the heating units.

Low temperature water is considered as that which affords a heat emission per square foot of radiation of from 150 to 160 Btu, while high temperature water will deliver from 200 to 240 Btu.

The fundamental rule in the design of a hot water system is that the total friction head in any circuit will equal the pressure head causing the water to flow in the circuit. This means that it is necessary to size the pipe in any circuit so that the friction loss produced by the movement of a sufficient volume of water to handle the heating load will not be greater than the available pressure head.

In designing a hot water heating system, it is necessary to determine:

1. The heat losses of the rooms or spaces to be heated. (See Chapter 7.)
2. The size and type of boiler. (See Chapter 25)
3. The location, type, and size of heating units. (See Chapter 30)
4. The method of piping.
5. The type and size of circulating pump (if forced circulation).
6 Suitable pipe sizes.
7. The type and size of expansion tank.

The unit, a square foot of equivalent direct radiation, EDR, has been used for many years for rating purposes in both steam and hot water systems, but its use, especially in hot water systems, has always resulted in complications and confusion. It is the plan of THE GUIDE *to eventually eliminate this*

empirical expression and to substitute a logical unit based on the Btu. The Mb, *the equivalent of 1000 Btu, and the* Mbh, *the equivalent of 1000 Btu per hour, which have been approved by the* A.S.H.V.E., *are used in this chapter on hot water systems to replace the square foot of radiation formerly used.*

ONE- AND TWO-PIPE SYSTEMS

Pipe systems may be divided into two general types, namely, *two-pipe* and *one-pipe* systems. In a two-pipe system the piping is arranged so that the water flows through only one radiator during a circuit through the system, so that all radiators are supplied with water at practically the same temperature as that in the boiler. In a one-pipe system, the water flows through more than one radiator during its circuit. In that case, the first radiator receives the hottest water; the second radiator, somewhat cooler water; the third one, still cooler; and so on. As the temperature of the water supplied to a radiator is lowered, the size of the radiator must be increased and, consequently, the total heating surface for a one-pipe system must be greater than that for a two-pipe system for the same service.

The use of forced circulation in one-pipe systems, however, practically eliminates this objection. As the velocity is *increased* in a one-pipe

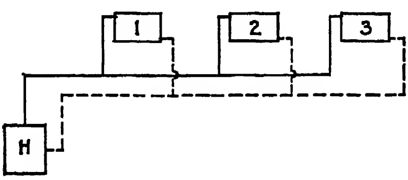


FIG. 1. A DIRECT RETURN SYSTEM

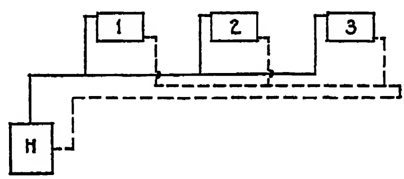


FIG. 2 A REVERSED RETURN SYSTEM

system, the drop in temperature is *decreased*, so that water at a higher average temperature is delivered to the radiators. This means that the radiators at the end of the main can be sized on the same basis as the radiators at the beginning of the main. If the system is correctly designed, the resulting error is less than the variation in calculating the heating load for the enclosure.

By making use of improved devices now available, forced circulation one-pipe systems may be calculated by the same procedure described later in the chapter for two-pipe systems. Operation as satisfactory as with a two-pipe system may be obtained.

Two-pipe systems may be divided into two classes, *direct return* systems (Fig. 1), and *reversed return* systems (Fig. 2). In a direct return system the water returns to the heater by a direct route after it has passed through its radiator and, as a result, the paths through the three radiators shown in Fig. 1 are of unequal lengths, the path through the first radiator being the shortest and that through the third radiator, the longest. In a reversed return system, the water returns to the heater by an indirect route after it has passed through the radiators, so that the paths leading through the three radiators shown in Fig. 2 are practically of equal length.

The reversed return system has an advantage over the direct return system in that it is more likely to function satisfactorily even though the pipe system is not accurately designed. For example, if in Fig. 2 all pipes are of one size, each of the three radiators will receive approximately the same quantity of hot water because the three paths are practically of equal length, whereas in Fig. 1, if all pipes are of the same size, Radiator 1 will receive more water than the others because the path through it is shorter than those through the other radiators. As a result, Radiator 1 will be filled with water at a higher average temperature than the remaining two radiators, and will therefore dissipate more heat. To prevent this unequal distribution of heat it is necessary to throttle the paths through Radiators 1 and 2 so that the friction heads of the three paths are equal when each radiator receives its proper quantity of water.

The two-pipe direct return system, with its inherent lack of balance, is the least satisfactory type of piping possible, yet is the most widely used. The modern applications of automatic heating require a system to be very nearly in balance so that uniform distribution of heat will be obtained.

Two-pipe systems must be balanced first by calculation and then by test after the plant is in operation. Unbalanced conditions in a forced circulation system are more detrimental to satisfactory operation than in the system circulated by gravity. The selection of orifices for correcting the unbalance must be more accurate. Due to the variations in water delivery from pipes, the accuracy of calculations is decreased, so that more reliance must be placed on actual test work. This is always costly and seldom completely satisfactory.

It would seem, then, that the *reversed return* two-pipe system, in which design errors are minimized, should be the logical choice.

A comparison of Fig. 1 and Fig. 2 may suggest that a reversed return system requires considerably longer mains than a direct return system. This is not always the case. For example, note the reversed return system of Fig. 3.

MECHANICAL CIRCULATION AND CIRCULATORS

The designer of a forced circulation system generally makes use of pumps available on the market unless he is able to buy to his specifications. Special equipment raises the initial cost of a system, so it is usual practice to incorporate stock pumps into the design wherever possible. Pumps of this type will have characteristics which govern, to a degree, the velocity selected for the heating system, yet a group of stock pumps has sufficient range of capacities to permit maintenance of an economical velocity. For example, suppose a system is to be designed to handle a 96 Mbh load, with a 20 F drop allowable in the system. 10 gpm will be required of the circulating equipment. Check pump characteristics to see which one will deliver 10 gpm against a head sufficiently high to allow a friction drop great enough to produce a satisfactory velocity in the piping system.

It is worthy of note that velocities may be too high, producing objectionable noises in the system. Higher velocities may be used with satisfactory results in industrial applications than in domestic systems.

Low head centrifugal pumps especially designed for hot water heating systems are used to provide the head necessary for forced circulation, and to improve the operation of gravity designed systems. These pumps operate with little noise and low power consumption, two features of prime importance to the satisfactory operation of a forced circulation system. They are installed in the return main close to the boiler, with all returns brought into the suction side of the pump. Gate valves should be installed on either side so that the pump can be removed without draining the system. A by-pass is not necessary as the friction drop through the pump is not sufficient to prevent gravity circulation if the pump should become inoperative.

Specially designed propeller type pumps are used also. This type of

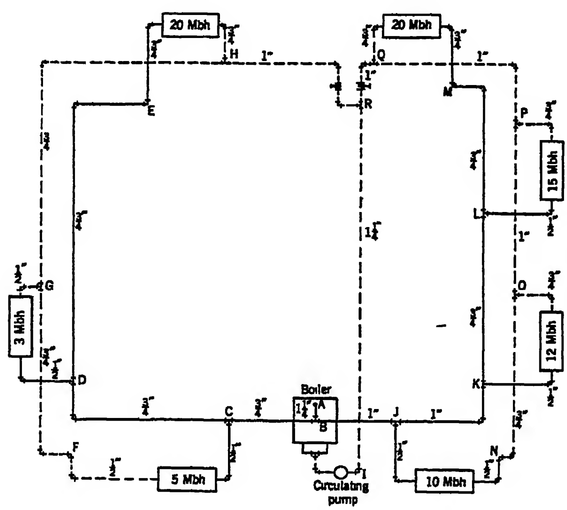


Fig. 3. A Forced Circulation Reversed Return System[a]

[a]Note that the numbers on the radiators indicate thousands of Btu per hour (Mbh) and not square feet

pump is installed in the return and is available for all the commercial pipe sizes used for hot water heating.

The motor may be controlled manually; however, forced circulation lends itself admirably to automatic control. In this case, the motor is controlled by a thermostat which can be tied in with the controls of the automatic firing device. In conjunction with flow control valves this permits an accurate control of the hot water in the radiation.

For exceptionally large installations such as central heating plants, circulating pumps of the centrifugal single stage type, having an average operating efficiency of 70 per cent against heads up to 125 ft, are sometimes used. It is generally advisable to install pumps in duplicate to provide for contingencies and to insure continuous operation. In such cases each pump should be made equal to the maximum capacity required.

PIPE SIZES

The pressure heads available in forced circulation systems are much greater than those in gravity circulation systems, consequently, higher velocities may be used in designing the system, with the result that smaller pipes may be selected and the first cost of the installation reduced. As the pipes of a heating system are reduced in size, the necessary increase in the velocity of the water increases both the cost of operation and the initial cost of the circulating equipment. The increased velocity of a forced circulation system offers a number of advantages, such as a much shorter heating-up period and a more flexible control of hot water circulation. This improved performance merits the small increase in operating cost necessary to mechanically circulate the system. The velocity required should be determined by calculation for the particular system under consideration.

Since the velocities in forced circulation systems are higher than those in gravity circulation systems, and since the friction heads in a heating system vary almost as the squares of the velocities, a given error in the calculation or assumption of a velocity is less important in a forced circulation system than in a gravity circulation system and, consequently, it is easier to design a satisfactory forced circulation system than a satisfactory gravity circulation system.

FORCED CIRCULATION

The following examples will illustrate the procedure to be followed in designing forced circulation systems:

Example 1. Assume that the longest path through 7 radiators shown in Fig. 3 consist of 200 ft of mains, 25 ft of radiator connections, 1 boiler, 1 radiator, 1 radiator valve, 1 stop cock, 12 ells, and 2 tees. Also assume that the short branch main contains the same number of fittings and that the main is 150 ft long with 7 ft of radiator connections. Design the piping for this system

Solution. The friction heads of boiler, radiator valve and tee may be expressed in terms of the friction head in 1 elbow according to the values given in Table 1. Having done this, the longest circuit consists of 225 ft of pipe and 34 elbow equivalents. The friction head of 1 elbow is approximately equivalent to that in a pipe having a length equal to 25 diameters. Assume that the average pipe size in this case will be 1 in. Referring to Table 2, 1 elbow equivalent of 1 in pipe is equal to 2 3 ft and the total equivalent length of the longest circuit is 299 ft of straight pipe. Similarly the equivalent length of the short branch is 231 ft.

Having determined the equivalent pipe length, the next step is to assume the rate at which the water is to be circulated through the complete system. The water may flow through the radiator so that it will cool 10 or 20 F or any other reasonable number of degrees, but in this case, assume a temperature drop of 20 F through the radiation. One gallon of water per minute with a density of 7 99 lb per gal at 215 F will deliver approximately 9600 Btu per hour, with a 20 F temperature drop

The total radiation load is 85,000 Btu per hour (85 Mbh) and therefore the pump must circulate 4250 lb of water per hour or 8.85 gpm for a temperature drop of 20 F.

Knowing that the rate of flow is 8 85 gpm, the next step is to determine from the pump characteristics which pump will produce a satisfactory velocity. Presume that four sizes of pumps are available delivering 8 85 gpm of water at the following heads: 18 in., 6 ft, 14 ft, and 25 ft. At these heads, the pumps would produce velocities sufficient to make available the following respective friction losses per foot: 60, 240, 562 and 1000 milinches. A comparison with Fig. 4 will show that little advantage is gained from the use of a 60 milinch friction loss, and that the use of a 1000 milinch friction loss will

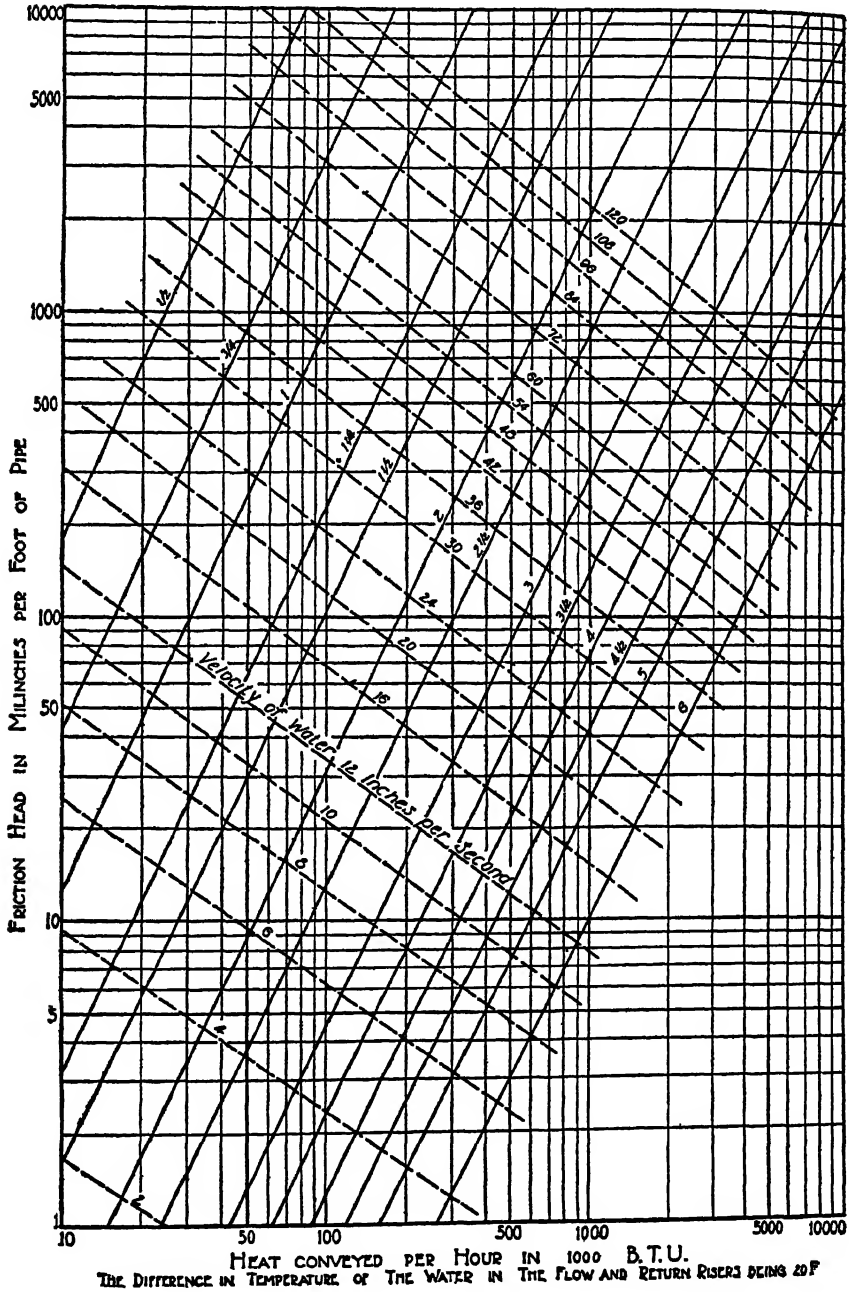


FIG. 4. FRICTION HEADS IN PIPES FOR A 20 F TEMPERATURE DIFFERENCE OF THE WATER IN THE FLOW AND RETURN LINES

TABLE 1 ELBOW EQUIVALENTS[a]

1 90-deg elbow	1 0
1 45-deg elbow	0.7
1 90-deg long turn elbow	0.5
1 open return bend	1 0
1 open gate valve	0.5
1 open globe valve	12.0
1 angle radiator valve	2.0
1 radiator	3.0
1 heater	3 0
1 tee	(Note[b])

[a]The loss of head in one elbow can be expressed in terms of the velocity head by the formula

$$h = \frac{v^2}{2g} \tag{1}$$

where

h = the loss of head in feet, v = the velocity of approach in feet per second, and $2g$ = 64 4 ft per second per second

[b]The loss of head in tees when water is diverted at right angles through a branch of the tee varies with the per cent diverted When the water diverted is less than 60 per cent of that approaching the tee, the loss of head, in elbow equivalents may be expressed as follows

$$h_e = \frac{v_1^2}{v_2^2} \tag{2}$$

where

h_e = the loss of head in elbow equivalents, v_1 = the velocity of approach, v_2 = the velocity of water diverted at right angles.

Values in elbow equivalents for the most common percentages of water diverted in a 1x1x1-in tee are as follows.

25%	16 0
33%	9 0
50%	4 0
100%	1 8

For other percentages the approximate values may be secured by interpolation When the water is diverted from the tee into a smaller size branch, as in a 1x1x¾-in tee, approximate values may be secured by means of Formula 2

obviously cost more for equipment and operation than the reduction in pipe size. The choice is therefore between 240 and 562 milinches If 562 milinches is selected, the circulation speed will be 50 per cent greater than for 240 milinches, but the power consumption will be increased The faster circulation will be of no practical importance in an average heating system as there could only be a difference of a few minutes in forcing water through a complete circuit of the system Therefore, use 240 milinches per foot with a static head of 6 ft on the pump. This will produce a satisfactory velocity with the most economical installation

The pipe size may now be selected from Fig. 4 making allowance for the fact that the 2 circuits are of unequal lengths Size the longest pipe circuit first. Section *AB* which is that portion of pipe connecting the boiler to the distributing main, must have a capacity of 85,000 Btu per hour (85 Mbh) and referring to Fig. 4, it will be noted that a 1¼ in. pipe is slightly too large to carry the total load Therefore a 1¼ in pipe may be satisfactorily selected for Sections *AB* and *RI*. The load branches at *B* with 28 Mbh in branch *BC* and 57 in *BJ* It will be noted that 28 Mbh is slightly over the capacity of a ¾ in. pipe, therefore use a ¾ in pipe in *BC* and a 1 in in Section *HR* Section *CD* should have a capacity for 23 Mbh, therefore use a ¾ in. pipe in *CD* and *GH*. A capacity of 20 Mbh is not sufficiently under the capacity of a ¾ in pipe to warrant the use of a ½ in. pipe, so use ¾ in in Sections *DE* and *FG*. The radiator branches are sized accordingly, with ½ in. up to 11,000 Btu (11 Mbh) and ¾ in. up to 25,000 Btu (25 Mbh)

Due to the difference in the equivalent lengths of each circuit, a static head of 6 ft on the pump would produce a greater velocity in the shorter branch than in the longer branch and consequently a higher friction loss per foot. This variation is usually a negligible factor in most installations and can be overlooked. Should the variation in friction head be sufficient to allow the use of smaller pipes, this factor should be taken into consideration. For purposes of illustration in this example, assume that Section *AB* is 4 ft, and then 4 × 240 = 960 milinches 71,800 − 960 = 70,840 milinches. 70,840

divided by 231 (total equivalent length of short branch) = 306 milinches per foot which is 66 milinches per foot more than is available in the longer circuit Therefore approximately 12 per cent more capacity is available in the pipes which will change the pipe size only a slight amount If it is necessary to correct this variation, a stop cock may be placed in the return line of the short branch and adjusted after installation

However, if the variation be of sufficient magnitude, the pipes in the shorter branch should be sized accordingly In this case the pipe size should be selected according to a frictional loss of 240 milinches per foot Due to the fact that Section *BJ* requires a capacity of 57 Mbh, which is slightly over 1 in, use a 1 in. pipe in *BJ* and *QR*. Section

Table 2 Capacities of Pipes in *Mbh* (1000 Btu per Hour) and **Velocities of Water in Pipes in Inches per Second** for Forced Circulation Systems with a Total Friction Head of 2 ft and for a Maximum Temperature Drop of 10 F[a]

1	2	3	4	5	6	7	8	9
Pipe Size (Inches)	Equivalent Length of Pipe (Feet[b])	Equivalent Total Length of Pipe in Feet in Longest Circuit						
		100	150	200	250	300	350	400
		Unit Friction Head, in Milinches						
		240	160	120	96	80	69	60
½	1	*6.2* **15**	*4.8* **12**	*4.1* **10**	*3.4* **9**	*2.9* **8**	*2 6* **7.5**	*2.4* **7**
¾	2	*13.2* **18**	*10.3* **14**	*8.6* **12**	*7.3* **11**	*6 2* **10**	*6 0* **9**	*5.5* **8.5**
1	2 3	*25.0* **22**	*19 2* **17**	*16 3* **15**	*14.4* **13**	*12.5* **12**	*12.0* **11**	*11.1* **10.5**
1¼	3 0	*52 8* **27**	*40.8* **21**	*34 8* **18**	*31.2* **16**	*27.8* **15**	*26.4* **14**	*24.0* **13**
1½	3 5	*79 2* **30**	*60 7* **23**	*51.2* **20**	*45.6* **18**	*40.8* **16**	*40.0* **15**	*36.0* **14**
2	4.0	*153.8* **36**	*120 0* **28**	*104 0* **24**	*93.5* **22**	*86.4* **20**	*81.5* **18**	*73.8* **17**
2½	6.0	*250 0* **41**	*192 0* **32**	*164 5* **28**	*149.0* **25**	*139.2* **22**	*135.8* **21**	*122.5* **19**
3	6 5	*444.0* **48**	*348.0* **37**	*294.0* **32**	*270.0* **29**	*254.0* **26**	*240.0* **24**	*223.0* **22**

[a]For other temperature drops the capacities of pipes are to be changed correspondingly. For example, for a temperature drop of 30 F, the capacities shown in this table are to be multiplied by 3 The velocities remain unchanged

[b]Approximate length of pipe in feet equivalent to one elbow in friction head This value varies with the velocity

JK has a load of 47 Mbh which is approximately 1 in. Therefore use 1 in. in *JK* and *PQ*. *KL* carries 35 Mbh which is approximately halfway between ¾ and 1 in Use ¾ in in *KL* and 1 in in *OP* Section *LM* requires a capacity of 20 Mbh, therefore use ¾ in. in *LM* and *NO* Size radiator branches as previously described

Many times a ½ in. pipe will prove to be too large, but at the present time it seems to be general practice to avoid pipes smaller than this size, especially for hot water installations.

If a number of heating systems are to be designed for similar conditions, *i.e.*, for a total friction head of 6 ft and a temperature drop through the radiators of 20 F when the maximum quantity of heat is being delivered

to the building, a table such as Table 3 may be prepared from the data of Fig. 4. Having this table, the pipe sizes for the system of Example 1 can be easily selected For example, for Sections *AB* and *RI*, each supplying 85 Mbh, the equivalent pipe length of the system is 299 ft. In the table the length shown nearest to this length is 300 ft. In the 300-ft column, a 1-in. pipe is too small and a 1¼-in. pipe is too large. The 1¼-in. pipe will therefore be selected. For other systems, it will be economical to operate with different friction heads, and tables may be prepared similar

TABLE 3. CAPACITIES OF PIPES IN *Mbh* (1000 BTU PER HOUR) AND VELOCITIES OF WATER IN PIPES IN INCHES PER SECOND FOR FORCED CIRCULATION SYSTEMS WITH A TOTAL FRICTION HEAD OF 6 FT AND FOR A MAXIMUM TEMPERATURE DROP OF 10 F[a]

1	2	3	4	5	6	7	8
Pipe Size (Inches)	Equivalent Length of Pipe (Feet[b])	Equivalent Total Length of Pipe in Feet in Longest Circuit					
		200	300	400	600	800	1000
		Unit Friction Head, in Milinches					
		360	240	180	120	90	72
½	1	*7.4* 18	*6.0* 15	*5.0* 13	*3.8* 10	*3.4* 9	*3.1* 7.5
¾	2	*15.8* 22	*12.7* 18	*10.8* 16	*8.4* 12	*7.7* 11	*6.7* 9
1	2.5	*30.0* 27	*24.0* 22	*20.4* 19	*15.8* 15	*13.9* 13	*12.5* 11
1¼	3.3	*64.8* 33	*52.5* 26	*44.4* 23	*33.6* 18	*30.0* 16	*26.8* 14
1½	4.0	*96.0* 37	*76.8* 31	*64.8* 26	*50.1* 20	*44.7* 18	*40.8* 15
2	5.0	*192.0* 44	*153.0* 36	*130.0* 30	*100.1* 24	*90.0* 21	*78.0* 18
2½	6.0	*300.0* 50	*244.0* 41	*206.0* 35	*161.0* 26	*144.0* 24	*130.0* 21
3	7.5	*550.0* 58	*436.0* 48	*368.0* 42	*287.0* 32	*249.0* 27	*228.0* 24

[a]For other temperature drops the capacities of pipes are to be changed correspondingly. For example, for a temperature drop of 30 F, the capacities shown in this table are to be multiplied by 3. The velocities remain unchanged

[b]Approximate length of pipe in feet equivalent to one elbow in friction head This value varies with the velocity

to Tables 2 and 4, which are based on total friction heads of 2 and 18 ft, respectively.

Example 2. Design a direct return two-pipe forced circulation system for the layout shown in Fig. 5 assuming a temperature drop of 10 F through the radiation. For this system the length of the pipe line from the boiler to the highest radiator on the farthest riser and back to the boiler is about 250 ft. There are about 16 elbow equivalents having an equivalent pipe length of about 50 ft, so the total equivalent pipe length is about 300 ft.

Solution. The same pipe size tables may be used as those developed for the reversed return system of Fig. 3, Table 3 which provides for a friction head of 6 ft.

Referring to the column for an equivalent total length of 300 ft for Sections *AB* and *KA*, each supplying 117.6 Mbh, it will be found that a 1½-in pipe is too small and a 2-in. pipe is too large. Consequently, a 1½-in pipe is selected for the flow line *AB*, and a 2-in pipe for the return line *KA*. For Sections *BC* and *JK*, each supplying 88 Mbh, a 1½-in pipe is only slightly too small and it is selected. The remaining pipe sizes are selected in a similar manner and recorded in Fig. 5. For a temperature drop of 10 F, 24.5 gpm of water must be circulated. The pump to select is one which has its highest efficiency when it is delivering 24.5 gpm against a 6-ft head.

To secure a correct distribution of hot water among the several risers it is necessary,

TABLE 4. CAPACITIES OF PIPES IN *Mbh* (1000 BTU PER HOUR) AND **Velocities of Water in Pipes in Inches per Second** FOR FORCED CIRCULATION SYSTEMS WITH A TOTAL FRICTION HEAD OF 18 FT AND FOR A MAXIMUM TEMPERATURE DROP OF 10 F[a]

1	2	3	4	5	6	7
		EQUIVALENT TOTAL LENGTH OF PIPE IN FEET IN LONGEST CIRCUIT				
PIPE SIZE (INCHES)	EQUIVALENT LENGTH OF PIPE (FEET[b])	200	400	600	800	1000
		UNIT FRICTION HEAD, IN MILINCHES				
		1080	540	360	270	216
½	1.0	*12.7* **32**	*8.6* **23**	*7.2* **18**	*6.2* **15**	*5.5* **13**
¾	2.0	*27.5* **40**	*18.7* **28**	*15.1* **22**	*13.7* **19**	*11.5* **17**
1	2.5	*55.0* **48**	*36.8* **34**	*30.0* **27**	*26.4* **23**	*22.6* **20**
1¼	3.0	*122.0* **59**	*81.5* **42**	*66.0* **33**	*58.3* **28**	*50.5* **25**
1½	4.0	*182.0* **66**	*122.0* **46**	*98.2* **37**	*86.2* **31**	*74.2* **27**
2	5.0	*371.0* **80**	*252.0* **56**	*201.0* **45**	*180.0* **38**	*151.0* **33**
2½	7.0	*598.0* **91**	*407.0* **65**	*323.0* **51**	*287.0* **43**	*240.0* **38**
3	9.0	*1110.0* **107**	*790.0* **76**	*598.0* **60**	*527.0* **51**	*443.0* **44**

[a]For other temperature drops the capacities of pipes are to be changed correspondingly. For example, for a temperature drop of 30 F, the capacities shown in this table are to be multiplied by 3. The velocities remain unchanged.

[b]Approximate length of pipe in feet equivalent to one elbow in friction head. This value varies with the velocity.

as previously stated, to introduce special resistances to balance the several risers, as follows.

The first riser is 80 ft nearer the boiler than the fifth riser. In order that the two may be balanced, *i e*, that they may operate under equal pressure heads, resistance must be added to the first riser equal to the friction head in the 80 ft of flow main from *B* to *F* plus that in the 80 ft of return main from *G* to *K*.

It will be noted from Table 3 that the unit friction head is about 240 milinches per foot. The total friction head in the flow and return mains between the first and fifth risers is therefore 160 × 240 or 38,400 milinches, or a little more than 3 ft, which must be supplied by additional resistance in the first riser to prevent its having an advantage over the fifth riser.

This resistance can be supplied by a calibrated and adjusted modulating valve or by

an orifice resistor in a union. If the orifice resistor is to be used, its size may be selected from Table 5 as follows:

The lower part of the first flow riser supplies 28.8 Mbh. According to Table 3, it should be a 1-in. pipe and would have a velocity of 22 in. per second, if it were supplying 24 Mbh. Since it is supplying 28.8 Mbh, the velocity will be about 26 in. per second. From Table 5 it will be found that for a 1-in. pipe and a velocity of 24 in. per second, an 0.45-in. orifice will produce a loss of head of 37,000 milinches. For a velocity of 26 in. per second, the loss of head will be somewhat more, probably about 43,000 milinches, the difference between it and the required resistance will be about 10 per cent, which is permissible, and the 0.45-in. orifice is selected.

The sizes of the orifice resistors for the second, third, and fourth risers are selected in a similar manner and found to be 0.45 in., 0.50 in., and 0.55 in., respectively.

If the design of the system of Fig. 5 is to be extremely refined, the gravity pressure heads produced by the risers should be taken into consideration. With water at 220 F and 210 F, respectively, in the risers, the

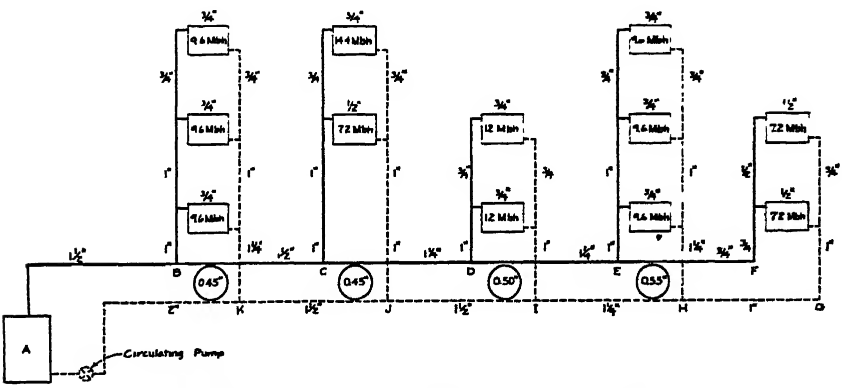


FIG. 5. A FORCED CIRCULATION DIRECT RETURN SYSTEM

gravity head is 50 milinches per foot of water column or 25 milinches per foot of flow and return pipe. The pump pressure head in this case is 240 milinches per foot of pipe, and the gravity head, being only one tenth as large as the pump head, may be neglected without serious error. This is generally done.

Temperatures of 220 F and 210 F would be used only during the coldest weather for which the system is designed. At other times the temperatures would be lower, the temperature drop smaller, and the gravity heads smaller. The pump pressure head remains constant throughout the season if the pump is operated at a constant speed and, consequently, the gravity head is generally less than one-tenth of the pump head.

Effect of Variations in Pipe Sizes

The pipe sizes for the several parts of the system selected from the tables are only approximately correct but the resulting error should be negligible as may be seen from the following study. Assume, as an extreme case, that the error in pipe size is so large that the water flows twice as fast through one of the radiators as through the others. This would make the friction head through this radiator almost four times as

Table 5 Friction Heads (in Milinches) of Central Circular Diaphragm Orifices in Unions

Diameter of Orifices (Inches)	Velocity of Water in Pipe in Inches per Second									
	2	3	4	6	8	10	12	18	24	36
					¾-in. Pipe					
0.25	1300	2900	5000	11,300	20,800	32,000	45,000			
0.30	650	1450	2500	5700	10,400	16,000	23,000	57,000		
0.35	330	740	1300	2900	5200	8000	12,000	26,000	47,000	
0.40	170	380	660	1500	2600	4000	6800	13,000	24,000	53,000
0.45		185	330	740	1300	2000	2900	6500	12,000	27,000
0.50			155	350	620	970	1400	3200	5700	13,000
0.55			75	170	300	480	700	1600	2800	6400
					1-in. Pipe					
0.35	900	2000	3500	7800	14,000	22,000	32,000			
0.40	460	1000	1800	4000	7200	12,000	17,000	37,000	65,000	
0.45	270	570	1000	2300	4100	6400	9300	21,000	37,000	
0.50	160	330	580	1400	2300	3700	5400	12,000	22,000	50,000
0.55		190	330	750	1300	2200	3000	7000	13,000	28,000
0.60			200	440	800	1300	1800	4200	7400	17,000
0.65			120	260	460	720	1100	2400	4300	10,000
					1¼-in. Pipe					
0.45	1000	2250	4000	8900	16,000	25,000	36,000			
0.50	660	1450	2600	5800	10,400	16,400	23,000	53,000		
0.55	430	950	1700	3800	6800	10,500	15,000	34,000	60,000	
0.60	280	630	1100	2500	4400	6900	10,000	22,000	40,000	
0.65	190	420	750	1700	3000	4700	6700	15,000	27,000	60,000
0.70		285	510	1150	2000	3100	4500	10,000	18,000	40,000
0.75		190	330	750	1300	2100	3000	6700	12,000	26,000
					1½-in. Pipe					
0.55	850	1900	3300	7400	13,000	21,000	30,000			
0.60	600	1300	2300	5400	8600	16,800	21,000	50,000		
0.65	400	850	1500	3600	7200	10,400	14,000	30,000	53,000	
0.70	260	600	1100	2600	4400	7000	10,000	21,000	39,000	
0.75	180	400	760	1800	3000	5000	7000	14,000	28,000	
0.80		300	540	1200	2200	3200	5000	10,200	19,000	45,000
0.85		200	380	860	1600	2300	3000	7800	13,000	30,000
					2-in. Pipe					
0.70	890	1850	3500	7400	14,000	22,300	33,000			
0.80	470	975	1800	3900	7400	11,700	17,000	37,000		
0.90	255	560	1000	2200	4200	6500	9500	20,500	38,000	
1.00	160	340	610	1320	2520	4000	5800	12,500	23,000	49,000
1.10		214	375	850	1600	2500	3700	7900	14,000	30,000
1.20			195	460	950	1360	1910	4200	8100	16,800
1.30				275	525	980	1375	3100	4400	8850

Note.—The losses of head for the orifices in the 1½-in. and 2-in. pipe were calculated from those in the smaller pipes, the calculations being based on the assumption that, for any given velocity, the loss of head is a function of the ratio of the diameter of the pipe to that of the orifice. This had been found to be practically true in the tests to determine the losses of head in orifices in ¾-in., 1-in., and 1¼-in. pipe, conducted by the Texas Engineering Experiment Station, and also in the tests to determine the losses of head in orifices in 4-in., 6-in., and 12-in. pipe, conducted by the Engineering Experiment Station of the University of Illinois. (*Bulletin* 109, Table 6, p. 38, Davis and Jordan).

large as those through the other radiators. The result would be that the water, in flowing through the radiator, would cool 5 F instead of 10 F. The mean water temperature in the radiator would then be 217½ F instead of 215 F, and the mean temperature difference, water to air, would be 147½ F instead of 145 F. The heat dissipated by the radiator would therefore be about 2 per cent more than calculated. It is evident that this difference in heat dissipation is smaller than the difference between

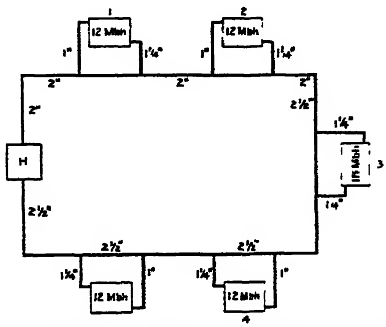


FIG. 6. A ONE-PIPE GRAVITY CIRCULATION SYSTEM

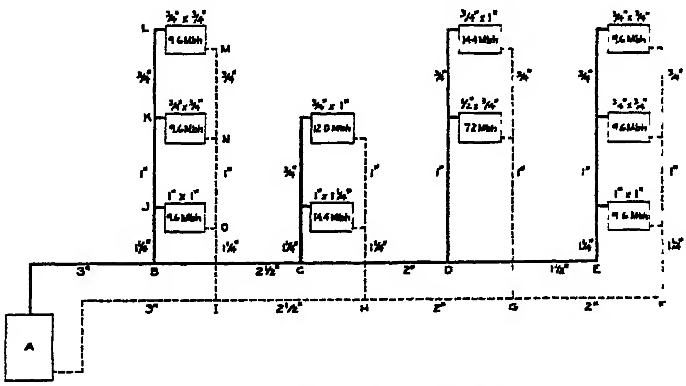


FIG. 7. A TWO-PIPE DIRECT RETURN GRAVITY CIRCULATION SYSTEM

the calculated heat losses and the actual heat losses, and also smaller than the average difference between the calculated radiator sizes and the nearest stock sizes selected.

GRAVITY CIRCULATION

For gravity circulation, the one-pipe system shown in Fig. 6 and the two-pipe direct return system shown in Fig. 7 are probably in most common use.

The one-pipe system has the disadvantage that the radiator nearest the boiler is the only one which receives water at approximately the temperature at which it leaves the boiler. All other radiators receive cooler water and must be proportionally increased in size, so the total heating

surface in the system is considerably larger than that in a corresponding two-pipe system.

The pipe sizes in gravity circulation systems may be varied. As the pipe sizes are decreased, the temperature drop through the radiators, which produces circulation, is increased and it becomes necessary to increase the temperature of the water leaving the boiler so that the mean temperature in the radiator remains constant. For example, Fig. 8 shows diagrammatically an elementary heating system which will function with either 1½-in. or 1-in. pipe. The radiator is required to deliver 27 Mbh, and the circuit consists of 30 ft of pipe and 20 elbow equivalents.

If 1½-in. pipe is used, the system will operate correctly if the water temperatures in the flow and return risers are 200 F and 180 F, respectively. The mean water temperature in the radiators will then be 190 F and, if the radiator is located in air having a temperature of 70 F, the size of the radiator must be sufficient to deliver 27 Mbh under these conditions.

If 1-in. pipe is used, the system will function correctly with water temperatures in the flow and return risers of 210 F and 170 F, or of 200 F

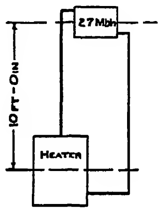


Fig. 8 An Elementary System

and 160 F. In the first case, the mean water temperature is again 190 F and the same size radiator may be used as with the 1½-in. pipe, but the temperature of the water leaving the boiler must be raised from 200 F to 210 F. In the second case, the temperature of the water leaving the boiler is the same as for the 1½-in. pipe, but the mean water temperature in the radiator is lowered from 190 F to 180 F, and theoretically the size of the radiator should be increased about 12½ per cent to deliver the required 27 Mbh (See Table 3, Chapter 6, 1933 Guide, also refer to Question 3, page 621).

This indicates the extent to which pipe sizes and radiator sizes may be decreased by increasing the temperatures of the water in the boiler, as is possible in closed systems and in open systems in which the open expansion tank is located sufficiently high to secure a pressure in the boiler equal to that existing in the boiler of the closed system.

Example 3. Design a one-pipe gravity circulation system for the layout shown in Fig. 6. Assume that the main circuit consists of 150 ft of pipe, 7 elbows, and one boiler.

Solution. Replace the boiler by 3 elbow equivalents and assume that the size of the main will be about 2 in. According to Table 6, Column 2, a 2-in elbow is equivalent to 4 ft of pipe, and the total equivalent length of the main will be about 150 plus 40, or 190 ft Assuming that the center of the boiler will be about 4 ft lower than the horizontal portion of the main and that the temperature drop in the system is to be 35 F, Table 6 may be used to determine the size of the mains. Note from Column 8, for a 200-ft length, that a 2-in. main will supply 48 Mbh and a 2½-in. main, 75 4 Mbh Since the system to be designed is to supply 66 Mbh, a 2-in pipe is too small and a 2½-in pipe

TABLE 6 CAPACITIES OF MAINS IN *Mbh*, FOR ONE-PIPE AND FOR TWO-PIPE DIRECT RETURN GRAVITY CIRCULATION SYSTEMS WITH A TOTAL FRICTION HEAD OF 0.6 IN., A TEMPERATURE DROP OF 35 F, WHEN THE MAINS ARE 4 FT ABOVE THE CENTER OF THE BOILER

1	2	3	4	5	6	7	8	9	10	11
Pipe Size (Inches)	Equivalent Length of Pipe (Feet[a])	Equivalent Total Length of Pipe in Feet in Longest Circuit								
		75	100	125	150	175	200	250	300	350
		Unit Friction Head, in Milinches								
		80	60	48	40	34	30	24	20	17
1½	3.0	*43.0*	*37.5*	*33.0*	*30.0*	*27.0*	*25.0*	*22.2*	*20.2*	*18.7*
2	4.0	*83.0*	*72.0*	*63.0*	*57.0*	*51.0*	*48.0*	*42.0*	*38.0*	*35.0*
2½	4.5	*140.0*	*115.0*	*100.0*	*90.0*	*81.5*	*75.[illegible]*	*67.2*	*61.0*	*56.0*
3	5.0	*234.0*	*204.0*	*175.5*	*160.0*	*143.0*	*133.0*	*110.0*	*107.5*	*100.0*
3½	5.5	*347.0*	*300.0*	*260.0*	*236.0*	*214.0*	*200.0*	*177.0*	*160.0*	*146.0*
4	6.0	*490.0*	*422.0*	*370.0*	*334.0*	*297.0*	*278.0*	*248.0*	*223.0*	*205.0*

[a]Approximate length of pipe in feet equivalent to one elbow in friction head. This value varies with the velocity

too large. The solution is to use some 2-in. and some 2½-in. pipe. Since the 2½-in. is nearer the correct size than the 2-in., select 2-in. pipe for the first 50 or 60 ft out of the boiler and 2½-in. for the remaining pipe back to the boiler.

Tables 7 and 8 may be used to design the radiator risers and connections. According to Table 7, for 12 Mbh the flow riser should be ¾ in. and the return riser 1 in., and the riser branches should be 1 in. and 1¼ in., respectively. Note that according to Table 8, both radiator tappings should be 1 in. To simplify the construction, select 1-in. flow

TABLE 7. MAXIMUM CAPACITIES OF RISERS[a] IN *Mbh*, AND Velocities of Water in Pipes in Inches Per Second FOR ONE-PIPE AND FOR TWO-PIPE DIRECT RETURN GRAVITY CIRCULATION SYSTEMS WITH A DROP OF 35 F THROUGH EACH RADIATOR

Pipe Size (Inches)		Equivalent Length of Pipe (Feet[c])	1st Floor[b]			2nd Floor	3rd and 4th Floors
				Vel. (In per Sec[d])			
Flow	Return		*Mbh*	Flow	Return	*Mbh*	*Mbh*
½	½	1.0				*5*	*6.2*
½	¾					*6.4*	*8.0*
¾	¾	1.5	*9*	2.3	2.3	*10.1*	*14.0*
¾	1		*12*	3.2	2.0	*12.8*	*17.1*
1	1	2.0	*18*	2.5	2.5	*20*	*26.0*
1	1¼		*21*	3.0	2.0	*25.2*	*34*
1¼	1¼	3.0	*26*	3.0	3.0	*43*	*55*
1¼	1½		*34*	4.0	2.5		
1½	1½	3.5	*48*	3.0	3.0		

[a]This table is based on pressure heads of 450, 1800, 3150, and 4500, respectively, for the first, second, third, and fourth floor radiators, and on friction heads of 200 milinches for the first floor radiators and connections, and 700 milinches for all other radiators and their connections.
[b]The riser branches, the piping which connects the risers to the mains, are to be one size larger than the risers.
[c]Approximate length of pipes in feet equivalent to one elbow in friction head. This value varies with the velocity.
[d]Velocities apply to the riser branches.

risers with 1-in. riser branches and 1-in. radiator tappings. Also select 1¼-in. return risers with 1¼-in riser branches, and 1¼-in. radiator tappings Similarly, for 18 Mbh, select 1¼-in flow and return risers and riser branches, and 1¼-in. radiator tappings.

To develop a rule for determining radiator sizes, assume a system similar to that of Fig. 6, in which the total temperature drop is to be 35 F and which is equipped with 7 radiators, all radiators dissipating equal quantities of heat. The mean temperature of the water in the radiators will be reduced 5 F for each successive radiator. If the mean temperature of the water in the first radiator is 200 F, the mean temperature of the water in the seventh radiator will be 170 F, and, according to Table 3, Chapter 6, of the 1933 Guide, the heat dissipation of these two radiators will be to each other as 868 is to 617, or as 140 is to 100, and therefore if the last radiator is to dissipate as much heat as the first, its size must be 40 per cent larger.

Example 4. Design a two-pipe, direct return, gravity circulation system for the layout shown in Fig. 7. Assume that the main circuit from the boiler to the farthest flow riser and from the farthest return riser back to the boiler consists of 160 ft of pipe, 6 elbows, and 1 boiler.

Table 8. Maximum Capacities of Radiator Connections in *Mbh*, for One-Pipe and for Two-Pipe Direct Return Gravity Circulation Systems with a Temperature Drop of 35 F Through Each Radiator

Pipe Size		Equivalent Length of Pipe (Feet[a])	1st Floor	2nd, 3rd, and 4th Floors
Flow	Return		*Mbh*	*Mbh*
½	½	1.0	4.1	5.9
½	¾		5.2	7.5
¾	¾	1.5	7.0	10.5
¾	1		9.1	13.0
1	1	2.0	12.5	17.8
1	1¼		17.5	23.2
1¼	1¼	3.0	23.3	33.2

[a]Approximate length of pipe in feet equivalent to one elbow in friction head. This value varies with the velocity.

Solution. Replacing the boiler by 3 elbow equivalents and assuming that the largest size of the main will be about 3 in, the total equivalent length of the main will be 160 plus 45, or 205 ft. Assuming that the center of the boiler will be about 4 ft lower than the horizontal portion of the main, and that the temperature drop will be 35 F for the system, the pressure head caused by the difference in weight between the water in the flow and return risers joining the mains to the boiler will be about 0.6 in. of water, or about one-fortieth of the pressure head produced by the circulating pump selected for the system of Fig 3

Table 6 may be used to determine the size of the main as follows Refer to Column 8 and note that for Sections *AB* and *IA*, which supply 105.6 Mbh, a 3-in. pipe is too large and a 2½-in. pipe is too small; hence, select 2½ in. for Section *AB* and 3 in. for Section *IA*. For Sections *BC* and *HI*, which supply 76.8 Mbh, a 2½-in. pipe is almost exactly the correct size and is selected for both sections.

For the forced circulation system of Fig. 5, the pressure head produced by the circulating pump is used to force the water through the mains and also through the risers Gravity circulation systems have two distinct pressure heads. One is produced by the difference in weight of the water in the flow and return risers adjacent to the boiler, and is the boiler pressure head, which in this case is 0.6 in. The other pressure head is produced by the difference in weight of the water in the flow and return risers adjacent to the radiators, and is the radiator pressure head. If the temperature drop through the radiators is about 35 F, and if the story heights of the building are 9 ft and the distance from the center of the first floor radiator to the average level of the main is 3 ft, the

radiator pressure head of the first floor radiator is about 450 milinches and the pressure heads of the radiators on the upper floor are 1350 milinches greater than those on the next lower floors.

Tables 6 and 7 are based on the assumption that the boiler pressure head must be equal to the friction head in the mains, and that the several radiator pressure heads must be equal to the respective radiator and riser friction heads.

To design the radiator risers, use Table 7 and begin with the set nearest the boiler. The first floor risers must supply 28 8 Mbh. According to the table, 1¼-in. flow and return risers will supply 26.0 Mbh, if the return riser is increased to 1½ in., the capacity will be increased to 34.0 Mbh This is considerably larger than necessary, and 1¼-in flow and return risers are selected. However, it must be remembered that the riser branches, which are the connections from the flow and return mains to the flow and return risers, are to be one size larger than the risers.

The second floor risers must supply 19.2 Mbh. According to the table, the capacity of 1-in flow and return risers is 20.0 Mbh, and that size is selected.

The third floor risers must supply 9 6 Mbh. If a ½-in. flow and a ¾-in. return riser are used, the capacity will be 8 0 Mbh, if both risers are ¾ in., the capacity will be 14.0 Mbh. The ¾-in. pipe is selected for both risers

To design the radiator connections, use Table 8 and note that for the first floor radiator connections the capacity of a ¾-in. flow and 1-in. return is 9.1 Mbh, and that of a 1-in. flow and a 1-in return is 12.5 Mbh The former is more nearly the correct size, but since it is difficult to secure a good flow through first floor radiators, the 1-in. flow and return connection is selected. For the two upper floors, the capacity of a ¾-in. flow and return connection is 10 5 Mbh, and that size is used.

As explained in the design of the forced circulation system of Fig. 5, the two-pipe direct return system of Fig. 7 will not function correctly unless its four sets of risers are balanced among themselves. This necessary balancing is accomplished by adding resistances to all risers, except the one farthest from the boiler, equal to the excess boiler pressure heads available for those risers above the boiler pressure head available for the farthest riser. For example, the first set of risers is 60 ft nearer the boiler than the last set. Since the flow and return mains are designed for a friction head of 3 milinches per foot (See Table 6, Column 8), the boiler pressure head available for the first set of risers is 360 milinches in excess of that available for the fourth set. The velocity in the riser branch is 3 in. per second (See Table 7) and, therefore, according to Table 5, an 0.65-in. orifice in a 1¼-in. union should be used. This will provide a resistance of about 420 milinches. In the same manner it is found that for the second set of risers a resistance of 240 milinches is required and that an 0.70-in. orifice in a 1¼-in. union will provide a resistance of 285 milinches. For the third set of risers, a resistance of 120 milinches is required and an 0.60-in. orifice in a 1-in. union will provide sufficient resistance.

EXPANSION TANKS

When water at ordinary temperatures is heated or cooled, its volume is increased or decreased. This variation in the volume of the water in a heating system is generally provided for by means of an expansion tank into which the water can flow from the system during the heating-up periods and from which it can flow back into the system during the cooling-down periods.

The expansion tank may be open or closed. In an open expansion tank (Fig. 9), the water is subjected to atmospheric pressure and can expand freely without a material increase in pressure. In a closed expansion

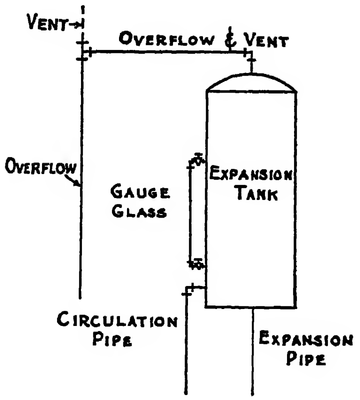


Fig. 9. An Open Expansion Tank

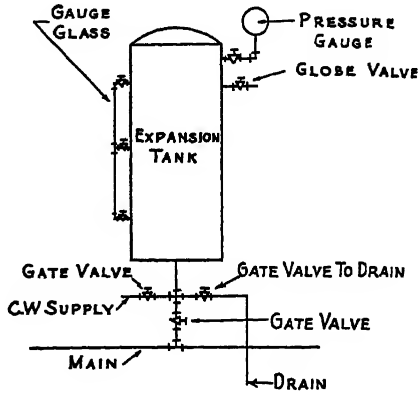


Fig 10. A Closed Expansion Tank

tank (Fig. 10), the water is subjected to the pressure of the compressed air within the tank, and as the water expands, the volume of the air in the tank is decreased and its pressure increased.

The open expansion tank must be placed at a sufficient elevation above the highest radiator to prevent boiling when the water in that radiator is at the highest temperature to which it is to be heated. For example, if the water is to be heated to 225 F on extremely cold days, the absolute pressure on the water in the highest radiator must be at least 19 lb per square inch. This pressure will be secured if the open expansion tank is located 15 ft above the highest radiator. If a closed expansion tank is used and is located 30 ft below the highest radiator, an absolute pressure of about 32 lb per square inch must be maintained in the expansion tank if the water in the highest radiator is to be heated to 225 F without danger of boiling.

The type of expansion tank used in a heating system, whether open or closed, has no influence on the operation of the system. The only function performed by the expansion tank is to provide for the variation in the

Table 9. Expansion Tank Sizes for Hot Water Heating Systems

Tank Size Gallons	Equivalent Direct Radiation Installed in Sq Ft	Capacity Direct Radiation Installed in Mbh
18	Up to 350	Up to 52 5
21	Up to 450	Up to 67.5
24	Up to 650	Up to 97.5
30	Up to 900	Up to 135.0
35	Up to 1100	Up to 165.0
40	Up to 1400	Up to 210.0
2–30	Up to 1600	Up to 240.0
2–30	Up to 1800	Up to 270.0
2–35	Up to 2000	Up to 300.0
2–40	Up to 2400	Up to 360.0

volume of the water in the system, and at the same time to maintain a sufficient pressure in the system to prevent boiling when the water is at the highest temperature for which the system is designed. The capacity of the cushion or expansion tank should not be less than the tank sizes indicated in Table 9 and in addition provisions must be made for draining it without emptying the system.

The capacity of the expansion tank should be at least twice the increase in volume produced when the water in the system is heated from its normal to its maximum temperature. When 25 gal of water are heated from 40 F to 200 F, the volume of water increases to 26 gal. A safe rule, therefore, is to make the water capacity of the expansion tank equal to 10 per cent of the capacity of the heating system.

In a forced circulation system, the expansion tank can either be connected to the flow or return main. In a gravity circulation system, the expansion tank should be connected to the flow riser so that air liberated from the water in the boiler may escape through the expansion tank.

The expansion tank should be protected so that the water in the tank or in the connecting pipe lines cannot freeze. If such water should freeze and the water in the system be heated to cause further expansion, the resulting force will burst the boiler or some other portion of the system.

RELIEF VALVES

All closed hot water heating systems should be equipped with a relief valve and compression tank, which should be of adequate size to relieve any pressure over the pressure rating of the boiler. Protection is thus afforded against any possible damages to the boiler due to overheating.

INSTALLATION DETAILS

The detailed installation of the pipe system should be governed by four fundamental rules:

1. All piping must be pitched either up or down so that all gases which are liberated from the water can move freely to a vented section of the system. Whenever practicable, the pipe line should be pitched so that gases flowing to a vent will flow in the same direction as the water. When a pipe system cannot be installed without creating *air pockets*, that is, sections in the system from which liberated gases cannot escape, such sections must be provided with automatic air relief valves or with air valves which may be operated manually when necessary, or trapped into a pressure tank.

2. All piping must be arranged so that the entire system can be drained, either to permit alterations or repairs, or to prevent freezing if the system is not to be operated during a cold period.

It is well to install a gate valve and union in every riser near the main to permit the draining of individual risers without draining the entire system. It is also well, in large installations, to divide the system into branches and to provide each branch with unions and valves so that any one branch can be drained without disturbing the remaining ones.

The dividing of large heating systems into branches or zones and providing each zone with individual valves has the further advantage of permitting a varying temperature control. For example, if a building is equipped with a forced circulating system and if the south rooms are on one branch of the main and the north rooms are on a separate branch, the valves may be set so that the water will circulate through the north branch with a temperature drop of, say, 10 F, and through the south branch with a temperature drop of, say, 20 F, thus delivering less heat to the south rooms than to the north

rooms. This arrangement is especially valuable when the regulating valves are controlled thermostatically by the temperatures in the two zones, because no matter how accurately the heating system may have been designed, the heat demand of any group of rooms varies with sunshine and with wind velocity, and these intermittent variations can be provided for only by the individual control made possible by changing the valve settings controlling the heat supplied to particular groups of rooms.

3 All piping must be installed so that it is free to expand and contract with changes of temperature without producing undue stresses in the pipes or connections For this purpose it is generally sufficient to allow for a variation in length of 1 in. for 100 ft of pipe.

4. The pipe system must be installed so that each circuit has its correct friction head To bring this about, it is necessary in some cases to minimize the friction, *i.e.*, to make the pipe line as short as possible and to provide as few fittings as possible, and in other cases it is necessary to increase the length of the pipe and the number of fittings so that, for every circuit, the friction head will be equal to the available pressure head.

The connections from the boiler to the mains should be short and direct, to reduce the friction head It is frequently possible to avoid an elbow and to reduce the length of the

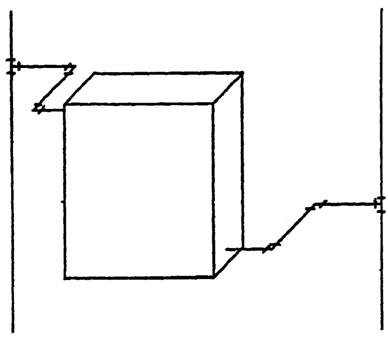


FIG. 11. METHOD OF CONNECTING RADIATOR TO ALLOW FOR EXPANSION OF PIPE

pipe by running the pipe in a diagonal direction, either in a horizontal or in a vertical plane.

The mains and branches should pitch up and away from the heater, generally not less than 1 in. in 10 ft. The flow main should always be covered; the return main should be covered except where it is to provide the heating surface for the basement.

The connections from mains to branches and to risers should be such that circulation through the risers will start in the right direction. Hence, in a one-pipe system the flow connection must be nearer the heater than the return connection. In a correctly-designed two-pipe system, the pressure in the flow main is higher than that in the return main, and a slight variation in the distances of the flow and return connections from the heater is not material, but it is generally best to have the two connections about equally distant from the heater.

In some cases it may be advisable to take the flow connection off the top of the main and the return connection from the side, but in most cases both connections should be at an angle of 45 deg. This method shortens the lines and substitutes 45-deg ells for 90-deg ells.

Preferably, connection of the flow riser to a radiator should be to the upper tapping, and connection of the return riser to a radiator should be to the lower tapping. When hot water enters at the top of a radiator it will distribute itself along the entire length of the radiator, and as it cools it will settle gradually to the bottom; the cool water may then be taken out of the radiator at either end.

With forced circulation and high velocities, it is advisable to let the water enter at the top of the radiator and leave at the bottom of the opposite end. With gravity circulation and low velocities it makes little difference whether the water leaves at the end at which it enters or at the opposite end.

The connections of the risers to the radiators should be such that provision is made for the vertical expansion of the risers. This can be accomplished as indicated in Fig. 11 by using one tee and two ells for each connection. These connections should be pitched upward or downward, whichever may be necessary to prevent the formation of air pockets and to permit draining.

PROBLEMS IN PRACTICE

1 ● Will altering a hot water heating system from an open to closed type system (*a*) increase the circulation and (*b*) give more heat?

a. No. Tests conducted by the A.S.H.V.E. indicate that there is little if any difference in the circulation when the system is under pressure. The difference in temperature between the supply and return, and the friction are the governing factors.

b. With a closed system the water may be carried at a higher temperature without boiling which permits warmer radiators.

2 ● What tends to prevent or to retard the circulation of water in hot water heating systems?

In both gravity flow and forced circulation systems, the friction which must be overcome when the water is flowing through pipes, fittings, valves, heaters, and radiators tends to prevent or retard circulation. For a given pipe the friction varies approximately as the 1.7 power of the velocity, and for given fittings, valves, heaters, and radiators, the friction varies approximately as the square of the velocity. It is therefore sufficiently accurate to express the friction in fittings, valves, heaters, and radiators in terms of the friction in one standard elbow, as shown in Table 1.

3 ● In the elementary heating system, Fig. 8, what is the pressure head maintaining the circulation if the water in the return riser is at 180 F and that in the flow riser is at 200 F?

It is found, from Table 7, Chapter 1, that 180 F water weighs 60.61 lb per cubic foot and 200 F water weighs 60.13 lb per cubic foot. The pressure head is independent of the size of the pipe. If the two risers were each 1 ft square, the water in the flow riser would weigh 601.3 lb and that in the return riser would weigh 606.1 lb. Thus the water in the return riser would weigh 4.8 lb more than that in the flow riser. Consequently, the resulting pressure head is 4.8 lb per square foot.

Pressure heads are generally expressed in feet, or inches, or milinches of water of a given temperature. In this case we are dealing with water at both 180 F and 200 F, so the pressure head is expressed in terms of 190 F water. Such water weighs 60.39 lb per cubic foot, and to secure a pressure of 4.8 lb per square foot, it is necessary to have a column of water having a weight of 4.8 divided by 60.39 = 0.0795 ft, or 0.9540 in., or 954 milinches. This is the pressure head which maintains the circulation.

4 ● In the elementary system of Question 3, if the radiator dissipates 14,000 Btu per hour, what is the velocity of the water in the pipe line, if the pipes are 1 in. in diameter? What, if they are ¾ in. in diameter?

Since the temperature drop through the radiator is from 200 F to 180 F or 20 F, every pound of water flowing through the radiators delivers 20 Btu; consequently, 14,000 divided by 20 = 700 lb of water, or for 190 F water, 700 divided by 60.39 = 11.59 cu ft of water must flow through the radiator and through the pipe lines every hour.

The interior area of a 1-in. pipe is 0.864 sq in. The velocity in the 1-in. pipe is 11.59 divided by 0.864 and multiplied by 144 = 1932 ft per hour or 6.44 in. per second.

For ¾-in. pipe, the interior area is 0.533, and the velocity is 6.44 multiplied by 0.864 and divided by 533 = 10.44 in. per second.

5 ● If, in the elementary heating system of Question 3, a 1-in. pipe line is used, what would be the friction head?

If the radiator is connected as shown in Fig. 11, with the heater connected to provide

freedom of expansion, the heating circuit may be assumed to consist of a heater, 25 ft of pipe, 8 elbows, 1 radiator valve, and 1 radiator. From Table 1 it appears that the heater and radiator are equivalent, in friction, to 6 elbows, hence, the circuit may be placed equal to 25 ft of pipe and 14 elbows.

From the diagram of Fig. 4 it appears that the friction head for a 1-in pipe and a velocity of 6 44 in per second is about 25 milinches per foot. For 25 ft of pipe, the friction head will be 625 milinches.

It appears from Table 1 that the friction head in one elbow is $\frac{v^2}{2g}$, or in this case 0 54 multiplied by 0 54 and divided by 64.4 = 0 0045 ft or 54 milinches. Hence, for the 14 elbows the friction is 756 milinches For the entire circuit, the friction head is the sum of the 625 milinches of the pipe plus the 756 milinches of the elbows, or 1381 milinches which equal 1 381 in.

6 ● If the elementary heating system of Question 3 is installed with a 1-in. pipe line, how will it function?

It is found from the answer to Question 3 that the pressure head is 954 milinches and from the answer to Question 5 that the friction head is 1381 milinches when the water is flowing with such velocity that the specified 14,000 Btu will be delivered with a 20 F temperature drop through the radiators. Since the pressure head is smaller than the friction head, the system will not function as planned for the water will flow through the system more slowly and remain in the radiator longer. The temperature drop through the radiator will be more than 20 F, and the difference in the weight of the water in the return and flow risers will be greater than that intended. The final result will be that the pressure head will become equal to the friction head at a value somewhere between 954 and 1381 milinches Since the average water temperature in the radiator will be less than 190 F, the radiator should be larger than the size given in Question 4.

7 ● Should a hot water heating system be designed to embody small pipes or large pipes?

As pipe sizes in gravity circulation heating are reduced, the friction head is increased and it is necessary to increase the temperature drop through radiators, this lowers the average temperature of the water in the radiators and necessitates an increase in the size of the radiators, so whereas the cost of the pipe in a system is reduced, the cost of the radiators is increased. For each installation there is a definite pipe size which entails maximum economy.

As pipe sizes in forced circulation systems are reduced, friction heads are increased so a circulating pump of greater size or capacity is required. Thus, by decreasing the size of the piping, both the first cost of the circulating pump and the cost of its operation are increased There is a definite pipe size for every installation which is most economical. For each installation of both types of systems there is a definite pipe size entailing maximum economy which can be determined by a series of comparative calculations.

8 ● What should be the size of the radiators for the elementary heating system of Question 3 in which the water enters the radiator with a temperature of 200 F and leaves with a temperature of 180 F? The average temperature of the water in the radiator is, approximately, 190 F.

If test results are available for the particular radiators to be used, and for the temperatures named, the size of the radiators should be selected from them. If no such test results are to be had, but if test results are available for the type of radiator to be used when it is supplied with 215 F steam and placed in a 70 F room, the required size may be determined by the following ratio. The required size is to the corresponding steam radiator size as $(215 - 70)^{1.3}$ is to $(190 - 70)^{1.3}$. This ratio works out to 1.28. Hence, the radiators should be 28 per cent larger under the conditions prescribed than are corresponding radiators under standard conditions. It is immaterial whether a radiator is filled with steam or with water, as long as the average temperature of its outer surface is the same in both cases.

Chapter 34

PIPE, FITTINGS, WELDING

Pipe Material, Types of Pipe Used, Dimensions of Pipe Commercially Available, Expansion and Flexibility of Pipe, Pipe Threads and Hangers, Types of Fittings, Welding as Applied to Erection of Piping, Valves, Corrosion of Piping

IMPORTANT considerations in the selection and installation of pipe and fittings for heating, ventilating, and air conditioning work are dealt with in this chapter.

PIPE MATERIALS

Use of corrosion-resistant materials for pipe, including special alloy steels and irons, wrought-iron, copper and brass, has increased considerably during the past few years. The recent development of copper, brass, and bronze fittings which can be assembled by soldering or sweating permits the use of thin-wall pipe and thereby has reduced the initial cost of such installation. The following brief discussion indicates the variety of pipe materials and the types of pipe available.

Wrought-Steel Pipe. Because of its low price, the great bulk of wrought pipe used for heating and ventilating work at the present time is of wrought steel. The material used for steel pipe is a mild steel made by the acid-bessemer, the open-hearth, or the electric-furnace process. Ordinary wrought-steel pipe is made either by shaping sheets of metal into cylindrical form and welding the edges together, or by forming or drawing from a solid billet. The former is known as *welded pipe*, the latter as *seamless pipe.*

Many types of welded pipe are available, although the smaller sizes most frequently used in heating and ventilating work are made by the lap-weld or butt-weld process. While the lap-weld process produces a better weld than the butt type, lap-weld pipe is seldom manufactured in nominal pipe sizes less than 2 in. Seamless pipe can be obtained in the small sizes at a somewhat higher cost.

Seamless steel pipe is frequently used for high pressure work or where pipe is desired for close coiling, cold bending, or other forming operation. Its advantages are its somewhat greater strength which permits use of a thinner wall and, in the small sizes, its freedom from the occasional tendency of welded pipe to split at the weld when bent.

Wrought-Iron Pipe. Wrought-iron pipe is considered to be more corrosion-resisting than ordinary steel pipe and therefore its somewhat higher

first cost can be justified on the basis of longer life expectancy. Wrought-iron pipe may be identified by the spiral line marked into each length, either knurled into the metal or painted on it in red or other bright color. Otherwise, there is little difference in the appearance of wrought-iron and steel pipe, although microscopic examination of polished and etched specimens will readily disclose the difference.

Cast Ferrous Pipe. There are now available several types of cast ferrous-metal pipe made of a good grade of cast-iron with or without additions of nickel, chromium, or other alloy. This pipe is available in sizes from 1½ in. to 6 in., and in standard lengths of 5 or 6 ft with external and internal diameters closely approximating those of extra-strong wrought pipe. Cast ferrous pipe may be obtained coupled, beveled for welding, or with ends plain or grooved for the several types of couplings It is easily cut and threaded as well as welded. The fact that it is readily welded enables the manufacturers to supply the pipe in any lengths practicable for handling.

Alloy Metal Pipe. Steel pipe bearing a small alloy of copper or other alloying element and iron pipe bearing a small alloy of copper and molybdenum have been claimed to possess more resistance to corrosion than plain steel pipe and they are advertised and sold under various trade names.

Copper Pipe and Fittings. Owing to its inherent resistance to corrosion, copper and brass pipe have always been used in heating, ventilating, and water supply installations, but the cost with standard dimensions for threaded connections has been high. The recent introduction of fittings which permit erection by soldering or sweating allows the use of pipe with thinner walls than are possible with threaded connections, thereby reducing the cost of installations.

The initial cost of brass and copper pipe installations generally runs higher than the corresponding job with steel pipe and screwed connections in spite of the use of thin-wall pipe, but the corrosive nature of the fluid conveyed or the inaccessibility of some of the piping may warrant use of a more expensive material than plain steel. The advantages of corrosion-resisting pipe and fittings should be weighed against the correspondingly higher initial cost.

COMMERCIAL PIPE DIMENSIONS

The *IPS* dimensions of commercial pipe universally used at the present time conform to the recommendations made by a Committee of the *A.S.M.E.* in 1886. Pipe up to 12 in. in diameter is made in certain definite sizes designated by nominal internal diameter which is somewhat different from the actual internal diameter, depending on the wall thickness required. There are three weights of wrought-iron and steel pipe commonly used, known as *standard-weight*, *extra-strong*, and *double extra-strong*. Because of the necessity of maintaining the same external diameter in all three weights for the same nominal size, the added wall thickness is obtained by decreasing the internal diameter. The term *full-weight*, when applied to sizes below 8 in., means that the pipe is up to the nominal weight per foot. When applied to sizes between 8 and 12 in., inclusive, it often indicates that the pipe has the heaviest of several wall

thicknesses listed. In sizes 14 in. and upward, pipe is designated by its outside diameter (O.D.) and the wall thickness is specified.

While the demands for pipe for the heating and ventilating industry are reasonably well served by the *standard-weight* and *extra-strong* pipe, demands for pipe for higher pressures and temperatures in industry resulted in the use of a multiplicity of wall thicknesses for all sizes. Even in heating installations, the erection of piping by welding was deemed to

TABLE 1 DIMENSIONS OF WELDED AND SEAMLESS STEEL PIPE

Nominal Pipe Size	Outside Diam	Nominal Wall Thicknesses for Schedule Numbers									
		Schedule 10	Schedule 20	Schedule 30	Schedule 40	Schedule 60	Schedule 80	Schedule 100	Schedule 120	Schedule 140	Schedule 160
1/8	0.405	—	—	—	0.068*	—	0.095*	—	—	—	—
1/4	0.540	—	—	—	0.088*	—	0.119*	—	—	—	—
3/8	0.675	—	—	—	0.091*	—	0.126*	—	—	—	—
1/2	0.840	—	—	—	0.109*	—	0.147*	—	—	—	0.187
3/4	1.050	—	—	—	0.113*	—	0.154*	—	—	—	0.218
1	1.315	—	—	—	0.133*	—	0.179*	—	—	—	0.250
1 1/4	1.660	—	—	—	0.140*	—	0.191*	—	—	—	0.250
1 1/2	1.900	—	—	—	0.145*	—	0.200*	—	—	—	0.281
2	2.375	—	—	—	0.154*	—	0.218*	—	—	—	0.343
2 1/2	2.875	—	—	—	0.203*	—	0.276*	—	—	—	0.375
3	3.500	—	—	—	0.216*	—	0.300*	—	—	—	0.437
3 1/2	4.000	—	—	—	0.226*	—	0.318*	—	—	—	—
4	4.500	—	—	—	0.237*	—	0.337*	—	0.437	—	0.531
5	5.563	—	—	—	0.258*	—	0.375*	—	0.500	—	0.625
6	6.625	—	—	—	0.280*	—	0.432*	—	0.562	—	0.718
8	8.625	—	0.250	0.277*	0.322*	0.406	0.500*	0.593	0.718	0.812	0.906
10	10.75	—	0.250	0.307*	0.365*	0.500*	0.593	0.718	0.843	1.000	1.125
12	12.75	—	0.250	0.330*	0.406	0.562	0.687	0.843	1.000	1.125	1.312
14 O. D.	14.0	0.250	0.312	0.375	0.437	0.593	0.750	0.937	1.062	1.250	1.406
16 O. D.	16.0	0.250	0.312	0.375	0.500	0.656	0.843	1.031	1.218	1.437	1.562
18 O. D.	18.0	0.250	0.312	0.437	0.562	0.718	0.937	1.156	1.343	1.562	1.750
20 O. D.	20.0	0.250	0.375	0.500	0.593	0.812	1.031	1.250	1.500	1.750	1.937
24 O. D.	24.0	0.250	0.375	0.562	0.687	0.937	1.218	1.500	1.750	2.062	2.312
30 O. D.	30.0	0.312	0.500	0.625	—	—	—	—	—	—	—

All dimensions are given in inches

The decimal thicknesses listed for the respective pipe sizes represent their nominal or average wall dimensions and include an allowance for mill tolerance of 12.5 per cent under nominal thicknesses

*Thicknesses marked with asterisk in Schedules 30 and 40 are identical with thicknesses for *standard-weight* pipe in former lists, those in Schedules 60 and 80 are identical with thicknesses for *extra-strong* pipe in former lists

The Schedule Numbers indicate approximate values of the expression 1000 x P/S.

warrant the use of pipe lighter than standard weight. For these reasons, a *Sectional Committee on Standardization of Wrought Iron and Wrought Steel Pipe and Tubing* functioning under the procedure of the *American Standards Association* was appointed to standardize the dimensions and materials of pipe.

The proposed pipe standard recommended by that sectional committee has set up several schedules of pipe including standard-weight and extra-strong thicknesses which are now included in Schedules 40 and 60, respectively. The schedules approved by the Sectional Committee are given in Tables 1 and 3 and the corresponding weights in Tables 2 and 4.

Standard-weight pipe is generally furnished with threaded ends in

random lengths of 16 to 22 ft, although when ordered with plain ends, 5 per cent may be in lengths of 12 to 16 ft. Five per cent of the total number of lengths ordered may be *jointers* which are two pieces coupled together. Extra-strong pipe is generally furnished with plain ends in random lengths of 12 to 22 ft, although 5 per cent may be in lengths of 6 to 12 ft.

In addition to *IPS* copper pipe, several varieties of copper tubing are in use with either flared or compression couplings or soldered joints. Dimensions of copper water tubing intended for plumbing, underground water service, fuel-oil lines, gas lines, etc., have been standardized by the U. S. Government and the *American Society for Testing Materials.* There are three standard wall-thickness schedules of copper water tubing classified in accordance with their principal uses as follows:

Class *K*—Designed for underground services and general plumbing service.

Class *L*—Designed for general plumbing purposes.

Class *M*—Designed for use with soldered fittings only.

In general, Type *K* is used where corrosion conditions are severe, and

TABLE 2. NOMINAL WEIGHTS OF WELDED AND SEAMLESS STEEL PIPE

Nominal Pipe Size Inches	Sched 10 Plain Ends	Sched 20 Plain Ends	Schedule 30		Schedule 40		Sched 60 Plain Ends	Sched 80 Plain Ends	Sched 100 Plain Ends	Sched 120 Plain Ends	Sched 140 Plain Ends	Sched 160 Plain Ends
			Plain Ends	Threads and Couplings	Plain Ends	Threads and Couplings						
1/8					0.25*	0 25*		0 32*				
1/4					0 43*	0.43*		0 54*				
3/8					0.57*	0.57*		0 74*				
1/2					0.86*	0.86*		1 09*				1.31
3/4					1 14*	1.14*		1.48*				1.94
1					1.68*	1 69*		2.18*				2.85
1 1/4					2.28*	2 29*		3.00*				3.77
1 1/2					2.72*	2 74*		3.64*				4.86
2					3 66*	3 68*		5.03*				7.45
2 1/2					5 80*	5.82*		7.67*				10 0
3					7 58*	7.62*		10.3*				14.3
3 1/2					9.11*	9.21*		12.5*				
4					10 8*	10 9*		15.0*		19.0		22.6
5					14.7*	14.9*		20.8*		27.1		33.0
6					19 0*	19 2*		28.6*		36.4		45.3
8		22.4	24.7*	25.0*	28.6*	28.8*	35.7	43.4*	50.9	60.7	67.8	74.7
10		28.1	34.3*	35.0*	40 5*	41.2*	54.8*	64.4	77.0	89.2	105.0	116.0
12		33.4	43.8*	45.0*	53 6	55.0	73.2	88.6	108.0	126.0	140.0	161 0
14 O. D.	36.8	45.7	54.6		63.3		85.0	107.0	131.0	147.0	171.0	190.0
16 O. D.	42.1	52.3	62.6		82 8		108.0	137.0	165.0	193.0	224.0	241.0
18 O. D.	47.4	59.0	82.0		105 0		133 0	171.0	208.0	239.0	275.0	304.0
20 O. D.	52.8	78.6	105.0		123 0		167.0	209.0	251.0	297.0	342.0	374.0
24 O. D.	63.5	94.7	141.0		171.0		231.0	297.0	361.0	416.0	484 0	536.0
30 O. D.	99.0	158.0	197.0									

Weights are given in pounds per linear foot and are for pipe with plain ends except for sizes which are commercially available with threads and couplings for which both weights are listed.

*The weights marked with asterisk in Schedules 30 and 40 are identical with weights for *standard-weight* pipe in former lists; those in Schedules 60 and 80 are identical with weights for *extra-strong* pipe in former lists

The Schedule Numbers indicate approximate values of the expression 1000 x P/S.

Types *L* and *M* where such conditions may be considered normal as, for instance, in heating work. Types *K* and *L* are available in both hard and soft tempers; Type *M* is available only in hard temper. Where flexibility is essential as in hidden replacement work or where as few joints as possible are desired as in fuel-oil lines, the soft temper is commonly used. New or exposed work generally employs copper pipe of a hard temper. All three classes are extensively used with soldered fittings.

Standard dimensions, weights, and diameter and wall thickness tolerances for these classes of copper tubing are given in Table 5. Copper pipe is also available with dimensions of steel pipe.

Refrigeration lines used in connection with air conditioning equipment also employ copper tubing extensively. For refrigeration use where tubing absolutely free from scale and dirt is required, bright annealed copper tubing that has been deoxidized is used. This tubing is available in a variety of sizes and wall thicknesses.

EXPANSION AND FLEXIBILITY

The increase in temperature of a pipe from room temperature to an operating steam or water temperature 100 F or more above room tem-

Table 3. Dimensions of Welded Wrought-Iron Pipe

Nominal Pipe Size	Outside Diameter	Nominal Wall Thicknesses for Schedule Numbers					
		Schedule 10	Schedule 20	Schedule 30	Schedule 40	Schedule 60	Schedule 80
1/8	0.405	—	—	—	0.070*	—	0.098*
1/4	0.540	—	—	—	0.090*	—	0.122*
3/8	0 675	—	—	—	0.093*	—	0.129*
1/2	0.840	—	—	—	0.111*	—	0.151*
3/4	1.050	—	—	—	0.115*	—	0.157*
1	1.315	—	—	—	0.136*	—	0.183*
1 1/4	1.660	—	—	—	0.143*	—	0.195*
1 1/2	1.900	—	—	—	0.148*	—	0.204*
2	2.375	—	—	—	0.158*	—	0.223*
2 1/2	2.875	—	—	—	0.208*	—	0.282*
3	3.5	—	—	—	0.221*	—	0.306*
3 1/2	4.0	—	—	—	0.231*	—	0.325*
4	4 5	—	—	—	0.242*	—	0.344*
5	5 563	—	—	—	0.263*	—	0.383*
6	6 625	—	—	—	0.286*	—	0.441*
8	8.625	—	—	0.283*	0.329*	—	0.510*
10	10.75	—	—	0.313*	0.372*	0.510*	0.606
12	12.75	—	—	0.336*	0.414	0.574	0.702
14 O. D.	14 0	0.250	0.312	0.375	0.437	0.625	0.750
16 O. D.	16 0	0.250	0.312	0.375	0.500	0.687	—
18 O. D.	18 0	0 250	0.312	0.437	0.562	0.750	—
20 O. D.	20.0	—	0.375	0.500	0.562	—	—

All dimensions are given in inches

The decimal thicknesses listed for the respective pipe sizes represent their nominal or average wall dimensions and include an allowance for mill tolerance of 12 5 per cent under the nominal thickness.

*Thicknesses marked with an asterisk in Schedules 30 and 40 are identical with thicknesses for *standard-weight* pipe in former lists, those in Schedules 60 and 80 are identical with thicknesses for *extra-strong* pipe in former lists.

The Schedule Numbers indicate approximate values of the expression 1000 x P/S.

perature results in an increase in length of the pipe for which provision must be made. The amount of linear expansion (or contraction in the case of refrigeration lines) per unit length of material per degree change in temperature is termed the *coefficient of linear expansion* of that material, or commonly, the *coefficient of expansion*. This coefficient varies with the material.

The linear expansion of cast iron, steel, wrought-iron, and copper pipe,

TABLE 4 NOMINAL WEIGHTS OF WELDED WROUGHT-IRON PIPE

Nominal Pipe Size (Inches)	Sched 10	Sched 20	Schedule 30		Schedule 40		Schedule 60	Schedule 80
	Plain Ends	Plain Ends	Plain Ends	Threads and Couplings	Plain Ends	Threads and Couplings	Plain Ends	Plain Ends
1/8	......	......	......	......	0.25*	0.25*	......	0.32*
1/4	......	......	......	......	0.43*	0.43*	......	0.54*
3/8	......	......	......	......	0.57*	0.57*	......	0.74*
1/2	......	......	......	......	0.86*	0.86*	......	1.09*
3/4	......	......	......	......	1.14*	1.14*	......	1.48*
1	......	......	......	......	1.68*	1.69*	......	2.18*
1 1/4	......	......	......	......	2.28*	2.29*	......	3.00*
1 1/2	......	......	......	......	2.72*	2.74*	......	3.64*
2	......	......	......	......	3.66*	3.68*	......	5.03*
2 1/2	......	......	......	......	5.80*	5.82*	......	7.67*
3	......	......	......	......	7.58*	7.62*	......	10.3*
3 1/2	......	......	......	......	9.11*	9.21*	......	12.5*
4	......	......	......	......	10.8*	10.9*	......	15.0*
5	......	......	......	......	14.7*	14.9*	......	20.8*
6	......	......	......	......	19.0*	19.2*	......	28.6*
8	......	......	24.7*	25.0*	28.6*	28.8*	......	43.4*
10	......	......	34.3*	35.0*	40.5*	41.2*	54.8*	54.4
12	......	......	43.8*	45.0*	53.6	55.0	73.2	88.6
14 O. D.	36.0	44.8	53.6	......	62.2	......	87.6	104.0
16 O. D.	41.3	51.4	61.4	......	81.2	......	111.0	......
18 O. D.	46.5	57.9	80.5	......	103.0	......	136.0	......
20 O. D.	......	77.0	103.0	......	115.0	......	......	......

Weights are given in pounds per linear foot and are for pipe with plain ends except for sizes which are commercially available with threads and couplings for which both weights are listed.

*Weights marked with an asterisk in Schedules 30 and 40 are identical with weights for *standard-weight* pipe in former lists; those in Schedules 60 and 80 are identical with weights for *extra-strong* pipe in former lists.

The Schedule Numbers indicate approximate values of the expression 1000 x P/S.

the materials most frequently used in heating and ventilating work, can be determined from Table 6.

The elongation values in Table 6 were computed from the following formula:

$$L_t = L_0 \left[1 + a \left(\frac{t - 32}{1000} \right) + b \left(\frac{t - 32}{1000} \right)^2 \right] \qquad (1)$$

where

L_t = length at temperature t degrees Fahrenheit, feet.

L_0 = length at 32 F, feet.

t = final temperature, degrees Fahrenheit.

a and b are constants as follows:

METAL	a	b
Cast-Iron	0.005441	0.001747
Steel	0.006212	0.001623
Wrought-Iron	0.006503	0.001622
Copper	0.009278	0.001244

The three methods by which the elongation due to thermal expansion may be taken care of are:

1. Expansion joints.
2. Swivel joints.
3. Inherent flexibility of the pipe itself utilized through pipe bends, right-angle turns, or offsets in the line.

TABLE 5 STANDARD DIMENSIONS, WEIGHTS, AND DIAMETER AND WALL THICKNESS TOLERANCES FOR COPPER WATER TUBES*

(All Tolerances Plus and Minus)

NOMINAL SIZE, IN	ACTUAL OUTSIDE DIAMETER, IN	PERMISSIBLE VARIATION IN MEAN OUTSIDE DIAMETER, IN		WALL THICKNESS, IN						WEIGHT PER FT LB		
				Class *K*		Class *L*		Class *M*				
		Annealed	Hard Drawn	Nominal	Permissible Variation	Nominal	Permissible Variation	Nominal	Permissible Variation	Class *K*	Class *L*	Class *M*
3/8	0.500	0.0025	0.001	0.049	0.004	0.035	0.0035	0.025	0.0025	0.269	0.198	0.144
1/2	0.625	0.0025	0.001	0.049	0.004	0.040	0.0035	0.028	0.0025	0.344	0.285	0.203
3/4	0.875	0.003	0.001	0.065	0.0045	0.045	0.004	0.032	0.003	0.641	0.455	0.328
1	1.125	0.0035	0.0015	0.065	0.0045	0.050	0.004	0.035	0.0035	0.839	0.655	0.464
1 1/4	1.375	0.004	0.0015	0.065	0.0045	0.055	0.0045	0.042	0.0035	1.04	0.884	0.681
1 1/2	1.625	0.0045	0.002	0.072	0.005	0.060	0.0045	0.049	0.004	1.36	1.14	0.94
2	2.125	0.005	0.002	0.083	0.005	0.070	0.005	0.058	0.0045	2.06	1.75	1.46
2 1/2	2.625	0.005	0.002	0.095	0.005	0.080	0.005	0.065	0.0045	2.92	2.48	2.03
3	3.125	0.005	0.002	0.109	0.005	0.090	0.005	0.072	0.0045	4.00	3.33	2.68
3 1/2	3.625	0.005	0.002	0.120	0.005	0.100	0.005	0.083	0.005	5.12	4.29	3.58
4	4.125	0.005	0.002	0.134	0.006	0.110	0.005	0.095	0.005	6.51	5.38	4.66
5	5.125	0.005	0.002	0.160	0.006	0.125	0.006	0.109	0.005	9.67	7.61	6.65
6	6.125	0.005	0.002	0.192	0.006	0.140	0.006	0.122	0.005	13.87	10.20	8.91

*From Standard Specifications for Copper Water Tube of the *American Society for Testing Materials*, *A.S.T.M.* Designation B88-33

Expansion joints of the slip-sleeve, diaphragm, or corrugated types made of copper, rubber, or other gasket material are all used for taking up expansion, but generally only for low pressures or where the inherent flexibility of the pipe cannot readily be used as in underground steam or hot water distribution lines.

Swivel joints are used extensively in low-pressure steam and hot water heating systems and in hot water supply lines. The swivel joints absorb the expansive movement of the pipe by the turning of threaded joints. In many cases the straight pipe in the offset of a swivel joint is sufficiently flexible to take up the expansion without developing enough thrust to produce swiveling in the threaded joint. This is preferable since con-

tinued turning in the threaded joint may in time result in a leak, particularly when the pressure is high. The amount of elongation which a swivel joint can take up is controlled by the length of the swing piece employed and by the lateral displacement which is permissible in the long pipe runs.

Probably the most economical method of providing for expansion of piping in a long run is to take advantage of the directional changes which must necessarily occur in the piping and proportion the offsets so that sufficient flexibility is secured. Ninety-degree bends with long, straight tangents in either a horizontal or a vertical plane are an excellent means for securing adequate flexibility with larger sizes of pipe. When flexibility cannot be obtained in this manner, it is necessary to make use of some type of expansion bend. The exact calculation of the size of expansion bends required to take up a given amount of thermal expansion

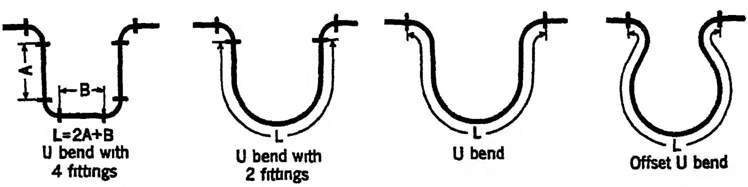


Fig. 1 Measurement of L on Various Pipe Bends

is relatively complicated[1]. The following approximate method, however, has been found to give reasonably good results and is deemed to be sufficiently accurate for most heating work.

Fig. 1 shows several types of expansion bends commonly used for taking up thermal expansion. The amount of pipe, L, required in each of these bends may be computed from the following formula.

$$L = 6.16 \sqrt{D \Delta} \tag{2}$$

where

L = length of pipe, feet.
D = outside diameter of the pipe used, inches.
Δ = the amount of expansion to be taken up, inches

This formula, based on the use of mild-steel pipe with wall thicknesses not heavier than extra-strong, assumes a maximum safe value of fiber stress of 16,000 lb per square inch. When square type bends are used, the width of the bend should not exceed about two times the height. It is further assumed that the corners are made with screwed or flanged elbows or with arcs of circles having radii five to six times the pipe diameter.

All risers must be anchored and safeguarded so that the difference in

[1]Piping Handbook, by Walker and Crocker, and A Manual for the Design of Piping for Flexibility by the Use of Graphs, by E A. Wert, S Smith, and E. T. Cope, published by *The Detroit Edison Company*.

length when hot from the length when cold shall not disarrange the normal and orderly provisions for drainage of the branches.

It is especially necessary with light-weight radiators so to anchor the piping and so to give it freedom for expansion that no strain therefrom shall be allowed to distort the radiators. When expansion strains from the pipes are permitted to reach these light metal heaters they usually emit sounds of distress which are exceedingly troublesome.

TABLE 6. THERMAL EXPANSION OF PIPE IN INCHES PER 100 FT[a]

(For superheated steam and other fluids refer to temperature column)

SATURATED STEAM			ELONGATION IN INCHES PER 100 FT FROM −20 F UP				SATURATED STEAM		ELONGATION IN INCHES PER 100 FT FROM −20 F UP			
Vacuum Inches of Hg	Pressure Pounds per Square Inch Gage	Temperature Degrees Fahrenheit	Cast-Iron Pipe	Steel Pipe	Wrought Iron Pipe	Copper Pipe	Pressure Pounds per Square Inch Gage	Temperature Degrees Fahrenheit	Cast-Iron Pipe	Steel Pipe	Wrought Iron Pipe	Copper Pipe
		−20	0	0	0	0	664.3	500	3.847	4.296	4.477	6.110
		0	0.127	0.145	0.152	0.204	795.3	520	4.020	4.487	4.677	6.352
		20	0.255	0.293	0.306	0.442	945.3	540	4.190	4.670	4.866	6.614
		40	0.390	0.430	0.465	0.655	1115.3	560	4.365	4.860	5.057	6.850
29.39	....	60	0.518	0.593	0.620	0.888	1308.3	580	4.541	5.051	5.268	7.123
28.89	....	80	0.649	0.725	0.780	1.100	1525.3	600	4.725	5.247	5.455	7.388
27.99	....	100	0.787	0.898	0.939	1.338	1768.3	620	4.896	5.437	5.660	7.636
26.48	....	120	0.926	1.055	1.110	1.570	2041.3	640	5.082	5.627	5.850	7.893
24.04	....	140	1.051	1.209	1.265	1.794	2346.3	660	5.260	5.831	6.067	8.153
20.27	....	160	1.200	1.368	1.427	2.008	2705	680	5.442	6.020	6.260	8.400
14.63	....	180	1.345	1.528	1.597	2.255	3080	700	5.629	6.229	6.481	8.676
6.45	....	200	1.495	1.691	1.778	2.500		720	5.808	6.425	6.673	8.912
	2.5	220	1.634	1.852	1.936	2.720		740	6.006	6.635	6.899	9.203
	10.3	240	1.780	2.020	2.110	2.960		760	6.200	6.833	7.100	9.460
	20.7	260	1.931	2.183	2.279	3.189		780	6.389	7.046	7.314	9.736
	34.5	280	2.085	2.350	2.465	3.422		800	6.587	7.250	7.508	9.992
	52.3	300	2.233	2.519	2.630	3.665		820	6.779	7.464	7.757	10.272
	74.9	320	2.395	2.690	2.800	3.900		840	6.970	7.662	7.952	10.512
	103.3	340	2.543	2.862	2.988	4.145		860	7.176	7.888	8.195	10.814
	138.3	360	2.700	3.029	3.175	4.380		880	7.375	8.098	8.400	11.175
	180.9	380	2.859	3.211	3.350	4.628		900	7.579	8.313	8.639	11.360
	232.4	400	3.008	3.375	3.521	4.870		920	7.795	8.545	8.867	11.625
	293.7	420	3.182	3.566	3.720	5.118		940	7.989	8.755	9.089	11.911
	366.1	440	3.345	3.740	3.900	5.358		960	8.200	8.975	9.300	12.180
	451.3	460	3.511	3.929	4.096	5.612		980	8.406	9.196	9.547	12.473
	550.3	480	3.683	4.100	4.280	5.855		1000	8.617	9.421	9.776	12.747

[a]From *Piping Handbook*, by Walker and Crocker. This table gives the expansion from − 20 F to the temperature in question. To obtain the amount of expansion between any two temperatures take the difference between the figures in the table for those temperatures. For example, if a steel pipe is installed at a temperature of 60 F and is to operate at 300 F, the expansion would be 2.519 − 0.593 = 1.926 in.

PIPE THREADS

All threaded pipe for heating and ventilating installations uses the American Standard taper pipe thread which is made with a taper of 1 in 16 measured on the diameter of the pipe so as to secure a tight joint. Threads of fittings are tapped to the same taper. The number of threads per inch varies with the different pipe sizes. All threaded pipe should be made up with a thread paste suitable for the service under which the pipe is to be used.

HANGERS AND SUPPORTS

Heating system piping requires careful and substantial support. Where changes in temperature of the line are not large, such simple methods of support may be utilized as hanging the line by means of rods or perforated strip from the building structure, or supporting it by brackets or on piers.

When fluids are conveyed at temperatures of 150 F or above, however, hangers or supporting equipment must be fabricated and assembled to permit free expansion or contraction of the piping. This can be accomplished by the use of long rod hangers, spring hangers, chains, hangers or supports fitted with rollers, machined blocks, elliptical or circular rings of larger diameter than the pipe giving contact only at the bottom, or trolley hangers. In all cases, allowance should be made for rod clearance to permit swinging without setting up severe bending action in the rods.

For pipes of small size, perforated metal strip is often used. For horizontal mains, the rod or strip usually is attached to the joists or steel work of the floor above. For long runs of vertical pipe subject to considerable thermal expansion, either the hangers should be designed to prevent excessive load on the bottom support when expansion takes place, or the bottom support should be designed to withstand the entire load.

TYPES OF FITTINGS

Fittings for joining the separate lengths of pipe together are made in a variety of forms, and are either screwed or flanged, the former being generally used for the smaller sizes of pipe up to and including 3½ in., and the latter for the larger sizes, 4 in. and above. Screwed fittings of large size as well as flanged fittings of small size are also made and are used for certain classes of work at the proper pressure.

The material used for fittings is generally cast-iron, but in addition to this malleable iron, steel and steel alloys are also used, as well as various grades of brass or bronze. The material to be used depends on the character of the service and the pressure.

As in the case of pipe, there are several weights of fittings manufactured. Recognized American Standards for the various weights are as follow:

Cast-iron pipe flanges and flanged fittings for 25 lb (sizes 4 in and larger), 125 lb, and 250 lb maximum saturated steam pressure.

Malleable iron screwed fittings for 150 lb maximum saturated steam pressure.

Cast-iron screwed fittings for 125 and 250 lb maximum saturated steam pressure.

Steel flanged fittings for 150 and 300 lb maximum steam service pressure.

The allowable cold water working pressures for these standards vary from 43 lb for the 25 lb standard to 500 lb for the 300 lb steel standard.

Screwed fittings include: nipples or short pieces of pipe of varying lengths; couplings, usually of wrought iron only; elbows for turning angles of either 45 deg or 90 deg; return bends, which may be of either the close or open pattern, and may be cast with either a back or side outlet; tees; crosses; laterals or *Y* branches; and a variety of plugs, bushings, caps, lock-nuts, flanges and reducing fittings. Reducing fittings as well as bushings, both of which are used in changing from one pipe size to another,

may have the smaller connection tapped eccentrically to permit free drainage of the water of condensation in steam lines or free escape of air in water lines.

Fittings for copper tubing are available in the soldered, flared. or compression types. Illustrations of each of these types is shown in Fig. 2. Fittings for copper pipe of *IPS* dimensions are available in screwed or soldered types of connection.

The compression type fitting is generally limited to smaller size tubing while the flared and soldered types are used in large and small sizes. While no effort has been made to standardize dimensions of flared tube fittings, manufacturers have quite generally used *S.A.E.* standard dimensions. Flared tube fittings are widely used in refrigeration work and the use of *S.A.E.* dimensions and a 45 deg flare renders most fittings

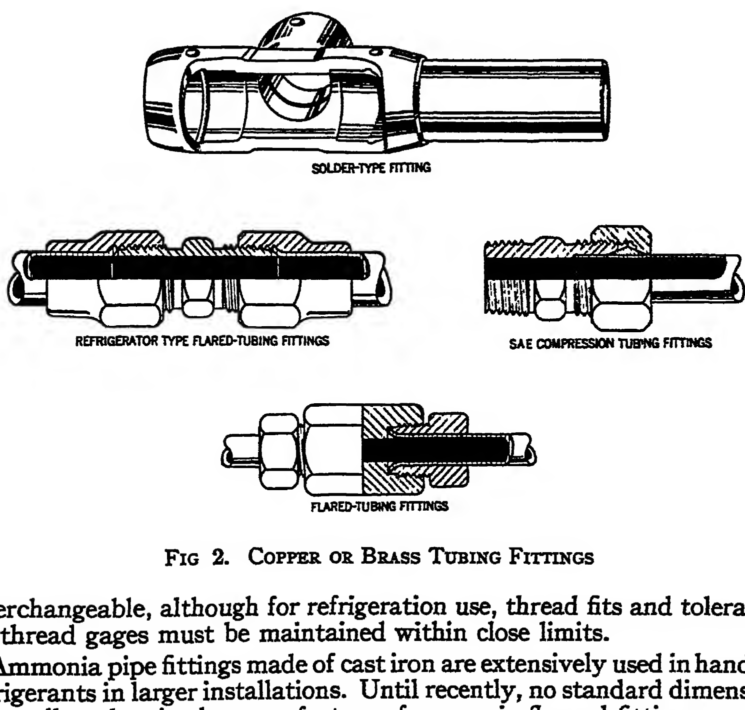


FIG 2. COPPER OR BRASS TUBING FITTINGS

interchangeable, although for refrigeration use, thread fits and tolerances on thread gages must be maintained within close limits.

Ammonia pipe fittings made of cast iron are extensively used in handling refrigerants in larger installations. Until recently, no standard dimensions were adhered to in the manufacture of ammonia flanged fittings or companion flanges with the result that fittings of different manufacturers were not interchangeable. A subcommittee of *A.S.A* Sectional Committee B16 has prepared proposed American Standard dimensions for ammonia flanged fittings and companion flanges for maximum service pressure of 300 lb per sq in. which will be available soon.

Thread Connections

Threads used for fittings are the same American Standard taper pipe threads as those used for pipe, and unless otherwise ordered, right-hand

threads are used. To facilitate drainage, some elbows have the thread tapped at an angle to provide a pitch of the connecting pipe of ¼ in. to the foot. These elbows are known to the trade as pitched elbows and are commercially available. Malleable iron fittings, like brass fittings, are cast with a round instead of a flat band or bead, or with no bead at all. Fittings are designated as male or female, depending on whether the threads are on the outside or inside, respectively.

Flanged fittings are generally used in the best practice for connecting all piping above 4 in. in diameter. While screwed fittings may be used for the larger sizes and are satisfactory under the proper working conditions, it will be found difficult either to make or to break the joints in these large sizes.

A number of different flange facings in common use are plain face, raised face, tongue and groove, and male and female. Cast-iron fittings for 125 lb pressure and below are normally furnished with a plain face, while the 250 lb cast-iron fittings are supplied with a $\frac{1}{16}$-inch raised face. The standard facing for steel flanged fittings for 150 and 300 lb is a $\frac{1}{16}$-inch raised face although these fittings are obtainable with a variety of facings. The gasket surface of the raised face may be finished smooth or may be machined with concentric or spiral grooves often referred to as serrated face or phonograph finish, respectively.

The dimensions of elbows, tees and crosses for 125 lb cast-iron screwed fittings are given in Table 7, whereas the dimensions for 125 lb cast-iron flanged fittings are given in Tables 8 and 9.

For low temperature service not to exceed about 220 F, a number of paper or vegetable fiber gasket materials will prove satisfactory; for plain raised face flanges, rubber or rubber inserted gaskets are commonly employed. Asbestos composition gaskets are probably the most widely used, particularly where the temperature exceeds 250 F. Jacketed asbestos and metallic gaskets may be used for any pressure and temperature conditions, but preferably only with a relatively narrow recessed facing.

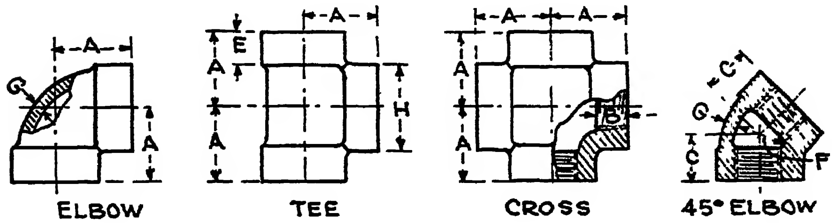
WELDING

Erection of piping in heating and ventilating installations by means of fusion welding has been commonly accepted in the past few years as a competitive method to the screwed and flanged joint. Since the question of economy of welding as against the use of screwed and flanged fittings is dependent on the individual job, the use of welding is generally recommended on the basis of a greatly reduced cost of maintenance and repair, of less weight resulting from the use of a lighter-weight pipe, and of increased economy in pipe insulation, hangers, and supports rather than on the basis of any economy that might be effected in actual erection by welding.

Fusion welding, commonly used in erection of piping, is defined as the process of joining metal parts in the molten, or molten and vapor states, without the application of mechanical pressure or blows. Fusion welding embraces gas welding and electric arc welding, both of which are commonly used to produce acceptable welds.

Welding application requires the same basic knowledge of design as do the other types of assembly, but in addition, requires a generous knowledge of the sciences involved, particularly as to welding qualities of metal, their reaction to extremely high temperatures, and the ability to determine and use only the best quality welding rods. This requirement applies equally to employer and employee with the employer accepting

TABLE 7. TENTATIVE AMERICAN STANDARD DIMENSIONS OF ELBOWS, 45 DEG ELBOWS, TEES, AND CROSSES (STRAIGHT SIZES) FOR 125 LB CAST-IRON SCREWED FITTINGS



	A	C	B	E	F		G	H
Nominal Pipe Size	Center to End, Elbows, Tees and Crosses	Center to End, 45 Deg Elbows	Length of Thread Min	Width of Band, Min	Inside Diameter of Fitting		Metal Thickness, Min	Outside Diameter of Band Min
					Min	Max		
¼	0.81	0.73	0.32	0.38	0.540	0.584	0.110	0 93
⅜	0.95	0 80	0.36	0.44	0 675	0 719	0 120	1.12
½	1.12	0 88	0.43	0 50	0 840	0.897	0 130	1.34
¾	1.31	0.98	0 50	0.56	1 050	1.107	0.155	1.63
1	1.50	1.12	0.58	0 62	1.315	1.385	0 170	1.95
1¼	1.75	1.29	0.67	0 69	1 660	1.730	0.185	2 39
1½	1.94	1 43	0.70	0.75	1.900	1 970	0.200	2.68
2	2.25	1.68	0.75	0.84	2.375	2.445	0 220	3.28
2½	2.70	1.95	0.92	0 94	2 875	2 975	0.240	3.86
3	3.08	2.17	0.98	1 00	3.500	3.600	0.260	4.62
3½	3.42	2.39	1.03	1 06	4 000	4.100	0.280	5.20
4	3.79	2 61	1.08	1 12	4 500	4.600	0.310	5.79
5	4.50	3.05	1.18	1 18	5.563	5.663	0.380	7 05
6	5.13	3 46	1.28	1 28	6.625	6.725	0.430	8.28
8	6.56	4.28	1.47	1 47	8 625	8.725	0.550	10.63
10	8.08	5.16	1 68	1 68	10.750	10.850	0.690	13.12
12	9.50	5 97	1 88	1.88	12 750	12.850	0.800	15.47
14 O.D.	10.40	....	2.00	2.00	14 000	14.100	0 880	16.94
16 O.D.	11.82	....	2 20	2 20	16 000	16.100	1.000	19 30

All dimensions given in inches

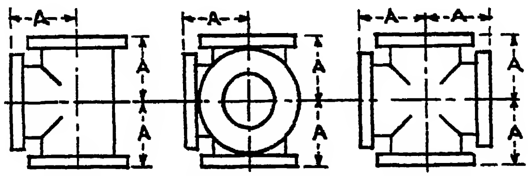
all of the responsibility. Thus the employer should select his welding mechanics with good judgment, provide them with first-class equipment and tools, arrange for their training and use of acceptable workmanship standards, and at regular intervals subject their work to prescribed tests. Industry will not accept the employment of mechanics of undetermined ability nor on the basis of past experience. Neither does industry accept the statement that a weld is only as good as the workman who makes it. The control Codes now in process of adoption will be the law governing

the use of the welding process. These Codes prohibit individual practices contrary to their specified procedure and rules of control, and this is predicated upon the sound requirement that the employer must assume full responsibility for the deposited weld.

It is advisable that this management responsibility be included in all welding specifications and that authoritative standards of workmanship also be specified. The standards of workmanship for this industry are as set forth in the *Standard Manual on Pipe Welding* of the *Heating, Piping and Air Conditioning Contractors National Association.*

A complete line of manufactured steel welding fittings is now available

TABLE 8. AMERICAN STANDARD DIMENSIONS OF TEES AND CROSSES (STRAIGHT SIZES) FOR 125 LB CAST-IRON FLANGED FITTINGS



TEE SIDE OUTLET CROSS

NOMINAL PIPE SIZE a-b	A	AA	DIAMETER OF FLANGE	THICKNESS OF FLANGE, MIN	METAL THICKNESS OF BODY, MIN
	CENTER TO FACE TEES AND CROSSES b-c	FACE TO FACE TEES AND CROSSES b-c			
1	3 1/2	7	4 1/4	7/16	7/16
1 1/4	3 3/4	7 1/2	4 5/8	1/2	7/16
1 1/2	4	8	5	9/16	7/16
2	4 1/2	9	6	5/8	7/16
2 1/2	5	10	7	11/16	7/16
3	5 1/2	11	7 1/2	3/4	7/16
3 1/2	6	12	8 1/2	13/16	7/16
4	6 1/2	13	9	15/16	1/2
5	7 1/2	15	10	15/16	1/2
6	8	16	11	1	9/16
8	9	18	13 1/2	1 1/8	5/8
10	11	22	16	1 3/16	3/4
12	12	24	19	1 1/4	13/16
14 O D.	14	28	21	1 3/8	7/8
16 O D.	15	30	23 1/2	1 7/16	1
18 O D.	16 1/2	33	25	1 9/16	1 1/16
20 O.D.	18	36	27 1/2	1 11/16	1 1/8
24 O D.	22	44	32	1 7/8	1 1/4
30 O.D.	25	50	38 3/4	2 1/8	1 7/16
36 O D.	28	56	46	2 3/8	1 5/8
42 O D.	31	62	53	2 5/8	1 13/16
48 O.D.	34	68	59 1/2	2 3/4	2

All dimensions given in inches

[a]Size of all fittings listed indicates nominal inside diameter of port.

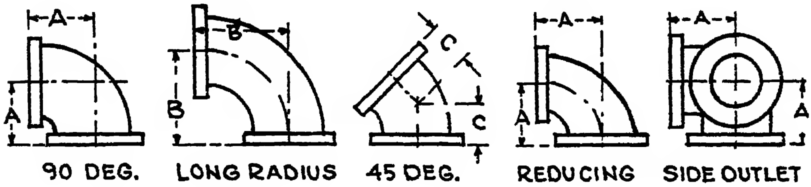
[b]Tees, side outlet tees, and crosses, 16 in and smaller, reducing on the outlet, have the same dimensions center to face, and face to face as straight size fittings corresponding to the size of the larger opening. Sizes 18 in. and larger, reducing on the outlet, are made in two lengths, depending on the size of the outlet

[c]Tees and crosses, reducing on run only, carry same dimensions center to face and face to face as a straight size fitting of the larger opening

with plain ends machine beveled for welding and with radii similar to short and long radius flanged fittings. Some typical types of these fittings are shown in Fig. 3. They are made in pipe sizes ¾ to 24 in., standard and extra heavy, in steel, wrought iron, brass, copper, and special alloys.

Socket welding fittings of forged steel are also commercially available. These fittings have a machined recess into which the pipe slips. A fillet weld between the pipe and socket edge provides a pressure-tight joint. A proposed American Standard containing dimensions of steel welding-neck flanges for pressures up to 1500 lb per sq in. has been developed in *A.S.A*

Table 9 American Standard Dimensions of Elbows for 125 Lb Cast-Iron Flanged Fittings



Nominal Pipe Size[a]	A	B	C	Diameter of Flange	Thickness of Flange, Min	Metal Thickness of Body Min
	Center to Face Elbow b-c-d	Center to Face Long Radius Elbow b-c-d	Center to Face 45 Deg Elbow c			
1	3½	5	1¾	4¼	7/16	7/16
1¼	3¾	5½	2	4⅝	½	7/16
1½	4	6	2¼	5	9/16	7/16
2	4½	6½	2½	6	⅝	7/16
2½	5	7	3	7	11/16	7/16
3	5½	7¾	3	7½	¾	7/16
3½	6	8½	3½	8½	13/16	7/16
4	6½	9	4	9	15/16	½
5	7½	10¼	4½	10	15/16	½
6	8	11½	5	11	1	9/16
8	9	14	5½	13½	1⅛	⅝
10	11	16½	6½	16	1 3/16	¾
12	12	19	7½	19	1¼	13/16
14 O.D.	14	21½	7½	21	1⅜	⅞
16 O D.	15	24	8	23½	1 7/16	1
18 O.D.	16½	26½	8½	25	1 9/16	1 1/16
20 O.D.	18	29	9½	27½	1 11/16	1⅛
24 O.D.	22	34	11	32	1⅞	1¼
30 O.D.	25	41½	15	38¾	2⅛	1 7/16
36 O.D.	28	49	18	46	2⅜	1⅝
42 O.D.	31	56½	21	53	2⅝	1 13/16
48 O.D.	34	64	24	59½	2¾	2

All dimensions given in inches

[a]Size of all fittings listed indicates nominal inside diameter of port

[b]Reducing elbows and side outlet elbows carry same dimensions center to face as straight size elbows corresponding to the size of the larger opening.

[c]Special degree elbows, ranging from 1 to 45 deg, inclusive, have the same center to face dimensions as given for 45 deg elbows and those over 45 deg and up to 90 deg, inclusive, shall have the same center to face dimensions as given for 90 deg elbows. The angle designation of an elbow is its deflection from straight line flow and is the angle between the flange faces

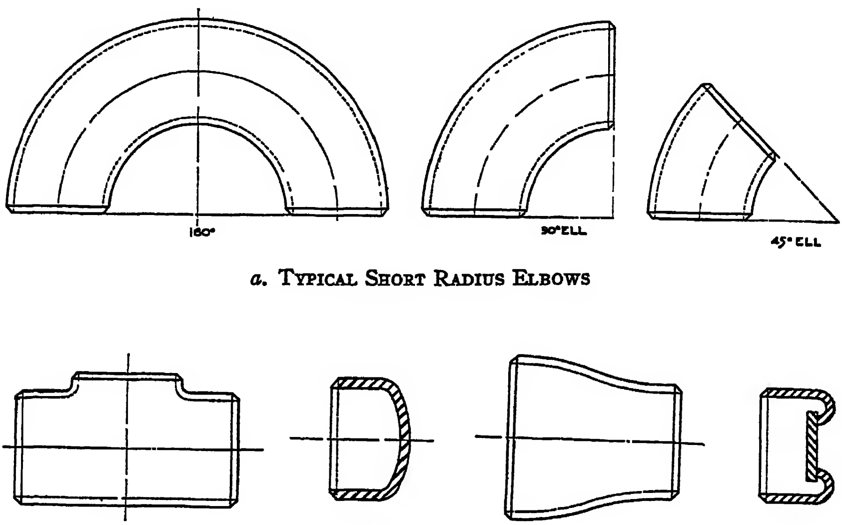
[d]Side outlet elbows shall have all openings on intersection center-lines.

Sectional Committee B16. Tables 10 and 11 give these dimensions for welding-neck flanges suitable for 150 and 300 lb per sq in. gage pressure.

VALVES

Valves are made with both threaded and flanged ends for screwed and bolted connections just as are pipe fittings.

The material used for valves of small size is generally brass or bronze for low pressures and forged steel for high pressures, while in the larger sizes either cast-iron, cast-steel or some of the steel alloys are employed.



a. Typical Short Radius Elbows

b. Tee — *c.* Forged Cap — *d* Concentric Reducer — *e.* End Closure

Fig. 3. Typical Welding Fittings

Practically all iron or steel valves intended for steam or water work are bronze-mounted or trimmed.

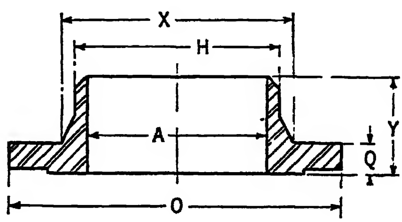
Brass, bronze, and iron valves are generally designed for standard or extra heavy service, the former being used up to 125 lb and the latter up to 250 lb saturated steam working pressure, although most manufacturers also make valves for medium pressure up to 175 lb steam working pressure. The more common types are gate valves or straightway valves, globe valves, angle valves, check valves and automatic valves, such as reducing and back-pressure valves.

Gate valves are the most frequently used of all valves since in their open position the resistance to flow is a minimum. These valves may be secured with either a rising or a non-rising stem, although in the smaller sizes the rising stem is more commonly used. The rising stem valve is desirable because the positions of the handle and stem indicate whether

the valve is open or closed, although space limitations may prevent its use. The globe valve is less expensive to manufacture than the gate valve, but its peculiar construction offers a high resistance to flow and may prevent complete drainage of the pipe line. These objections are of particular importance in heating work.

Check valves are automatic in operation and permit flow in only one direction, depending for operation on the difference in pressure between

TABLE 10. PROPOSED DIMENSIONS OF STEEL WELDING NECK FLANGES FOR MAXIMUM STEAM SERVICE PRESSURE OF 150 LB PER SQ IN. (GAGE) AT A TEMPERATURE OF 500 F, AND 100 LB AT 750 F



NOMINAL PIPE SIZE	DIAMETER OF FLANGE	THICKNESS OF FLG MIN	DIAMETER OF HUB	HUB DIAM BEGINNING OF CHAMFER	LENGTH THRU HUB	DIAM FOR STANDARD PIPE	DIAM OF BOLT CIRCLE	NO OF BOLTS	SIZE OF BOLTS
	O	*Q*	*X*	*H*	*Y*	*A*			
1	4 1/4	7/16	1 15/16	1.32	2 3/16	1 05	3 1/8	4	1/2
1 1/4	4 5/8	1/2	2 5/16	1 66	2 1/4	1 38	3 1/2	4	1/2
1 1/2	5	9/16	2 9/16	1 90	2 7/16	1 61	3 7/8	4	1/2
2	6	5/8	3 1/16	2.38	2 1/2	2.07	4 3/4	4	5/8
2 1/2	7	1 1/16	3 9/16	2.88	2 3/4	2.47	5 1/2	4	5/8
3	7 1/2	3/4	4 1/4	3.50	2 3/4	3.07	6	4	5/8
3 1/2	8 1/2	13/16	4 13/16	4.00	2 13/16	3.55	7	8	5/8
4	9	15/16	5 5/16	4.50	3	4.03	7 1/2	8	5/8
5	10	15/16	6 7/16	5.56	3 1/2	5.05	8 1/2	8	3/4
6	11	1	7 9/16	6 63	3 1/2	6 07	9 1/2	8	3/4
8	13 1/2	1 1/8	9 11/16	8.63	4	7.98	11 3/4	8	3/4
10	16	1 3/16	12	10 75	4	10.02	14 1/4	12	7/8
12	19	1 1/4	14 3/8	12.75	4 1/2	12.00	17	12	7/8
14 O. D.	21	1 3/8	15 3/4	14.00	5	13.25	18 3/4	12	1
16 O. D.	23 1/2	1 7/16	18	16.00	5	15.25	21 1/4	16	1
18 O. D.	25	1 9/16	19 7/8	18.00	5 1/2	17.25	22 3/4	16	1 1/8
20 O D.	27 1/2	1 11/16	22	20 00	5 11/16	19.25	25	20	1 1/8
24 O. D.	32	1 7/8	26 1/8	24 00	6	23.25	29 1/2	20	1 1/4

All dimensions given in inches

A raised face of 1/16 in is included in *thickness of flange minimum.*

It is recommended that the taper of the hub should not exceed 6 deg for a reasonable distance back of the chamfer in order to reduce the heat transfer while welding

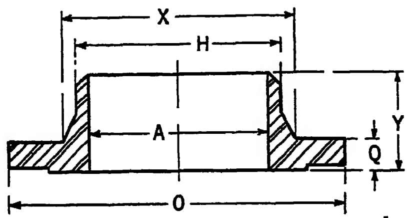
the two sides of the valve. The two principal kinds of check valves are the swing check in which a flapper is hinged to swing back and forth, and the lift check in which a dead weight disc moves vertically from its seat.

Valves commonly used for controlling steam or water supply to radiators constitute a special class since they are manufactured to meet heating system requirements. These valves are generally of the angle type and are usually made of brass. Graduations on the heads or lever

handles are often supplied to indicate the relative opening of the valve in any position. Standard roughing-in dimensions for angle-type valves are given in Table 12.

Automatic control of steam supply to individual radiators can be

Table 11. Proposed Dimensions of Steel Welding Neck Flanges for Maximum Steam Service Pressure of 300 Lb per Sq In (Gage) at a Temperature of 750 F



Nominal Pipe Size	Diam of Flange	Thickness of Flange Min	Diam of Hub	Hub Diam Beginning of Chamfer	Length Thru Hub	Diam for Standard Pipe	Diam for Extra Strong Pipe	Diam of Bolt Circle	No of Bolts	Size of Bolts
	O	*Q*	*X*	*H*	*Y*	*A*	*A*			
*2	6 1/2	7/8	3 5/16	2.38	2 3/4	2 07	1.94	5	8	5/8
2 1/2	7 1/2	1	3 15/16	2.88	3	2.47	2.32	5 7/8	8	3/4
3	8 1/4	1 1/8	4 5/8	3.50	3 1/8	3.07	2.90	6 5/8	8	3/4
3 1/2	9	1 3/16	5 1/4	4.00	3 3/16	3.55	3.36	7 1/4	8	3/4
4	10	1 1/4	5 3/4	4.50	3 3/8	4.03	3.83	7 7/8	8	3/4
5	11	1 3/8	7	5.56	3 7/8	5 05	4.81	9 1/4	8	3/4
6	12 1/2	1 7/16	8 1/8	6.63	3 7/8	6.07	5.76	10 5/8	12	3/4
8	15	1 5/8	10 1/4	8.63	4 3/8	7.98	7.63	13	12	7/8
10	17 1/2	1 7/8	12 5/8	10.75	4 5/8	10.02	9.75	15 1/4	16	1
12	20 1/2	2	14 3/4	12.75	5 1/8	12.00	11.75	17 3/4	16	1 1/8
14 O. D	23	2 1/8	16 3/4	14.00	5 5/8	13 25	....	20 1/4	20	1 1/8
16 O. D	25 1/2	2 1/4	19	16.00	5 3/4	15 25	....	22 1/2	20	1 1/4
18 O. D	28	2 3/8	21	18.00	6 1/4	17.25	....	24 3/4	24	1 1/4
20 O. D.	30 1/2	2 1/2	23 1/8	20.00	6 3/8	19 25	....	27	24	1 1/4
24 O D.	36	2 3/4	27 5/8	24.00	6 5/8	23 25	....	32	24	1 1/2

*For sizes below 2 in use dimensions of 600 lb flanges

All dimensions given in inches.

A raised face of 1/16 in is included in *thickness of flange minimum*

It is recommended that the taper of the hub should not exceed 6 deg for a reasonable distance back of the chamfer in order to reduce the heat transfer while welding

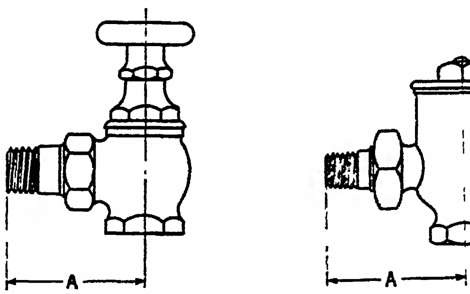
effected by use of direct-acting radiator valves having a thermostatic element at the valve, or near to it. The direct-acting valve is usually an angle-type valve containing a thermostatic element which permits the flow of steam in accordance with room temperature requirements. These valves usually are capable of adjustment to permit variation in room temperature to suit individual taste.

Ordinary steam valves may be used for hot water service by drilling a 1/16-in. hole through the web forming the seat to insure sufficient circulation to prevent freezing when the valve is closed. Valves made particularly

for use in hot water heating systems are of less complex design, one type consisting of a simple butterfly valve, and another of a quick opening type in which a part in the valve mechanism matches up with an opening in the valve body.

In one-pipe steam-heating systems, automatic air valves are required at the radiators. Two common types of air valves available are the vacuum type and the straight-pressure type. Vacuum valves permit the expulsion of air from the radiators when the steam pressure rises and, in addition, act as checks to prevent the return of air into the radiator when

TABLE 12. STANDARD ROUGHING-IN DIMENSIONS ANGLE TYPE VALVES



SIZE OF VALVE	DIMENSION A STEAM AND HOT WATER ANGLE VALVES AND UNION ELBOWS EFFECTIVE JANUARY 1 1926	DIMENSION A MODULATING VALVES EFFECTIVE JANUARY 1, 1926	DIMENSION A RETURN LINE VACUUM VALVES EFFECTIVE JANUARY 1, 1926
½	2¼	2¾	3¼
¾	2¾	2¾	-----
1	3	3	-----
1¼	3½	3½	-----
1½	3¾	3¾	-----
2	4¼	4¼	-----
Tolerance	±⅛	±⅛	-----

All dimensions given in inches.

Connecting ends shall be threaded and gaged as to threading according to the American (Taper) Pipe Thread Standard, *A.S.A.* No. B2—1919.

The standardization of the Roughing-in Dimensions of Angle Steam and Hot Water, and Modulating Radiator Valves was made possible by the cooperation of the Manufacturers Standardization Society of the Valves and Fittings Industry.

a vacuum is formed by the condensation of steam after the supply pressure has dropped. Ordinary air valves permit the expulsion of air from the radiator when steam is supplied under pressure, but when the pressure dies down and a vacuum tends to be formed the air is drawn back into the radiator.

A system operating continuously or intermittently and supplied with vacuum valves will generally heat more quickly than one provided with non-vacuum air valves; thus, it will effect considerable economy of fuel because the idle period during which no heat is delivered is shortened. In those cases, where a system is equipped with vacuum air valves and which has been cold for several days, the system will probably have an

internal pressure within the radiator closely approaching atmospheric. At such times, the vacuum valve will not vent the system any more rapidly than the ordinary type. Automatic air valves are provided with a float to close them in case the radiator becomes flooded with water because it does not drain properly.

CORROSION[2]

Corrosion is sometimes encountered in heating work on the outside of buried pipes or the inside of steam heating systems; it is seldom experienced in hot water heating systems unless the water is frequently renewed. Piping buried in the ground is quite successfully protected by coatings of the asphaltic type which are usually applied hot and often reinforced with fabric wrappings. Galvanizing by the hot-dip process and painting with specially prepared mixtures also afford some protection.

Internal corrosion in steam heating systems occurs principally in the condensate return pipes and is nearly always caused by oxygen or carbon dioxide, or both, in solution in the condensate. Oxygen may enter the heating system with the steam, owing to its presence in the boiler-feed water, or it may enter as air through small leaks, particularly in systems which operate at sub-atmospheric pressures. When a steam heating system is operated intermittently, air rushes in during each shutdown period and oxygen is absorbed by the condensate which clings to the interior surfaces of the pipes and radiators. The rate of corrosion depends upon the amounts of oxygen and carbon dioxide present in solution, upon the operating temperature, and upon the length of time that the pipe surfaces are in contact with gas-laden condensate.

Another possible cause of corrosion is a flow of electric current sometimes resulting from faulty electrical circuits which should be corrected. Electrolytic corrosion also may occur because of the presence of two dissimilar metals, such as brass and iron, but the condensate in practically all steam heating systems is such a weak electrolyte that this cause of corrosion is very infrequent.

If trouble is experienced from corrosion, oxygen should be eliminated from the feed water by proper deaeration with commercial apparatus. The elimination of the oxygen due to air leakage is more difficult because of the multitude of small leaks which exist around valve stems and in pipe joints. In vacuum systems, however, an attempt should be made to minimize such leakage.

Carbon dioxide in varying amounts is contained in steam produced from the majority of water supplies. It is formed from the breaking down of carbonates and bicarbonates which are present in nearly all natural waters. It can be partly removed by chemical treatment and deaëration, but there is no simple method whereby it can be entirely eliminated.

These gases cause corrosion only when in solution in the condensate; when they are mixed with dry steam their corrosive effect is negligible.

---

[2]New Light on Heating System Corrosion, by J. H. Walker (*Heating and Ventilating*, May, 1933). Corrosion Studies in Steam Heating Systems, by R. R. Seeber, F. A. Rohrman and G. E. Smedberg, (A.S.H.V.E. TRANSACTIONS, Vol. 40, 1934). Corrosion Studies in Steam Heating Systems, by R. R. Seeber, F. A. Rohrman and G. E. Smedberg, (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, February, 1936). Tests of a Gas Vapor Heating System, by R. R. Seeber, G. E. Smedberg and A. E. Felt, (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, April, 1936).

The amount of gas in solution depends upon the partial pressure of that gas in the atmosphere above the surface of the solution, in accordance with the well known physical law of Henry and Dalton[3]. The exact application of this law, however, assumes equilibrium conditions which do not always exist under the flow conditions prevailing in a heating system.

Distinction should be made between corrosion in heating systems proper and in the condensate discharge lines from other apparatus using steam, such as water heaters, kitchen equipment, and sterilizers. Experience has shown that in heating systems the partial pressures of the gases do not reach such magnitudes as to cause harmful amounts of gas to become dissolved in the condensate when steam supplies are of reasonable purity. In other kinds of steam-using apparatus which are not ordinarily well vented, the gases tend to accumulate in the steam space and to become dissolved in the condensate in appreciable concentrations. Consequently, corrosion is frequently observed in the condensate discharge lines from such apparatus, but this does not necessarily indicate that equally serious corrosion is taking place in the heating system supplied with steam from the same source.

When corrosive conditions are believed to exist, their seriousness should be determined by actual measurement, rather than by inference from isolated instances of pipe failures. The *National District Heating Association* has perfected a corrosion tester for measuring the inherent corrosiveness of existing conditions. This corrosion tester consists of a frame supporting three coils of wire which are carefully weighed. After the tester has been inserted in the pipe line for a definite length of time, the loss of weight of the coils, referred to an established scale, indicates the relative corrosiveness of the condensate. Accompanying such corrosion measurements, a careful chemical analysis should be made of the condensate, and the findings will serve as a basis for an intelligent study of the problem.

Corrosion, if found to exist, can be lessened or overcome by several means. If the steam supply is found to be definitely contaminated, proper chemical treatment of the water, followed by deaëration, is an obvious remedy. The leaks in the piping system, particularly in vacuum systems, should be stopped so far as is practicable.

Some success has been reported with the use of inhibitors, chief among which are oil, and sodium silicate. Oil may be fed into the main steam-supply pipe by means of a sight-feed lubricator. The type of oil known as 600-W is usually recommended. In the present state of knowledge on this point, the quantity to be fed can best be determined by trial. The use of sodium silicate, fed in a similar manner, is reported to be successful but it has not been widely used.

In view of the fact that corrosion is most frequently found in the return lines from special equipment, which constitute a relatively small part of the total piping in a building, a simple solution of the corrosion problem may be to use non-corroding materials in those certain portions of the piping system, since the higher cost will usually be an unappreciable portion of the total. Brass and copper are undoubtedly less subject to

[3]Some Fundamental Considerations of Corrosion in Steam and Condensate Lines, by R. E. Hall and A. R. Mumford (A.S.H.V.E. TRANSACTIONS, Vol 38, 1932).

this type of corrosion than the ferrous metals, and considerable attention is now being given to corrosion-resistant linings for ferrous pipe. Cast-iron pipe, sometimes alloyed with other metals, also deserves consideration.

PROBLEMS IN PRACTICE

1 ● What is the meaning of IPS brass pipe?

It means that the brass pipe has the same external diameter as steel pipe in the same nominal pipe size and that the wall thickness is sufficient to allow cutting of threads for use with standard size threaded fittings

2 ● Why is thin-walled copper pipe made up with sweated joints?

If the pipe were threaded it would be necessary to use at least standard-weight wall thickness on account of the metal removed in threading. Flared ends with coupling nuts may be used, but this construction is expensive and hard to keep tight

3 ● How are pipes designated in diameters of 12 in. and less?

By weight and nominal size, referring to the approximate inside diameter

4 ● How are pipe sizes designated in diameters of 14 in. and more?

By wall thickness and outside diameter

5 ● Why are expansion joints required in steam pipes?

To care for the change in length of the line brought about by a change in temperature.

6 ● What devices are used for taking up expansion?

Expansion joints, swivel joints, and the inherent flexibility of the pipe itself.

7 ● Where are swivel joints principally used?

In branch connections to radiators, and in the risers of multi-story buildings where they are installed between the floor joists.

8 ● Name three grades of American Standard screwed pipe fittings.

125-lb cast-iron, 150-lb malleable iron, and 250-lb cast-iron

9 ● In what sizes are American Standard cast-iron flanges and flanged fittings for 25-lb saturated steam pressure made?

In nominal sizes from 4 in. to 72 in., inclusive.

10 ● What fittings are generally used for threaded connections in low pressure heating systems?

Cast-iron.

Chapter 35

WATER SUPPLY PIPING AND WATER HEATING

Maximum Possible Flow, Maximum Probable Flow, Average Probable Flow, Factor of Usage, Kind of Pipe Used, Sizing of Risers, Sizing of Mains, Sizing of Systems, Hot Water Supply, Hot Water Heating, Hot Water Storage, Swimming Pool Heating Requirements

DOMESTIC water supply systems present the engineer with a design problem that requires combining the somewhat empirical rules and formulæ in use with the more or less exact hydraulic principles involved. Unlike heating and ventilating layouts, there are practically no definite data for estimating the quantity of water likely to be consumed or the probable rate of water flow at any particular moment.

Metered results in one building often show two or three times the metered amount in another building of the same size and with the same type of tenants. In hotels, one riser will often have an almost constant flow that may never be reached by another at peak load. In office buildings, the women's toilets show a far greater daily consumption than those of the men, yet at no time will they approach the hourly consumption of the men's toilet during the first hour of the day. This condition has led to a multiplicity of rules of practice which vary as much as the data used. All must of necessity be based on an assumed rate of consumption and on an assumed probability of simultaneous use, and while the formulæ employed may have been derived on sound technical basis the assumptions are often in error.

To arrive at a safe standard, the approximate rate of flow of each fixture to be supplied must be known and the probable number of fixtures in use at any one time must be assumed. Obviously, the maximum number of fixtures assumed to be in use must be taken at the peak of demand and the lines must be made adequate to supply such a peak regardless of the riser or branch on which the demand may occur. This means that all water piping under the usual conditions will be over-sized.

In tall buildings it is customary to divide the water supply systems, both hot and cold, into sections of 10 to 20 stories. Such *zoning*[1] or

---

[1]It is impractical to attempt to size piping so as to produce the proper pressure on fixtures at different levels by employing friction, owing to the fact that this friction will be built up to the amount desired only in times of maximum demand and at all other times the friction will be only a fraction of the maximum friction so that the fixtures by this method are subjected to a varying pressure on the water supply line. A much more practical method is to throttle the flow at the fixture, or to use flow regulators, so that the quantity of water delivered will approximate the fixture demands and so that this is accomplished without splashing or noise.

sectionalizing is for the purpose of avoiding excessive pressures on the fixtures in the lower stories of each system. This limits the consideration of water pipe sizes to horizontal mains and to risers not exceeding 20 stories in height or about 200 ft.

For the purpose of this chapter the following terms will be used and should be clearly distinguished from one another:

Maximum Possible Flow: The flow which would occur if the outlets on all fixtures were opened simultaneously. This condition is seldom, if ever, obtained in actual practice except in cases of gang showers controlled from one common valve, and similar conditions

Maximum Probable Flow: The maximum flow which any pipe is likely to carry under the peak conditions. This is the most important amount to be considered in pipe sizing.

Average Probable Flow: The flow likely to be required through the line under normal conditions.

It is evident that any pipe adequate to take care of the *maximum*

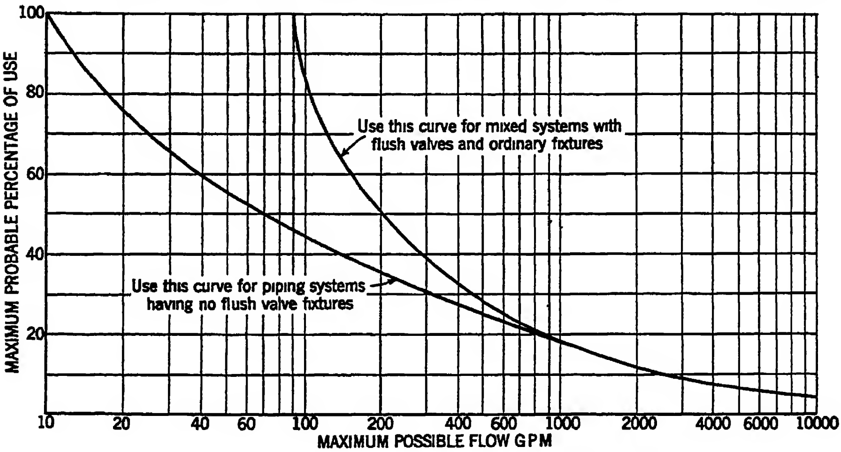


FIG. 1. CHART SHOWING RELATION BETWEEN MAXIMUM POSSIBLE FLOW AND MAXIMUM PROBABLE PERCENTAGE OF USE

probable flow will also be more than able to take care of the *average probable flow*, and hence the latter has no bearing on the pipe size.

MAXIMUM PROBABLE FLOW

There are two factors to be considered in calculating the maximum probable flow, namely, (1) the quantity of water that will flow from the outlets when they are open, and (2) the number of outlets likely to be open *at the same time.* Table 1 shows the maximum approximate rate of flow from each fixture when it is in use, and will serve as a guide in estimating maximum probable flow demands although there is considerable variation in different fixtures and valves. Probably the flow under normal water pressures, or with the pressure properly throttled, will not differ greatly from the values stated. With the aid of this table it is possible to calculate the maximum possible flow with all outlets open in both the hot and cold water lines.

To obtain the maximum probable flow it is necessary to multiply the maximum possible flow by a factor of usage, and this factor varies with the installation and the number of fixtures in the installation. It is evident that with two fixtures it is quite possible that both will at some time be in operation simultaneously. With 200 fixtures, it is unlikely the entire 200 would ever operate at the same time. Consequently, the factor of usage reduces as the number of fixtures becomes greater, all other things being equal.

TABLE 1 APPROXIMATE FLOW FROM FIXTURES UNDER NORMAL WATER PRESSURES

FIXTURES	COLD WATER (GALLONS PER MINUTE)	HOT WATER (GALLONS PER MINUTE)
Water-closets, flush valve	45[a]	0
Water-closets, flush tank	10	0
Urinals, flush valve	30[a]	0
Urinals, flush tank	10	0
Urinals, automatic tank	1	0
Urinals, perforated pipe per foot	10	0
Lavatories	3	3
Showers, 4 in. heads, ½ in. inlets	3	3
Showers, 6 in. heads or larger	6	6
Needle bath	30	30
Shampoo spray	1	1
Liver spray	2	2
Manicure table	1½	1½
Baths, tub	5	5
Kitchen sink	4	4
Pantry sink, ordinary	2	2
Pantry sink, large bibb	6	6
Slop sinks	6	6
Wash trays	3	3
Laundry tray	6	6
Garden hose bibb	10	0

[a]Actual tests on water-closet flush valves indicate 40 gpm as the maximum rate of flow with 30 lb pressure at the valve; this would increase to 60 gpm (about 50 per cent) at 90 lb pressure. The 45 gpm has been taken as an average flow; possibly, with very low pressures just sufficient to operate the flush valve, 30 gpm could be allowed with safety Urinal flush valves would vary proportionately in the same manner

In practice all the elements will vary according to conditions; in the case of flush valve closets the duration of flush with the kind and condition of supply apparatus, the interval between flushes with the number of people using the system and their habits; and the length of the rush period with the type of installation and its location. The effect of each of these time elements on the results should be considered in connection with any data on which it is based before passing judgment on the selection of the factor of usage. The longer the duration of the flush the greater is the probability of overlapping flow. In selecting the factor of usage shown in Fig. 1 for systems having flush valves, 10 seconds was chosen as the maximum duration of flush, a value that represents an approximate average as water closets are installed.

While the curve has been calculated for systems composed of water closets alone, it is possible to calculate probabilities for mixed systems of water closets and other smaller fixtures. It has been found however that for two systems both having the same maximum possible flow, one composed entirely of water closets and the other a mixed system of water

closets and smaller fixtures, the probability of a given rate of flow is greater for the system composed of water closets than for the mixed

TABLE 2. SCHEDULE OF SIZES FOR DOWN-FEED RISER (SEE FIG 2)

Portion of Riser	Allowable Drop per Lb 100 Ft	Maximum Probable Flow, Gallons per Minute: 5	10	15	20	25	30	40	50	60	70	80	90	100	125	150	200	250
T	3 5	¾	1	1¼	1¼	1½	1½	2	2	2	2½	2½	2½	3	3	3	3½	3½
S	20	¾	¾	1	1	1	1¼	1¼	1¼	1½	1½	2	2	2	2	2	2½	3
R	30	¾	¾	¾	1	1	1	1¼	1¼	1¼	1½	1½	1½	2	2	2	2½	2½
Q	30	¾	¾	¾	1	1	1	1¼	1¼	1¼	1½	1½	1½	2	2	2	2½	2½
P	30	¾	¾	¾	1	1	1	1¼	1¼	1¼	1½	1½	1½	2	2	2	2½	2½
O	30	¾	¾	¾	1	1	1	1¼	1¼	1¼	1½	1½	1½	2	2	2	2½	2½
N	30	¾	¾	¾	1	1	1	1¼	1¼	1¼	1½	1½	1½	2	2	2	2½	2½
M	30	¾	¾	¾	1	1	1	1¼	1¼	1¼	1½	1½	1½	2	2	2	2½	2½
L	30	¾	¾	¾	1	1	1	1¼	1¼	1¼	1½	1½	1½	2	2	2	2½	2½
K	30	¾	¾	¾	1	1	1	1¼	1¼	1¼	1½	1½	1½	2	2	2	2½	2½
J	30	¾	¾	¾	1	1	1	1¼	1¼	1¼	1½	1½	1½	2	2	2	2½	2½
I	30	¾	¾	¾	1	1	1	1¼	1¼	1¼	1½	1½	1½	2	2	2	2½	2½
H	30	¾	¾	¾	1	1	1	1¼	1¼	1¼	1½	1½	1½	2	2	2	2½	2½
G	30	¾	¾	¾	1	1	1	1¼	1¼	1¼	1½	1½	1½	2	2	2	2½	2½
F	30	¾	¾	¾	1	1	1	1¼	1¼	1¼	1½	1½	1½	2	2	2	2½	2½
E	30	¾	¾	¾	1	1	1	1¼	1¼	1¼	1½	1½	1½	2	2	2	2½	2½
D	30	¾	¾	¾	1	1	1	1¼	1¼	1¼	1½	1½	1½	2	2	2	2½	2½
C	30	¾	¾	¾	1	1	1	1¼	1¼	1¼	1½	1½	1½	2	2	2	2½	2½
B	30	¾	¾	¾	1	1	1	1¼	1¼	1¼	1½	1½	1½	2	2	2	2½	2½
A	30	¾	¾	¾	1	1	1	1¼	1¼	1¼	1½	1½	1½	2	2	2	2½	2½

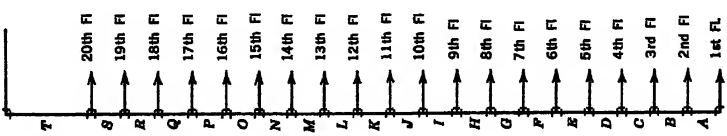


FIG. 2. TYPICAL RISER FOR 20-STORY BUILDING

system. The use of this chart then would produce results which would be on the safe side for mixed systems.

For systems composed entirely of fixtures other than flush valve fixtures the curve has been extended for smaller maximum possible flow values.

This chart applies to a normal building and not to installations where the inmates may all be required, for instance to bathe on certain days of the week and at certain hours of those days, or in schools for example where all the showers in the gymnasium may be used simultaneously after instruction periods. In such special cases a new factor of usage must be developed based on the maximum probable usage under the conditions involved

Example 1. Assume that in a normal building, such as a residential hotel or an apartment house, there are 50 flush valve water-closets, 50 lavatories, 50 sinks and 50 baths, and that it is desired to determine the maximum probable flow in a line supplying all of these fixtures with both hot and cold water

Cold Water

50 W. C x 45 gpm.......2250 gpm
50 Lavs. x 3 gpm.......150 gpm
50 Sinks x 4 gpm.......200 gpm
50 Baths x 5 gpm.......250 gpm

Maximum possible flow.......2850 gpm

Fig. 1 shows a factor of usage of 9 per cent

Maximum probable flow of cold water is 2850 × 0 09.......257 gpm

Hot Water

50 Lavs x 3 gpm.......150 gpm
50 Sinks x 4 gpm.......200 gpm
50 Baths x 5 gpm.......250 gpm

Maximum possible flow.......600 gpm

Fig. 1 shows a factor of usage of 23 per cent

Maximum probable flow of hot water is 600 × 0 23.......138 gpm

Total for main supplying cold and hot water (2850 + 600) × 0 08.......276 gpm

It should be noted that this is a *rate of flow* or an *instantaneous demand*

KIND OF PIPE USED

Before entering into the actual sizing of pipe, it is necessary to consider the kind of pipe to be used, and to make suitable allowance for corrosion and fouling during the lifetime of the system. For example, if brass, copper or alloy pipe is contemplated, it is probable that the quantities indicated in Example 1 are ample; if galvanized pipe is to be used, then it is quite likely that after a period of say 15 years the area may be decreased as much as 25 per cent and the quantitities of water assumed should be increased by 35 per cent to allow for this reduction of area; if the water contains lime it is possible that 50 per cent of the area may be lost and in such cases the flow should be doubled and no branch pipe connected to fixtures should be less than ¾ in. In all of the following calculations, the assumption is made that the water is fairly good and that a corrosion resistant type of pipe is to be used.

SIZING A DOWN-FEED RISER

Down-feed systems are commonly used for tall buildings. In sizing a riser arranged for down-feed, the gravity head permits a pressure drop that is almost prohibitive in an up-feed riser. There is a gain in riser head of 0.43 × 100 or 43 lb per 100 ft of run and hence it is quite permissible to size such a riser on the basis of a pressure drop of 30 lb per 100 ft of run, as the difference between the 43 lb generated and the 30-lb drop under maximum probable demand is ample to take care of the friction caused by the fittings. This method applied to the typical riser shown in Fig. 2 gives the schedule of sizes indicated in Table 2 for any flow from 5 to 250 gal.

SIZING AN UP-FEED RISER

When the riser is an up-feed, the opposite condition occurs; that is, there is a drop in pressure as the top of the riser is approached, due to the natural reduction in the gravity pressure, and to this must be added the pipe friction plus that introduced by the pipe fittings, all of which produce an excessive drop when compared to the conditions existing with a down-feed riser.

To size an up-feed riser the minimum pressure of the street main, or other source of supply, should be ascertained and from this should be subtracted the pressure to be maintained at the highest fixture, namely, 15 lb per square inch, plus the height in feet above the source of water pressure, multiplied by 0.43 to change from feet of head to pounds of pressure. The total length of run from the source of pressure to the farthest and highest fixture should be ascertained, and this should be changed to equivalent length of run to allow for the loss occasioned by the pipe fittings. Table 3 gives the additional lengths necessary to allow for the various fittings and valves. The drop allowable in pressure per 100 ft of run may then be obtained by multiplying the surplus pressure (over that required for the gravity head and to supply 15 lb at the fixture) by 100 and by dividing this by the equivalent length of run to the farthest or highest fixture.

Table 3 Approximate Allowances for Fittings and Valves in Feet of Straight Pipe

Size of Pipe (Inches)	Type of Fitting or Valve					
	90-Deg Elbow	45-Deg Elbow	Return Bend	Gate Valve	Globe Valve	Angle Valve
½	4	3	8	2	48	8
¾	5	3	10	3	60	10
1	5	3	10	3	60	10
1¼	6	4	12	3	72	12
1½	7	5	14	4	84	14
2	7	5	14	4	84	14
2½	10	7	20	5	120	20
3	12	8	24	6	144	24
4	18	13	36	9	216	36
5	25	18	50	13	300	50
6	30	21	60	15	360	60

Where street water pressures are available the pressure drop through the meter and service pipe must be taken into consideration. Table 4 shows the pressure loss through meters. It also gives the minimum sizes of recommended service and maximum meter deliveries.

Example 2. Assume a street pressure of 60 lb, the height of the highest fixture 50 ft, and the length of the longest run 200 ft. Without knowing the additional length of pipe to be added for the fittings it will be assumed that this is about 100 ft. The surplus pressure which will be available for pressure drop will then be 60 lb − (15 lb + 50 ft × 0.43 lb) = 60 lb − (15 lb + 21.5 lb) = 23.5 lb.

To change this into drop per 100 ft: $\dfrac{23.5 \text{ lb} \times 100}{200 \text{ ft} + 100 \text{ ft}} = 7.8$ lb per 100 ft.

The pipe may then be sized from the maximum probable flow by selecting a size that does not give a drop in excess of 7.8 lb per 100 ft.

It will be seen from Example 2 that it is impossible to size up-feed risers without determining the drop allowable in both the horizontal feed mains and the toilet room branches. Having once ascertained this allowable drop, it is simply a matter of applying it throughout the system.

Table 4. Pressure Loss Through Water Disc Meters[a]

A. W W A Standards

Rate of Flow Gpm	Approx Pressure Loss Through Meters, Lb per Sq In Pipe Size (In)							
	5/8	3/4	1	1½	2	3	4	6
5	1.5	0.5	0.2					
10	6.0	2.0	1.0	0.2				
15	14.0	5.0	2.0	0.6	0.2			
20	25.0	9.0	3.5	1.0	0.4			
25		13.5	5.5	1.5	0.6			
30		19.5	8.0	2.0	0.9			
35			11.0	3.0	1.0			
40			14.0	4.0	1.5			
45			18.0	5.0	2.0			
50			22.0	6.0	2.5	0.7		
75				14.0	5.5	1.5		
100				25.0	10.0	2.8	1.0	
125					15.0	4.0	1.5	
150					22.0	6.0	2.2	
175						8.0	3.0	
200						10.4	4.0	1.0
250							16.0	1.5
300							23.0	2.2
350								3.0
400								4.0
500								6.5
600								9.0
800								16.0
1000								25.0

Minimum Size of Service Recommended

Rate of Flow Gpm	Approx. Minimum Pipe Size of Service, Main to Meter (In) Maximum Length (Ft)				
	30	75	100	150	200
1–20	¾	¾	1	1	1
20–30	¾	1	1	1½	1½
30–50	1	1½	1½	1½	1½
50–100	1½	1½	2	2	2
100–150	1½	2	2	2½	2½

Safe Maximum Delivery of Meters

Meter Size In.	Capacity, Gpm Based on 25 Lb Loss Through Meter
5/8	20
¾	34
1	53
1½	100
2	160
3	315
6	500
8	1000

[a]Pressure loss through compound and current meters are less than shown in table. For exact information consult manufacturers.

HORIZONTAL SUPPLY MAINS

The horizontal mains supplying the risers at the top of a down-feed system must be liberally sized unless the house tank is set at a much higher elevation than usual. To provide a gravity head on the highest fixtures of 15 lb per square inch it is necessary for the water line in the house tank to be nearly 40 ft higher, and with the line loss considered this becomes about 45 ft. Such heights are not often practical and as a result the pressure on the highest fixtures either is reduced to 7 lb (which is sufficient to operate a flush valve), or flush tank water-closets are substituted, or a separate cold and hot water supply is installed with a small pneumatic tank to give the increase in pressure necessary. The chief objection to the use of a pneumatic tank is that a separate hot water heater is required and this heater must be located either sufficiently below the highest fixtures to obtain a gravity circulation, or it must be provided with a circulating pump in order to force the hot water to the top floor level.

The most common solution is to place the house tank as high as the structural and architectural conditions will permit and then to use liberally-sized lines between the house tank and the upper fixtures, say for the two top stories, below which the riser sizes may be reduced to those indicated in Fig. 2 and Table 2. Where the house tank is only one story above the top fixtures, flush tank water-closets must be used and the drop in the entire run from the house tank down to the farthest fixture should not exceed 1 lb; the less, the better. This means that if the total equivalent run to the farthest top fixtures supplied is 300 ft, the drop per 100 ft should not exceed $\frac{1 \text{ lb} \times 100}{300}$ or 0.33 lb per 100 ft. The friction curves shown in Fig. 3 may be used for quickly determining the proper size of pipe to give any desired drop in pounds per 100 ft of equivalent run.

OVERHEAD DISTRIBUTION MAIN

Example 3. Suppose an installation has a house tank in which the water line is 20 ft above the level of the top fixtures to be supplied and that the length of run to the farthest fixtures on this level is 400 ft with the pipe fittings adding another 200 ft, making an equivalent length of 600 ft. What would be the size of main coming out of the tank where a maximum flow rate of 400 gpm may be expected, of the horizontal main where a maximum flow rate of 200 gpm may be expected, and of the riser down to the fixture level where the maximum flow rate is approximately 100 gpm?

Here the level of the water in the house tank is 20 ft above the faucet of the highest fixture and the gravity pressure will be 0.43 lb × 20 ft = 8.6 lb and, if a total pressure drop of 1 lb is assumed, the pressure on the farthest fixture under times of peak load will be 8.6 lb − 1 lb = 7.6 lb while the drop per 100 ft of equivalent run will have to be $\frac{1 \text{ lb} \times 100}{600} = 0.1667 \text{ lb}$.

Referring to Fig. 3 it will be noted that where the flow through the main is 400 gpm, an 8 in. pipe would be required; that where the flow is reduced to 200 gpm, a 6-in. pipe would be sufficient; and that where the flow is 100 gpm in the riser branch and riser, a 5-in. size would be correct. Of course these are somewhat excessive flows and the head from the tank is small so that large sizes are to be expected. It would be necessary to carry a 5-in. riser down to the branch to the top floor, then reduce to 4 in. for the branch to the floor below the top, and below this the sizes in Table 2 could be followed. In such a case, flush tank closets should doubtless be substituted.

Had the tank been set 10 ft higher, the head available to be used up in friction, but

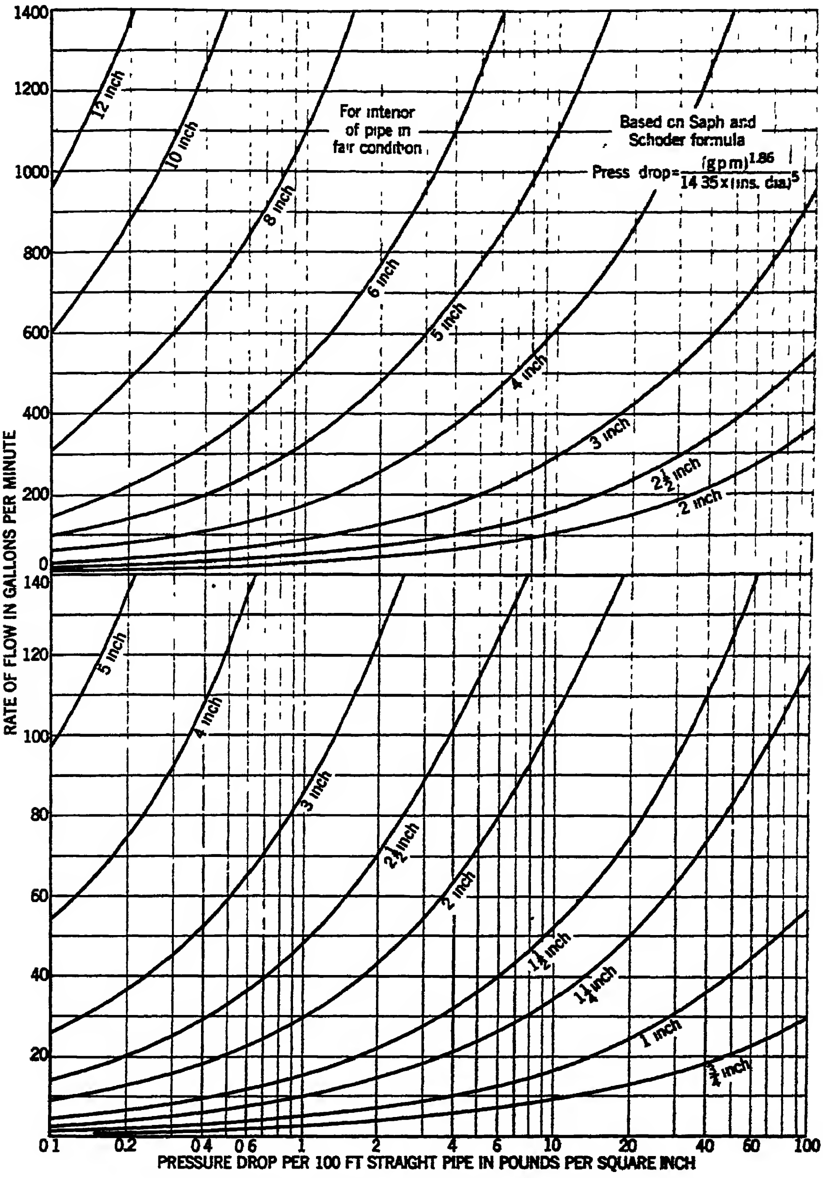
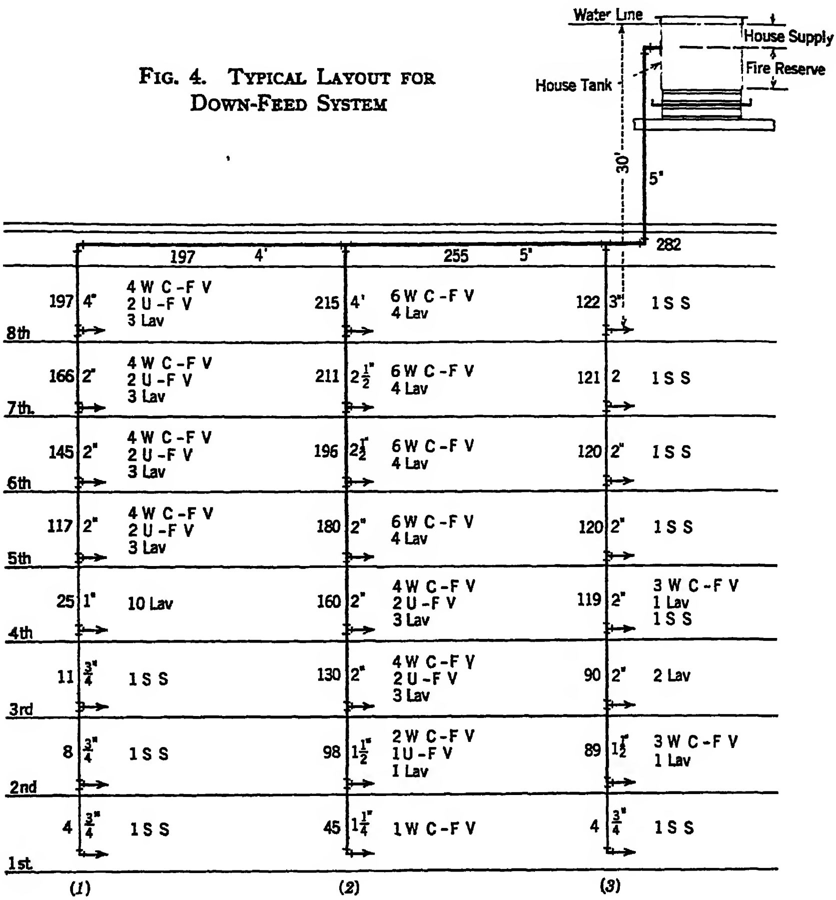


FIG. 3. CHART GIVING PRESSURE DROP FOR VARIOUS RATES OF FLOW OF WATER

still giving the same pressure at the top fixtures, would have been 0.43 lb × 10 ft or 4.3 lb greater and this, with the 1 lb drop used previously, would give a total allowable drop of 1 lb + 4.3 lb = 5.3 lb which, divided by the 600 ft equivalent run gives a drop per 100 ft of $\frac{5.3 \times 100}{600} = 0.9$ **lb**

Fig. 4. Typical Layout for Down-Feed System



and, with this drop, the sizes according to the chart (Fig. 3) are 6 in., 5 in., and 4 in., respectively, while if the run is reduced to 200 ft instead of 600 ft, the allowable drop will be $\frac{5.3 \text{ lb} \times 100}{200} = 2.7$ lb per 100 ft. This gives 5 in., 4 in., and 3 in., respectively, for the flows of 400, 200, and 100 gpm.

From Example 3 it is evident that, while the down-feed system possesses certain economies in size for the riser portion, it is quite likely to involve large distribution main sizes, especially when the tank is not elevated to a considerable degree.

SIZING A PIPING SYSTEM

Example 4. Fig. 4 shows a typical layout with three risers extending eight stories and with the fixtures noted on each floor. First this will be solved for a down-feed arrangement assuming that the level of the water in the house tank is 30 ft above the fixtures on the top floor, that the length of run from the tank to the farthest fixture is 200 ft, equivalent length of fittings 100 ft, and the pressure required at the fixture is 7 lb.

TABLE 5 TYPICAL CALCULATION OF PIPE SIZES ON DOWN-FEED RISER WITH FLUSH VALVE WATER-CLOSETS AND URINALS

(Riser No. 1 Fig 4)

FLOOR OF BLDG	FIXTURES ON FLOOR	GPM PER FIXTURE	MAXIMUM GPM ON FLOOR	MAXIMUM GPM ON RISER	PROBABLE USE (PER CENT)	PROBABLE DEMAND RISER GPM	ALLOWABLE DROP LB PER 100 FT	PIPE SIZE IN
1st	1 S S	4	4	4	100	4	30	¾
2nd	1 S S	4	4	8	100	8	30	¾
3rd	1 S S	4	4	12	92	11	30	¾
4th	10 Lav	3	30	42	58	25	30	1
5th	4 W C 2 U 3 Lav	45 30 3	180 60 9 249	291	40	117	30	2
6tn	4 W C 2 U 3 Lav	45 30 3	180 60 9 249	510	27	145	30	2
7th	4 W C 2 U 3 Lav	45 30 3	180 60 9 249	789	21	166	30	2
8th	4 W C 2 U. 3 Lav	45 30 3	180 60 9 249	1038	19	197	2	4

The 30-ft head is equal to a static pressure of 0.43 × 30 or 12.9 lb per square inch and to maintain a pressure of 7 lb at the highest fixtures the drop allowable in pressure is 12.9 − 7.0 lb or 5.9 lb. As the total equivalent run is 300 ft, this is a drop per 100 ft of 1.97 lb, or practically 2 lb. Therefore, all risers and mains from the top floor back to the tank must be sized on the basis of a drop of 2 lb per 100 ft. Tables 5, 6, 7 and 8 show the schedule for Risers Nos. 1, 2 and 3 with the maximum possible flow taken from Table 1, the percentage of use at the peak taken from Fig. 1, and the maximum probable flow at the peak worked out for each portion of the riser, the riser sizes being taken from Table 2 as far as possible and from Fig. 3 where the amounts exceed the values given in this table; a drop of 30 lb per 100 ft is used except on the riser from the top floor back to the tank where 2 lb per 100 ft is the allowable limit.

The reduction in pipe size which would occur if flush tank water-closets were used on the top floor and only 3 lb pressure used on the fixtures is given in Tables 9 and 10. This illustrates why flush tank closets so frequently are substituted on the uppermost floor when a house tank is the source of water pressure.

If it is now assumed that Riser No. 1 is to be fed from the bottom and the minimum street pressure is 75 lb with the top fixture of the riser 80 ft above the main, the problem would be solved by determining the maximum rate of flow in each portion of the riser as shown in Table 11 and then finding the allowable drop which can be used per 100 ft. The 80 ft of riser height will use up 0.43 lb × 80 = 34.4 lb and the pressure at the top of the required 15 lb will make the total reduction 49.4 lb, leaving a balance of 25.6 lb which may be used up in friction. If the distance from the street main to the bottom of the riser, which will be assumed to be the farthest one on the horizontal line, is 100 ft, and if the fittings are sufficient to add another 100 ft, as well as the 80 ft of vertical distance up the riser, the total equivalent run will be 280 ft, which will be taken as an even 300 ft.

Table 6. Typical Calculation of Pipe Sizes on Down-Feed Riser with Flush Valve Water-Closets and Urinals

(Riser No 2. Fig. 4)

Floor of Bldg	Fixtures on Floor	Gpm per Fixture	Maximum Gpm on Floor	Maximum Gpm on Riser	Probable Use (per cent)	Probable Demand Riser Gpm	Allowable Drop Lb per 100 Ft	Pipe Size In
1st	1 W C	45	45	45	100	45	30	1¼
2nd	2 W C 1 U 1 Lav	45 30 3	90 30 3 123	168	58	98	30	1½
3rd	4 W C 2 U 3 Lav.	45 30 3	180 60 9 249	417	31	130	30	2
4th	4 W C. 2 U 3 Lav	45 30 3	180 60 9 249	666	24	160	30	2
5th	6 W C 4 Lav	45 3	270 12 282	948	19	180	30	2
6th	6 W. C 4 Lav.	45 3	270 12 282	1230	16	196	30	2½
7th	6 W. C. 4 Lav.	45 3	270 12 282	1512	14	211	30	2½
8th	6 W. C 4 Lav	45 3	270 12 282	1794	12	215	2	4

Then the allowable drop per 100 ft will be $\frac{25.6 \text{ lb} \times 100}{300} = 8.5$ lb and the sizes shown in Fig. 5 are based on this amount of drop Of course the other risers will have the same maximum flows at the bottom as they formerly had at the top, namely 215 and 122 gal, respectively, for Risers Nos 2 and 3. Combining these maximum flows in the same manner as pursued in the down-feed system it is seen that the maximum flow between Riser No. 2 and Riser No. 3 is 255 gpm, and between Riser No. 3 and the street main, 282 gpm which at a drop of 8 5 lb gives the main sizes indicated. It will be noted that in determining the maximum flow in an up-feed riser it is necessary to begin at the top floor and work down instead of beginning at the bottom floor and working up as was done in the down-feed sizing.

SIZING UP-FEED AND DOWN-FEED HOT WATER SYSTEMS

Hot water supply systems, when of the circulating type, have a few differences to be considered although the same general principles of sizing apply to these lines as to the cold water lines. Owing to the fact that there are no flush valves on the hot water piping and also because many plumbing fixtures have no hot water connections, the sizes of the hot water piping in general will be considerably less than the cold water piping in the same building. On the other hand it is almost invariably

TABLE 7. TYPICAL CALCULATION OF PIPE SIZES ON DOWN-FEED RISER WITH FLUSH VALVE WATER-CLOSETS AND URINALS

(Riser No 3 Fig. 4)

FLOOR OF BLDG.	FIXTURES ON FLOOR	GPM PER FIXTURE	MAXIMUM GPM ON FLOOR	MAXIMUM GPM ON RISER	PROBABLE USE (PER CENT)	PROBABLE DEMAND RISER GPM	ALLOWABLE DROP LB PER 100 FT	PIPE SIZE IN
1st	1 S S	4	4	4	100	4	30	¾
2nd	3 W C 1 Lav	45 3	135 3 138	142	63	89	30	1½
3rd	2 Lav	3	6	148	61	90	30	1½
4th	3 W. C 1 Lav 1 S. S	45 3 4	135 3 4 142	290	41	119	30	2
5th	1 S S	4	4	294	41	120	30	2
6th	1 S S	4	4	298	40	120	30	2
7th	1 S S	4	4	302	40	121	30	2
8th	1 S S	4	4	306	40	122	2	3

required that a gravity circulation be kept up in such hot water lines and this often has a considerable influence on the size. There are three methods of arranging circulation lines, as follows:

1. By using the plain up-feed with a return carried back from the top of the riser and paralleling it.
2. By carrying a supply riser up in one location thus supplying fixtures on up-feed, then crossing over at the top and coming down past another collection of fixtures and supplying these by a down-feed.
3. By carrying all of the water to the top of the building and dropping risers wherever needed, feeding all hot water on a down-feed system.

TABLE 8. SIZE OF DISTRIBUTION MAIN FOR DOWN-FEED SYSTEMS (SEE FIG. 4)

RISER NO	MAXIMUM GPM RISER	MAXIMUM GPM MAIN	PROBABLE USE (PER CENT)	PROBABLE GPM	ALLOWABLE DROP LB PER 100 FT	SIZE OF MAIN IN.
1	1038	1038	18	187	2	4
2	1794	2832	9	255	2	4
3	306	3138	9	282	2	5

In the first instance the up-feed riser may be sized for the same pressure drop as used for the cold water riser and, from the top of the riser *just below the top fixture connection*, a return circulation line may be carried back to the main return line in the basement and connected through a check valve, set on a 45-deg angle, and a gate valve; these return circulation lines should never be less than ¾ in., and on the farther half of the risers, not less than 1 in. to favor circulation in the far end. Typical top and bottom connections for such risers are shown in Fig. 6.

Table 9. Typical Calculation of Pipe Sizes on Down-Feed Risers with Flush Tank Water-Closets and Urinals on Top Floor Only (See Fig. 4)

Floor of Bldg.	Fixtures on Floor	Gpm per Fixture	Maximum Gpm on Floor	Maximum Gpm on Riser	Probable Use (per cent)	Probable Demand Riser Gpm	Allowable Drop Lb per 100 Ft	Pipe Size In
	Riser No 1							
7th and below				789	21	166	30	2
8th	4 W C. 2 U 3 Lav	10 10 3	40 20 9 69	858	20	172	3 3	4
	Riser No. 2							
7th and below				1512	14	211	30	2½
8th	6 W C. 4 Lav	10 3	60 12 82	1594	14	223	3.3	4
	Riser No. 3							
7th and below				302	40	121	30	2
8th	1 S. S.	4	4	306	40	122	3.3	3

For the second arrangement of hot water risers (Fig. 7*b*), circulation lines are run back from the last fixture supplied to the main return circulation line in the same manner as just described, using ¾ in. for the near risers and 1 in. for the far risers. The sizing is much more difficult, as it is necessary to start at the bottom floor of the return riser and work back to the top of this riser and then carry the maximum flow across on to the top of the corresponding supply riser and work down on this riser from the top floor to the bottom. Naturally this gives a much greater flow in the supply riser and aids circulation by reducing pipe friction. The allowable loss per 100 ft in such lines must be made about half that used for the cold water risers which do not have the combined up- and down-travel which the hot water must make.

In the third and most common arrangement (Fig. 7*c*) all of the water is carried from the tank or heater directly to the top of the building and is there distributed to the risers which are down-feed and may be sized in the

Table 10. Summary of Riser Sizes to Given Main Sizes with Flush Tank Water-Closets and Urinals on Top Floor Only. (See Fig. 4)

Riser No.	Maximum Gpm Riser	Maximum Gpm Main	Probable Use (per cent)	Probable Gpm	Allowable Drop Lb per 100 Ft	Size of Main In.
1	858	858	20	172	3 3	4
2	1594	2452	10	245	3.3	4
3	306	2758	9	248	3.3	4

regular down-feed manner if the total equivalent run either from the street main or house tank is taken into consideration. The return circulation lines from the bottom of each riser should be arranged in the manner already outlined and any riser not going to the basement to supply fixtures must have these returns carried down to the basement from the termination of the supply riser at whatever level it may end.

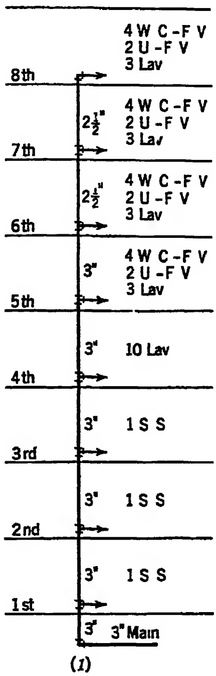


FIG. 5. UP-FEED SYSTEM

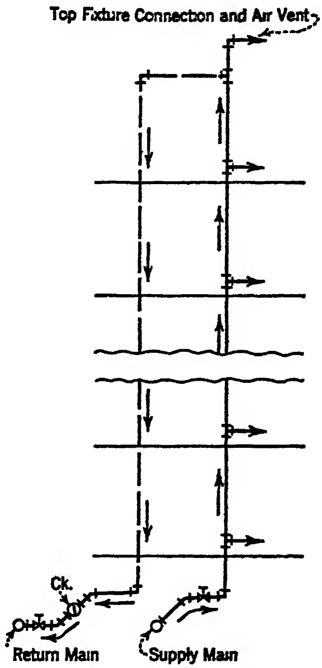


FIG. 6. SUPPLY AND RETURN MAIN CONNECTIONS FOR HOT WATER SUPPLY SYSTEM

All risers, both hot and cold, should be valved at the main with an extra check valve on the hot water return circulation so that the risers may be cut off and repaired when necessary without disturbing the service in the remainder of the system.

HOT WATER SUPPLY

Having designed the service hot water piping, the next step is to furnish some means of heating the water and in this respect it is necessary to pass from the maximum probable flow to the *maximum probable hourly demand*, which is quite different. If an instantaneous heater were used, it would require adequate capacity to provide for the heating of the water as fast as it is drawn and a heater of this type should be sized on the basis of the maximum probable flow with the accompanying heavy drafts on the heating device and with intervals of no draft at all. To balance these inequalities of flow the storage-type heater is often utilized so that the

Table 11. Typical Calculation of Pipe Sizes on Up-Feed Riser with Flush Valve Water-Closets and Urinals (See Fig. 5)

Floor of Bldg	Fixtures on Floor	Gpm per Fixture	Maximum Gpm on Floor	Maximum Gpm on Riser	Probable Use (per cent)	Probable Demand Riser Gpm	Allowable Drop Lb per 100 Ft	Pipe Size In
8th	4 W C 2 U. 3 Lav	45 30 3	180 60 9 249	249	44	109	8 5	2½
7th	4 W C. 2 U 3 Lav	45 30 3	180 60 9 249	498	28	139	8 5	2½
6th	4 W. C 2 U 3 Lav	45 30 3	180 60 9 249	747	22	164	8 5	3
5th	4 W. C 2 U 3 Lav	45 30 3	180 60 9 249	996	18	179	8 5	3
4th	10 Lav	3	30	1026	18	185	8 5	3
3rd	1 S S	4	4	1030	18	186	8 5	3
2nd	1 S S	4	4	1034	18	187	8 5	3
1st	1 S S.	4	4	1038	18	188	8 5	3

Table 12. Suggested Storage Tank Sizes for Homes and Apartments[a]

All Year Service Based on Boiler Water at 180 F				Service During Heating Season Based on Boiler Water at 215 F			
Tank Capacity Gal	Piping Connections		Number of Baths or Families	Tank Capacity Gal	Piping Connections		Number of Baths or Families
	Boiler, In	Tank, In.			Boiler, In	Tank, In.	
30	1	¾	1	30	1	¾	1
35	1¼	¾	1	40	1	¾	1
40	1¼	¾	1–2	52	1	¾	1
50	1¼	¾	1–2	66	1¼	1	1–2
60	1¼	1	1–2	82	1¼	1	2–3
72	1½	1	2–3	100	1¼	1	3
80	2	1	2–3	120	1¼	1	4
100	2	1¼	3–4	144	1½	1	5
125	2	1¼	4–5	160	2	1¼	6
150	2	1¼	5–6	200	2	1¼	6–7
200	2	1½	6–7	250	2	1½	7–9
250	2½	1½	7–9	300	2	1½	9–11
300	2½	1½	9–11	400	2½	2	11–15
400	3	2	11–15	500	2½	2	15–18
500	3	2	15–18	600	3	2½	18–21

[a]See pages 661 and 721 for further data.

water demand can be heated during periods of light demand and stored up for use during the periods of heavy demand. The total water consumption per person usually varies between 100 and 150 gal per day when laundry and culinary operations for the occupants are carried out on the same premises. The maximum hourly demand under these conditions will be found to be about one-tenth of the average daily consumption.

If one-third of the total water used is hot water and 125 gal per day is assumed as a fair average of consumption per person, it is apparent that each person uses about 40 gal of hot water per day. If one-tenth of this represents the peak hourly load, then 4 gph must be allowed per person for the heaviest demand. If the average occupancy of apartments is 3 persons, the peak hour demand per apartment will be about 12 gph. It is customary to allow 10 gph of heating capacity per apartment. Water in excess of this heating capacity drawn out during the peak hours is

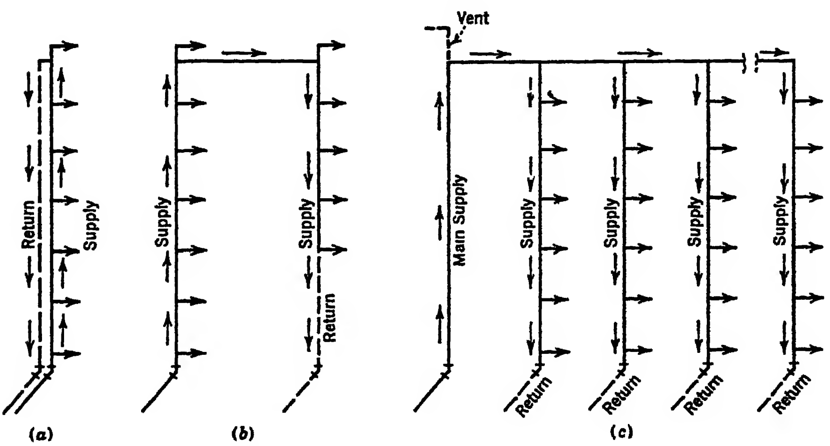


FIG 7. METHODS OF ARRANGING HOT WATER CIRCULATION LINES

provided for by storage in the hot water tank where this water is heated during hours when the demand is below the average. Table 12 gives suggested storage tank sizes for homes and apartments based on the number of families or baths.

HOT WATER HEATERS

Various types of heaters are available for supplying the hot water for domestic service in buildings. In any hot water supply system the water should be heated to a temperature between 150 and 180 F. Where the hot water requirements include supplies for kitchens, laundries or process work, the higher temperatures are used. In buildings where steam is available throughout the year, the hot water supply is usually taken from this source. In smaller domestic installations the fuel-burning device is generally automatically arranged so that hot water is supplied the entire year and not merely when the boiler is used for heating purposes.

Water is heated by various methods using heat exchangers arranged so that the boiler heating medium gives up its heat to the water in the hot water circulating system. These heat exchangers may be classified as follows:

1. Submerged steam heating coil in storage tank.
2. Submerged water heating coil in storage tank. (Fig. 8)
3. Indirect water heater, mounted on side of boiler below water line. (Fig. 9).
4. Submerged indirect water heater, placed in boiler below water line. (Fig. 10).

The efficiency of these heaters may be estimated as nearly 100 per cent as the heat loss from surface radiation of the heater and tank shell when covered with insulating material is generally reduced to a minimum. The capacities of these heaters are usually available from manufacturers

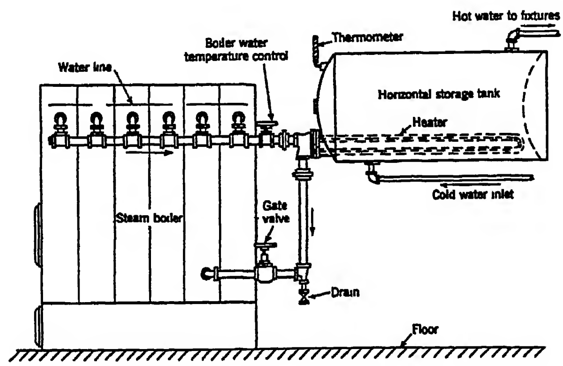


FIG. 8. HOT WATER HEATING COIL SUBMERGED IN STORAGE TANK

rating tables. The area of the inside surface of a heating coil may be determined from the following equation:

$$A = \frac{Q \times 8.33\,(t_o - t_i)}{K_o \times t_m} \tag{1}$$

where

A = surface area of coil, square feet.
Q = quantity of water heated, gallons per hour.
t_o = hot water outlet temperature, degrees Fahrenheit.
t_i = cold water inlet temperature, degrees Fahrenheit.
K_o = coefficient of heat transmission, Btu per hour per square foot surface.
For copper or brass coils K_o = 240 (steam) and 100 (hot water).
For iron coils K_o = 160 (steam) and 67 (hot water).
t_m = logarithmic mean of the difference between the temperature of the heating medium and the average water temperature. t_m is approximately = $\left[t_s - \frac{(t_o + t_i)}{2}\right]$

Equation 1 may also be used for determining revised heating coil ratings under different temperature conditions as stated in the manufacturers ratings. When selecting a water heater, the conditions of operation should be carefully considered, as well as the location of the

storage tank and the piping arrangement between the boiler, heater and tank. It is generally good practice to allow a margin of safety when selecting an indirect heater of the proper size to provide for loss of efficiency due to the accumulation of scaling on the coils and piping. Heat exchangers classified according to (3) and (4) may be used with or without a storage tank, but when tanks are omitted, the indirect water heaters should be increased in size so as to heat the water instantaneously as it is needed.

The storage tank should be installed as high as possible. Horizontal tanks are preferable for all medium size installations and absolutely essential on larger installations. Where possible the storage tank should be installed with the bottom of the tank at or above the boiler water line. Horizontal storage tanks smaller than 18 or 20 in. diameter are not

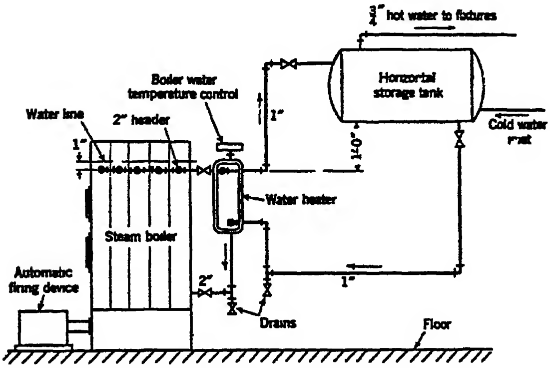


FIG. 9. INDIRECT WATER HEATER MOUNTED ON SIDE OF BOILER

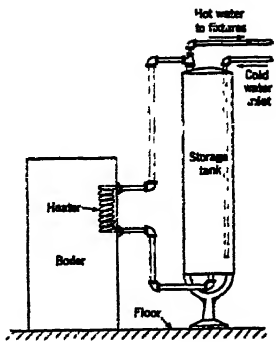


FIG. 10. INDIRECT WATER HEATER PLACED IN BOILER

recommended because of the difficulty of preventing the hot and cold water from mixing, and especially is this an important consideration when large quantities of water are withdrawn.

Pipe sizes between the water heater and boiler should be full size of the heater tappings (Table 12). When a heater is connected to a horizontal sectional boiler, it is recommended that connections be made to all sections and joined together a few inches below the water line as shown in Fig. 8, so that steaming is prevented in those sections which are not connected to the header.

When a steam coil is used for heating the water, an automatic thermostatic valve may be installed in the steam supply to the coil. The operation of this automatic valve is controlled by a thermostat located in the storage tank which permits the proper amount of steam to enter the coil so as to maintain an even water temperature.

An indirect water heater may be used on either a steam or hot water system, and generally this type of heater is provided with a temperature control device located in the boiler water circulating connection to the water heater. The setting on this thermostatic valve may be as low as

140 F or as high as 180 F and may be readily adjusted to meet particular requirements. With this type of control it is impossible to overheat the hot water supply which is an important safety consideration in some installations. This type of system may also be conveniently used during the non-heating season with the operation of the fuel burning device controlled by the water heater thermostat. (See Chapter 14). During the heating season the water heater temperature control functions as a low limit control.

When an indirect water heater is applied to a gravity hot water system, it is necessary to provide a valve in the supply to the heating system to prevent the flow of hot water from the boiler when heat is not required in the house. This valve may be controlled from a room thermostat and the automatic fuel-burning device controlled from the water heater thermostat. To prevent circulation in a forced hot water heating system flow control valves may be installed in the flow and return lines which act merely as check valves when the circulating pump is not operating. In this arrangement the pump is controlled by the room thermostat and the automatic fuel-burning device is controlled from the water heater thermostat.

STORAGE CAPACITY AND BOILER ALLOWANCES

The amount of storage provided in the hot water tank or heater is somewhat a matter of choice but is usually made ample to carry over the peak shortage which is likely to occur and is based on the assumption that only 75 per cent of the storage capacity will be available, as it has been found that if more than this amount is withdrawn from storage, the tank is so cooled down as to make the balance useless. The general rule may be cited that the less the heating capacity the greater must be the storage, and the greater the storage the less may be the heating capacity down to a point where the heating capacity will fail to be sufficient to heat up the tank storage during the periods of small load.

Example 5. A heater to supply 500 persons will have an average daily use of about 500×40 gal $= 20{,}000$ gal and this is an average of $\frac{20{,}000 \text{ gal}}{24} = 833$ gph but the peak hour will require $\frac{1}{10}$ of 20,000 = 2000 gal and the shortage during the peak hour, if the heating capacity is made to suit the average hourly use of 833 gal, will be $2000 - 833 = 1167$ gal so that the storage capacity, based on 75 per cent being available from this capacity without cooling the tank excessively, will be $\frac{1167}{0.75} = 1556$ gal.

Should it be desired to reduce the size of storage tanks and to use a greater heating capacity, it is only necessary to increase the heating capacity to say 1200 gph which then gives $2000 - 1200 = 800$ gal as the shortage during the peak hour, and the necessary storage will be $\frac{800 \text{ gal}}{0.75} = 1067$ gal; or the heating capacity can be increased to 1500 gal, leaving a shortage of $2000 - 1500 = 500$ gal.

Good design requires that the heating capacity be made as small as possible without introducing undesirable amounts of storage, as the heating capacity directly determines the load on the source of heat.

As indicated in Example 5, the heating load is proportional to the heating capacity and the boiler capacity must be increased for higher heating capacities and may be reduced for smaller heating capacities with greater storage. It may be assumed that a boiler capacity of about 4 sq ft of equivalent steam heating surface[3] (radiation) must be provided for every gallon of water heated 100 F or from 50 F to 150 F, which is the temperature rise most commonly assumed and required. On this basis it will be seen that the various conditions cited in Example 5 will require additional boiler capacity as follows:

Heating Capacity (gph)	Additional Boiler Capacity (Sq Ft EDR)
833	3332
1200	4800
1500	6000

From this it is apparent that it is less costly to provide ample storage and to reduce boiler capacity than to diminish the storage and supply a greatly increased boiler capacity to compensate.

The boiler allowance value of 4 sq ft of equivalent steam radiation for each gallon of water heated through a temperature range of 100 F is based on an hourly heating rate. When reduced heating capacities are desired for economic reasons of boiler design and selection, engineers frequently recommend that the heating rate be extended over a period of two hours in which case the boiler allowance value would be reduced to 2 sq ft of equivalent steam radiation. Similarly any other heating rate may be established and a corresponding value of boiler allowance determined.

Reliable information based upon the installations of several heaters in existing heating systems indicates varying arbitrary values of boiler allowances to be used. When these values are selected for usage, a careful analysis of the varying factors involved in determining these values should be considered so that the proper heating allowances may be provided.

ESTIMATING HOT WATER DEMAND BY FIXTURES

In buildings where the occupancy is doubtful and only the number of plumbing fixtures can serve as a basis for determining the probable hot water demand, the problem is not so simple owing to the fact that a fixture gives no information as to how heavy a service may be demanded from the fixture and this amount of service is really the governing factor in making an estimate of the probable hot water demand. Table 13 may prove of some value in this respect as it gives the maximum assumed quantity of hot water per hour which will be demanded of any fixture and then gives a percentage of this amount which may be assumed as probable in different types of buildings. Table 14 gives approximate hot water requirements in various types of buildings.

Example 6. Let it be assumed that an apartment house with 20 apartments has 20 baths, 20 lavatories, 20 kitchen sinks and 20 laundry trays; what is the probable maximum hourly demand for hot water?

---

[3]Actual requirement for 100-deg temperature difference $= \frac{100 \times 8.33}{240} = 3.33$ sq ft per gallon of water heated.

20 Baths at 40 gal and 33 per cent	270 gal
20 Lavs. at 20 gal and 25 per cent	100 gal
20 Sinks at 30 gal and 33 per cent	200 gal
20 Trays at 50 gal and 60 per cent	600 gal
Total	1170 gal
Probable peak use at one time	35 per cent
Probable actual peak demand	409 gph

If three persons are assumed to an apartment the total daily use of hot water should approximate 20 × 3 × 40 gal = 2400 gal and if the peak hour is 10 per cent of this amount, the peak hour by this method shows a probable demand of one-tenth of 2400 gal, which indicates that the values in Table 13 are safe.

SWIMMING POOL HEATING REQUIREMENTS

Swimming pools present a problem of hot water heating demand which is frequently overestimated. Few outdoor swimming pools require water heating, and in some cases they require the addition of cold water to regulate the temperature. The recirculation system of a swimming pool consists of the pumps, hair and lint catchers, and filters together with all necessary pipe connections to the inlets and outlets of the pool. The water heater, the sterilizing equipment and suction cleaner are usually installed or connected to the recirculation system and may be considered as integral parts of the system.

The recirculation system and all its component parts should be designed to provide the required volume of circulation so that the water turnover ratio is at least two times per day and where heavy loads are anticipated the turnover ratio should be increased to three times or more. Many states have regulations prescribing the circulation turnover.

The water heaters for swimming pools are usually instantaneous steam

Table 13. Ordinary Maximum Hourly Demand for Hot Water for Various Fixtures in Gallons and Probable Percentage of Usage

Type of Building	Lavatories		Baths	Showers	Slop Sinks	Kitchen Sinks	Pantry Sinks	Foot Baths	Wash Trays	Av. Max Use[a]
	Private	Public								
Maximum Probable Usage GPH	20	20	40	300	30	30	20	20	50	
	Probable Usage in Per Cent of Maximum Ordinary Use									
Apt. house	25	50	33	67	67	33	50	25	60	35
Club	25	75	50	67	67	67	100	25	80	60
Gym.	25	100	100	100	—	—	—	100	—	80
Hospital	25	75	50	33	67	67	100	25	80	45
Hotel	25	100	50	33	100	67	100	25	80	70
Industrial	25	150	100	100	67	67	—	100	—	90
Laundries	25	100	—	—	33	—	—	—	100	100
Office building	25	75	—	—	50	—	—	—	—	20
Baths	25	150	150	100	50	—	—	—	—	100
Residences	25	—	50	33	50	33	50	50	60	50
Schools	25	75	—	100	67	33	100	50	—	25
Y. M. C. A.	25	100	100	100	67	67	100	100	80	75

[a]Percentage of fixtures likely to be demanding maximum probable usage at any one time.

coil heaters. These heaters should be sized so that they will have sufficient capacity to heat the water delivered by the circulating pump 15 F per hour.

The water temperature in a pool is usually maintained at about 72 F. A few states have regulations prohibiting higher water temperatures than 70 F. The room temperature should be approximately 5 F higher, but not more than 8 F higher nor less than 2 F lower, than the water temperature.

Example 6. Assume a swimming pool 75 ft long, 30 ft wide with an average depth of 6 ft. If the water is to be heated from a temperature of 50 to 65 F, what capacity heater and steam consumption is required with a turnover ratio of two times per day?

Pool volume $75 \times 30 \times 6 \times 7.5 = 100{,}000$ gal.

With a turnover ratio of twice in 24 hr, the heating capacity is: $\dfrac{100{,}000 \times 2}{24} = 8333$ gal per hour

The steam consumption would be $\dfrac{8333 \times 8.33\,(65 - 50)}{970} = 1080$ lb steam per hour.

Regulation of swimming pool temperatures is essential for successful operation and economy. It is therefore recommended that the steam supply to the heater be provided with a by-pass which may be used for pool filling and initial heating and that a smaller by-pass be installed with an automatic control valve having the capacity to heat the circulation water approximately 5 F per hour.

TABLE 14. HOT WATER CONSUMPTION IN VARIOUS TYPES OF BUILDINGS FOR DIFFERENT PURPOSES

TYPE OF BUILDING	CONDITIONS	GALLONS
Hotels	Room with basin only	10 (per day)
	Room with bath	
	(Transient)	40 (per day)
	(Men)	40 (per day)
	(Mixed)	60 (per day)
	(Women)	80 (per day)
	Two-room suite and bath	80 (per day)
	Three-room suite and bath	100 (per day)
Public Buildings	Public bath or lavatory	150 (per day per fixture)
	Public shower	200 (per day per fixture)
	Public lavatory with attendant	200 (per day per fixture)
Industrial Buildings	Per office employee	2 (per day)
	Per factory employee	5 (per day)
	Cleaning floors	3 (per 1000 sq ft per day)
Restaurants	$0.50 Meals	0.5 (per customer with hand washing)
		1.0 (per customer with machine washing)
	$1.00 Meals	1.0 (per customer with hand washing)
		2.0 (per customer with machine washing)
	$1.50 Meals	1.5 (per customer with hand washing)
		4.0 (per customer with machine washing)

PROBLEMS IN PRACTICE

1 ● The heating capacity of an indirect water heater is 100 gal per hour, using steam at 215 F and raising the water from a temperature of 50 to 150 F. Determine the heating capacity of the same water heater using water at a temperature of 180 F for the heating medium.

Using Equation 1, and because the surface area of the water heater is the same for each condition, the two conditions may be equated as follows:

$$\frac{100 \times 8.33\ (150 - 50)}{240\left[215 - \frac{(150 + 50)}{2}\right]} = \frac{Q \times 8.33\ (150 - 50)}{100\left[180 - \frac{(150 + 50)}{2}\right]}$$

Q = 28.98 gal per hour, capacity of heater using water at a temperature of 180 F

2 ● Why is it impractical to size water supply piping so pipe friction will produce an equal pressure on each fixture?

Because the friction would be built up only in periods of maximum flow and at all other times it would be only a fraction of that required

3 ● What is the purpose of zoning water supply systems in tall buildings?

To avoid excessive pressures in the lower stories.

4 ● Define the maximum possible flow, the maximum probable flow, and the average probable flow.

The maximum possible flow is the flow which would occur if all of the outlets on the system were opened at one and the same time. The maximum probable flow is the flow which will occur with probable peak conditions The average probable flow is the flow likely to occur under a normal condition of use.

5 ● What is the factor of usage?

This is the percentage of the maximum possible flow which is likely to occur at peak load

6 ● How many feet higher than the uppermost fixtures must the water line in a house tank be to provide about 15 lb per square inch pressure at the fixture outlet?

Allowing for pipe losses, about 45 ft.

7 ● What methods of hot water circulation commonly are employed with hot water supply systems?

a. Up-feed risers with returns having no connections paralleling the risers.

b. Up-feed risers with returns in other locations, and with connections taken off both supply and return

c. One main up-feed riser, without connections, supplying all down-feed risers for all fixtures.

8 ● Which method of hot water supply generally is the most satisfactory?

The single main up-feed riser supplying drop risers for all fixtures.

9 ● How much of the water stored in a hot water storage tank really is available for use?

About 75 per cent, because when only 25 per cent of the original water remains in the tank it has been so cooled down by the entering water that it is too cold for satisfactory use.

10 ● In cases of intermittent demand, does a large hot water storage tank increase or decrease the steam load for water heating?

It decreases the steam load in cases of intermittent demand but causes no change in the steam load if the demand is constant.

Chapter 36

INSULATION OF PIPING

Heat Losses from Bare Pipes, Steam and Hot Water Lines, Low Temperature Pipe Insulation, Pipe Sweating, Heat Losses from Pipe Surfaces, Economical Pipe Thicknesses, Underground Insulation

PIPE insulation performs an important function in preventing loss of heat where steam or hot water are conveyed from one part of a building to another, and in reducing the absorption of heat by cold pipes as well as preventing condensation on the outer surfaces.

HEAT LOSSES FROM BARE PIPE

Heat losses from horizontal bare iron pipes, based on data obtained from tests conducted at the Mellon Institute, are given in Table 1. These losses are expressed in Btu per hour per linear foot of pipe per degree Fahrenheit difference in temperature between the steam or hot water in the pipe and the air surrounding the pipe. The monetary value of the loss of heat given in Table 1 may be obtained by means of Fig. 1 for various heating system efficiencies, temperature differences, and calorific values and costs of coal. To solve a problem, select the proper heat loss coefficient from Table 1 and locate this value on the upper left hand margin of the chart. Then draw lines in the order indicated by the dotted lines, the dollar value of the heat loss per 100 linear feet of pipe per 1000 hours being given on the upper right hand scale. In using this chart, the cost of coal should also include the labor for handling it, boiler room expense, etc.

Heat losses from horizontal copper pipes based on tests at the A.S.H.V.E. Laboratory, are given in Table 2[1].

In order to determine heat losses per linear foot of pipe from known losses per square foot, it is necessary to know the area in square feet per linear foot of pipe. Table 3 gives these areas for various standard pipe sizes while Table 4 gives the area in square feet for flanges and fittings for various standard pipe sizes.

Very often, even where pipes are thoroughly insulated, flanges and fittings are left bare due to the belief that the losses from these parts are not large. However, the fact that a pair of 8-in. standard flanges having an area of 2.41 sq ft would lose, at 100 lb steam pressure, an amount of

[1]Heat Emission from Iron and Copper Pipe, by F. C. Houghten and Carl Gutberlet (A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932).

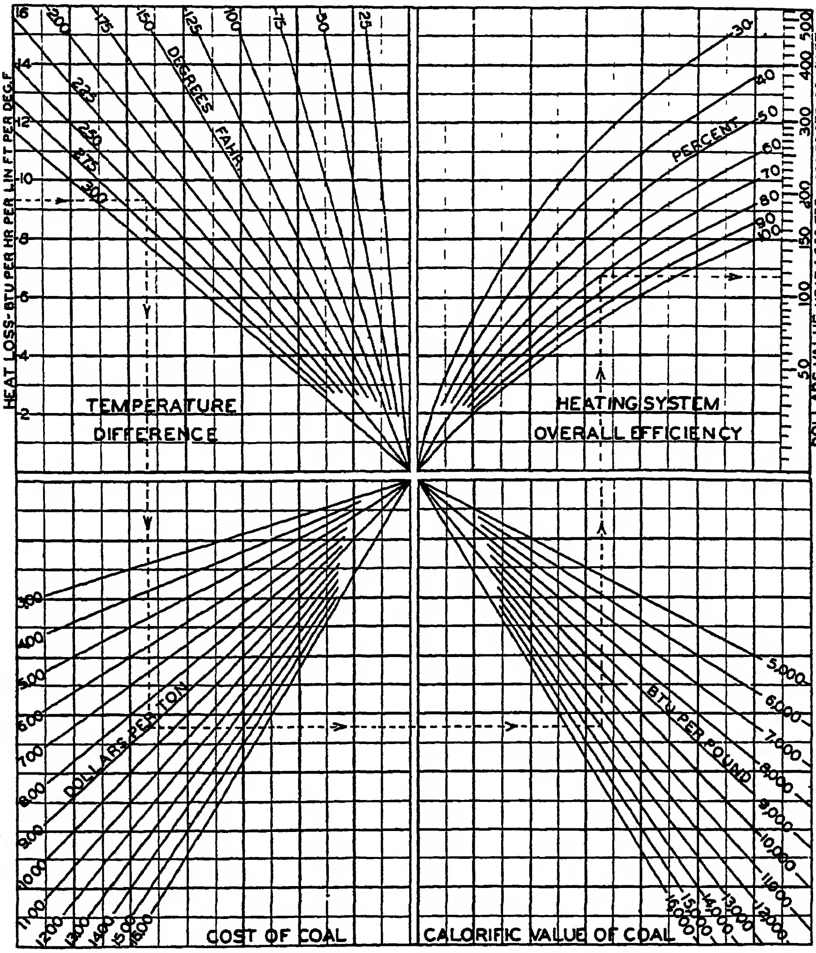


FIG. 1. CHART FOR ESTIMATING DOLLAR VALUE OF HEAT LOSS FROM BARE IRON PIPES. (SEE TABLE 1)[a]

[a]This chart is based on 100 linear feet per 1000 hours For fractions or multiples of these factors, multiply by proper percentage.

heat equivalent to more than a ton of coal per year shows the necessity for insulating such surfaces.

STEAM AND HOT WATER LINES

The conductivities of various materials used for insulating steam and hot water pipes are given in Table 5. In this table the conductivities are given as functions of the mean temperatures or the mean of the inner and

TABLE 1. HEAT LOSSES FROM HORIZONTAL BARE IRON PIPES

Expressed in Btu per linear foot per degree Fahrenheit difference in temperature between the pipe and surrounding still air at 70 F

Nominal Pipe Size (Inches)	Hot Water				Steam		
	120 F	150 F	180 F	210 F	227.1 F (5 Lb)	297.7 F (50 Lb)	337.9 F (100 Lb)
	Temperature Difference						
	50 F	80 F	110 F	140 F	157.1 F	227.7 F	267.9 F
½	0.543	0.573	0.605	0.638	0.656	0.742	0.796
¾	0.660	0.690	0.729	0.762	0.781	0.886	0.955
1	0.791	0.829	0.878	0.920	0.953	1.084	1.166
1¼	0.979	1.02	1.087	1.15	1.184	1.345	1.450
1½	1.09	1.15	1.220	1.29	1.335	1.520	1.640
2	1.34	1.40	1.491	1.58	1.637	1.866	2.015
2½	1.58	1.67	1.778	1.87	1.937	2.215	2.388
3	1.88	1.99	2.100	2.22	2.301	2.641	2.853
3½	2.13	2.24	2.380	2.51	2.585	2.972	3.215
4	2.36	2.50	2.650	2.78	2.873	3.312	3.582
4½	2.60	2.75	2.920	3.08	3.170	3.655	3.956
5	2.87	3.02	3.200	3.38	3.493	4.030	4.368
6	3.39	3.56	3.775	4.01	4.115	4.755	5.153
8	4.32	4.55	5.050	5.14	5.270	6.120	6.635
10	5.32	5.61	5.925	6.34	6.551	7.592	8.245
12	6.25	6.62	6.995	7.46	7.670	8.900	9.670

TABLE 2. HEAT LOSSES FROM HORIZONTAL BARE COPPER PIPES

Expressed in Btu per linear foot per degree Fahrenheit difference in temperature between the pipe at 210 F and surrounding still air at 70 F

Nominal Pipe Size (Inches)	Outside Diameter (Inches)	Temperature Difference 140 F	Nominal Pipe Size (Inches)	Outside Diameter (Inches)	Temperature Difference 140 F
½	0.625	0.405	2½	2.625	0.986
¾	0.875	0.478	3	3.125	1.131
1	1.125	0.550	3½	3.625	1.276
1¼	1.375	0.623	4	4.125	1.422
1½	1.625	0.695	5	5.125	1.713
2	2.125	0.841	6	6.125	2.004

TABLE 3. RADIATING SURFACE PER LINEAR FOOT OF PIPE

Nominal Pipe Size (Inches)	Surface Area (Sq Ft)	Nominal Pipe Size (Inches)	Surface Area (Sq Ft)	Nominal Pipe Size (Inches)	Surface Area (Sq Ft)
½	0.22	2	0.622	5	1.456
¾	0.275	2½	0.753	6	1.734
1	0.344	3	0.917	8	2.257
1¼	0.435	3½	1.047	10	2.817
1½	0.498	4	1.178	12	3.338

TABLE 4. AREAS OF FLANGED FITTINGS, SQUARE FEET[a]

Nominal Pipe Size (Inches)	Flanged Coupling		90 Deg Ell		Long Radius Ell		Tee		Cross	
	Standard	Extra Heavy	Standard	Extra Heavy	Standard	Extra Heavy	Standard	Extra Heavy	Standard	Extra Heavy
1	0 320	0.438	0.795	1.015	0.892	1.083	1.235	1.575	1.622	2.07
1¼	0.383	0.510	0.957	1 098	1.084	1.340	1.481	1.925	1.943	2.53
1½	0.477	0 727	1.174	1.332	1.337	1.874	1.815	2.68	2.38	3.54
2	0.672	0 848	1.65	2 01	1.84	2.16	2.54	3.09	3.32	4.06
2½	0.841	1.107	2.09	2.57	2 32	2.76	3.21	4.05	4.19	5.17
3	0.945	1 484	2.38	3.49	2 68	3.74	3.66	5.33	4.77	6.95
3½	1.122	1.644	2.98	3.96	3.28	4.28	4.48	6.04	5.83	7.89
4	1.344	1.914	3.53	4.64	3.96	4.99	5.41	7.07	7.03	9.24
4½	1.474	2.04	3.95	5.02	4.43	5.46	6.07	7.72	7.87	10.07
5	1.622	2.18	4 44	5.47	5.00	6.02	6.81	8.52	8.82	10.97
6	1.82	2.78	5.13	6.99	5.99	7.76	7.84	10.64	10.08	13.75
8	2.41	3.77	6.98	9.76	8.56	11.09	10.55	14.74	13.44	18.97
10	3.43	5.20	10.18	13.58	12.35	15.60	15.41	20.41	19 58	26.26
12	4 41	6.71	13 08	17.73	16.35	18 76	19.67	26.65	24.87	34 11

[a]Including areas of accompanying flanges bolted to the fitting.

outer surface temperatures of the insulations. This method of stating conductivities makes it possible readily to calculate the heat loss through single or compound sections. It should be emphasized that the conductivities given in Table 5 for the various insulations are the average of values obtained from a number of tests made on each type of material, also that all variables due to differences in thickness, pipe sizes, and air conditions are eliminated. Individual manufacturer's materials will, of course, vary in conductivity to some extent from these values.

The heat losses through six of the types of insulation given in Table 5 for 1, 1½ and 2-in. thick materials, and for temperatures commonly encountered in engineering practice can be obtained from Tables 6 to 11,

TABLE 5 CONDUCTIVITIES (k) OF VARIOUS TYPES OF INSULATING MATERIALS FOR MEDIUM AND HIGH TEMPERATURE PIPES[a]

Types of Insulating Materials	Mean Temperature				
	100 F	200 F	300 F	400 F	500 F
85 per cent Magnesia Type	0 425	0.465	0.505	0.550	0.590
Corrugated Asbestos Type (4 Plies per 1 in. thick)	0 530	0.650	0.770	0.890	
Corrugated Asbestos Type (8 Plies per 1 in. thick)	0.480	0.555	0.630	0.705	
Laminated Asbestos Type (30–40 Laminations per 1 in. thick)	0.360	0.415	0.470	0.525	0.585
Laminated Asbestos Type (20 Laminations per 1 in. thick)	0 545	0.605	0.665	0.725	0.785
Rock Wool Type	0.350	0.410	0.470	0.530	0 590
High Temperature Type (Diatomaceous Earth and Asbestos)	0.515	0.545	0.575	0.605	0.635
Brown Asbestos Type (Felted Fibre)	0.600	0.640	0.675	0.715	0.750

[a]R. H. Heilman, *Mechanical Engineering*, Vol 46 (1924), p. 593.

TABLE 6. COEFFICIENTS OF TRANSMISSION (U) FOR PIPES INSULATED WITH 85 PER CENT MAGNESIA TYPE INSULATION

These coefficients are expressed in Btu per hour per square foot of pipe surface per degree Fahrenheit difference in temperature between pipe and surrounding still air at 70 F

Thickness of Insulation (Inches)	Nominal Pipe Size (Inches)	Hot Water				Steam		
		120 F	150 F	180 F	210 F	227.1 F (5 Lb)	297.7 F (50 Lb)	337.9 F (100 Lb)
		Temperature Difference						
		50 F	80 F	110 F	140 F	157.1 F	227.7 F	267.9 F
	½	0.744	0.754	0.764	0.774	0.779	0.802	0.814
	¾	0.672	0.681	0.689	0.697	0.701	0.721	0.731
	1	0.613	0.621	0.629	0.637	0.641	0.659	0.670
	1¼	0.562	0.570	0.577	0.585	0.589	0.606	0.617
	1½	0.532	0.539	0.546	0.553	0.557	0.573	0.582
	2	0.500	0.506	0.512	0.519	0.523	0.538	0.547
	2½	0.475	0.481	0.487	0.493	0.497	0.512	0.520
	3	0.455	0.461	0.467	0.474	0.477	0.492	0.500
1	3½	0.441	0.447	0.452	0.458	0.462	0.475	0.483
	4	0.429	0.435	0.441	0.446	0.449	0.463	0.471
	4½	0.420	0.425	0.431	0.437	0.440	0.453	0.460
	5	0.411	0.416	0.422	0.427	0.430	0.443	0.450
	6	0.402	0.408	0.413	0.419	0.422	0.435	0.442
	8	0.387	0.392	0.397	0.403	0.405	0.418	0.425
	10	0.375	0.380	0.385	0.390	0.393	0.405	0.412
	12	0.369	0.374	0.378	0.383	0.386	0.398	0.405
	½	0.617	0.625	0.633	0.642	0.646	0.665	0.676
	¾	0.550	0.558	0.566	0.573	0.577	0.596	0.606
	1	0.496	0.503	0.511	0.518	0.522	0.540	0.549
	1¼	0.453	0.459	0.465	0.472	0.475	0.490	0.498
	1½	0.424	0.430	0.436	0.442	0.445	0.459	0.467
	2	0.394	0.400	0.405	0.410	0.413	0.427	0.434
	2½	0.371	0.376	0.382	0.386	0.389	0.401	0.408
	3	0.352	0.357	0.362	0.367	0.370	0.380	0.387
1½	3½	0.339	0.343	0.347	0.351	0.354	0.364	0.370
	4	0.328	0.333	0.337	0.341	0.343	0.353	0.359
	4½	0.320	0.324	0.328	0.332	0.334	0.343	0.350
	5	0.312	0.316	0.320	0.324	0.326	0.336	0.342
	6	0.303	0.307	0.311	0.315	0.318	0.328	0.333
	8	0.287	0.291	0.295	0.299	0.301	0.311	0.316
	10	0.276	0.280	0.284	0.288	0.290	0.299	0.304
	12	0.272	0.275	0.279	0.283	0.285	0.294	0.299
	½	0.543	0.551	0.558	0.565	0.569	0.587	0.597
	¾	0.484	0.490	0.497	0.503	0.507	0.523	0.532
	1	0.433	0.439	0.445	0.451	0.454	0.467	0.476
	1¼	0.393	0.398	0.403	0.409	0.412	0.424	0.432
	1½	0.365	0.370	0.376	0.381	0.384	0.397	0.402
	2	0.338	0.343	0.347	0.351	0.354	0.364	0.370
	2½	0.316	0.320	0.324	0.328	0.331	0.341	0.347
	3	0.297	0.301	0.305	0.309	0.312	0.321	0.326
2	3½	0.284	0.288	0.292	0.295	0.297	0.306	0.311
	4	0.275	0.278	0.282	0.285	0.287	0.296	0.301
	4½	0.266	0.270	0.273	0.276	0.278	0.286	0.290
	5	0.258	0.262	0.265	0.268	0.270	0.278	0.283
	6	0.250	0.254	0.257	0.260	0.262	0.270	0.274
	8	0.236	0.239	0.242	0.245	0.247	0.255	0.258
	10	0.224	0.227	0.230	0.233	0.235	0.242	0.246
	12	0.219	0.222	0.225	0.228	0.230	0.237	0.240

Table 7. Coefficients of Transmission (*U*) for Pipes Insulated with Corrugated Asbestos Type Insulation (4 Plies per Inch Thickness)

These coefficients are expressed in Btu per hour per square foot of pipe surface per degree Fahrenheit difference in temperature between pipe and surrounding still air at 70 F

Thickness of Insulation (Inches)	Nominal Pipe Size (Inches)	Hot Water 120 F	150 F	180 F	210 F	Steam 227.1 F (5 Lb)	297.7 F (50 Lb)	337.9 F (100 Lb)
		Temperature Difference 50 F	80 F	110 F	140 F	157.1 F	227.7 F	267.9 F
1	½	0.890	0.919	0.949	0.978	0.995	1.065	1.106
	¾	0.803	0.829	0.857	0.883	0.898	0.961	0.997
	1	0.731	0.756	0.780	0.804	0.818	0.876	0.909
	1¼	0.671	0.693	0.716	0.738	0.751	0.804	0.834
	1½	0.635	0.656	0.677	0.698	0.710	0.760	0.788
	2	0.595	0.615	0.635	0.656	0.667	0.715	0.742
	2½	0.567	0.586	0.605	0.624	0.635	0.680	0.705
	3	0.544	0.562	0.580	0.598	0.608	0.652	0.677
	3½	0.527	0.544	0.561	0.578	0.588	0.631	0.654
	4	0.513	0.530	0.548	0.565	0.575	0.616	0.639
	4½	0.502	0.518	0.535	0.551	0.561	0.601	0.624
	5	0.490	0.507	0.523	0.539	0.549	0.588	0.611
	6	0.480	0.496	0.512	0.528	0.538	0.577	0.599
	8	0.462	0.477	0.493	0.508	0.517	0.554	0.575
	10	0.447	0.462	0.476	0.491	0.500	0.537	0.557
	12	0.441	0.456	0.470	0.485	0.493	0.529	0.550
1½	½	0.737	0.762	0.787	0.812	0.826	0.884	0.918
	¾	0.657	0.679	0.702	0.725	0.737	0.790	0.820
	1	0.594	0.614	0.634	0.654	0.666	0.713	0.740
	1¼	0.542	0.559	0.577	0.596	0.606	0.649	0.673
	1½	0.507	0.524	0.541	0.558	0.568	0.609	0.632
	2	0.471	0.487	0.503	0.519	0.528	0.565	0.587
	2½	0.443	0.458	0.473	0.488	0.497	0.533	0.553
	3	0.421	0.435	0.449	0.463	0.472	0.506	0.525
	3½	0.403	0.417	0.430	0.443	0.451	0.483	0.502
	4	0.393	0.405	0.418	0.432	0.439	0.471	0.489
	4½	0.383	0.394	0.407	0.420	0.428	0.460	0.476
	5	0.372	0.384	0.397	0.409	0.417	0.447	0.463
	6	0.362	0.374	0.387	0.399	0.406	0.436	0.452
	8	0.343	0.354	0.366	0.378	0.385	0.413	0.429
	10	0.328	0.339	0.351	0.362	0.369	0.397	0.413
	12	0.323	0.334	0.346	0.357	0.364	0.391	0.407
2	½	0.648	0.670	0.692	0.713	0.726	0.779	0.810
	¾	0.578	0.598	0.617	0.637	0.648	0.694	0.720
	1	0.518	0.535	0.552	0.570	0.580	0.622	0.645
	1¼	0.469	0.485	0.501	0.517	0.527	0.566	0.587
	1½	0.438	0.452	0.467	0.481	0.490	0.526	0.545
	2	0.404	0.417	0.430	0.444	0.452	0.483	0.502
	2½	0.379	0.391	0.403	0.415	0.422	0.451	0.466
	3	0.356	0.367	0.378	0.390	0.397	0.425	0.440
	3½	0.339	0.350	0.361	0.373	0.380	0.406	0.421
	4	0.328	0.339	0.350	0.360	0.367	0.392	0.406
	4½	0.318	0.328	0.339	0.350	0.357	0.381	0.395
	5	0.308	0.318	0.329	0.340	0.346	0.370	0.384
	6	0.299	0.309	0.319	0.329	0.335	0.358	0.371
	8	0.282	0.291	0.301	0.310	0.315	0.336	0.349
	10	0.267	0.276	0.285	0.294	0.299	0.319	0.332
	12	0.263	0.272	0.280	0.289	0.294	0.314	0.325

Table 8 Coefficients of Transmission (U) for Pipes Insulated with Corrugated Asbestos Type Insulation (8 Plies per Inch Thickness)

These coefficients are expressed in Btu per hour per square foot of pipe surface per degree Fahrenheit difference in temperature between pipe and surrounding still air at 70 F

Thickness of Insulation (Inches)	Nominal Pipe Size (Inches)	Hot Water				Steam		
		120 F	150 F	180 F	210 F	227.1 F (5 Lb)	297.7 F (50 Lb)	337.9 F (100 Lb)
		Temperature Difference						
		50 F	80 F	110 F	140 F	157.1 F	227.7 F	267.9 F
1	½	0.801	0.820	0.838	0.857	0.868	0.913	0.939
	¾	0.723	0.739	0.756	0.773	0.783	0.824	0.847
	1	0.658	0.673	0.688	0.704	0.713	0.751	0.772
	1¼	0.606	0.619	0.633	0.647	0.655	0.688	0.707
	1½	0.573	0.586	0.599	0.612	0.619	0.652	0.670
	2	0.538	0.550	0.562	0.575	0.581	0.612	0.629
	2½	0.511	0.523	0.534	0.546	0.553	0.582	0.599
	3	0.489	0.501	0.512	0.524	0.531	0.558	0.575
	3½	0.474	0.485	0.496	0.507	0.514	0.542	0.557
	4	0.461	0.472	0.482	0.493	0.500	0.527	0.542
	4½	0.451	0.462	0.472	0.482	0.489	0.515	0.530
	5	0.442	0.452	0.462	0.473	0.479	0.505	0.520
	6	0.432	0.442	0.452	0.463	0.468	0.493	0.508
	8	0.416	0.426	0.436	0.446	0.451	0.475	0.489
	10	0.402	0.412	0.421	0.430	0.435	0.459	0.473
	12	0.397	0.406	0.415	0.424	0.429	0.452	0.466
1½	½	0.664	0.679	0.695	0.711	0.720	0.759	0.780
	¾	0.593	0.607	0.621	0.636	0.643	0.677	0.697
	1	0.535	0.547	0.560	0.573	0.580	0.611	0.629
	1¼	0.488	0.499	0.510	0.522	0.528	0.556	0.572
	1½	0.457	0.467	0.478	0.490	0.496	0.522	0.537
	2	0.425	0.434	0.444	0.455	0.460	0.485	0.499
	2½	0.399	0.408	0.418	0.428	0.434	0.457	0.471
	3	0.378	0.387	0.396	0.405	0.411	0.433	0.446
	3½	0.363	0.371	0.380	0.388	0.393	0.415	0.427
	4	0.353	0.361	0.369	0.378	0.383	0.403	0.415
	4½	0.343	0.351	0.360	0.368	0.373	0.393	0.404
	5	0.334	0.342	0.350	0.358	0.363	0.383	0.394
	6	0.325	0.333	0.341	0.349	0.353	0.373	0.383
	8	0.309	0.316	0.324	0.332	0.336	0.355	0.365
	10	0.295	0.303	0.310	0.318	0.322	0.340	0.350
	12	0.291	0.298	0.306	0.313	0.317	0.335	0.344
2	½	0.585	0.599	0.613	0.627	0.635	0.668	0.688
	¾	0.520	0.533	0.545	0.558	0.565	0.595	0.612
	1	0.465	0.476	0.487	0.498	0.504	0.532	0.547
	1¼	0.422	0.432	0.442	0.452	0.458	0.483	0.497
	1½	0.394	0.403	0.412	0.422	0.427	0.450	0.462
	2	0.364	0.372	0.380	0.388	0.393	0.415	0.427
	2½	0.339	0.347	0.355	0.363	0.367	0.387	0.398
	3	0.319	0.327	0.334	0.342	0.346	0.365	0.375
	3½	0.304	0.311	0.318	0.326	0.330	0.349	0.358
	4	0.295	0.302	0.308	0.315	0.319	0.336	0.345
	4½	0.285	0.292	0.299	0.306	0.310	0.327	0.336
	5	0.278	0.284	0.290	0.297	0.301	0.317	0.326
	6	0.269	0.275	0.282	0.288	0.292	0.307	0.315
	8	0.253	0.259	0.265	0.270	0.273	0.288	0.296
	10	0.240	0.245	0.251	0.257	0.260	0.275	0.282
	12	0.236	0.241	0.247	0.253	0.256	0.270	0.277

Table 9. Coefficients of Transmission (U) for Pipes Insulated with Laminated Asbestos Type Insulation (30 to 40 Laminations per Inch Thickness)

These coefficients are expressed in Btu per hour per square foot of pipe surface per degree Fahrenheit difference in temperature between pipe and surrounding still air at 70 F

Thickness of Insulation (Inches)	Nominal Pipe Size (Inches)	Hot Water				Steam		
		120 F	150 F	180 F	210 F	227.1 F (5 Lb)	297.7 F (50 Lb)	337.9 F (100 Lb)
		Temperature Difference						
		50 F	80 F	110 F	140 F	157.1 F	227.7 F	267.9 F
1	½	0.605	0.620	0.635	0.650	0.658	0.695	0.716
	¾	0.546	0.560	0.573	0.586	0.594	0.627	0.645
	1	0.498	0.510	0.522	0.534	0.541	0.570	0.587
	1¼	0.457	0.468	0.480	0.491	0.497	0.525	0.540
	1½	0.432	0.442	0.453	0.464	0.470	0.496	0.511
	2	0.406	0.416	0.426	0.437	0.442	0.467	0.481
	2½	0.385	0.395	0.405	0.415	0.420	0.443	0.457
	3	0.370	0.379	0.389	0.398	0.403	0.425	0.438
	3½	0.359	0.367	0.376	0.385	0.390	0.413	0.426
	4	0.349	0.358	0.366	0.375	0.380	0.402	0.414
	4½	0.341	0.350	0.359	0.367	0.372	0.393	0.405
	5	0.334	0.342	0.351	0.359	0.364	0.384	0.395
	6	0.327	0.335	0.343	0.351	0.356	0.376	0.387
	8	0.314	0.322	0.330	0.338	0.343	0.362	0.373
	10	0.304	0.312	0.320	0.328	0.332	0.350	0.361
	12	0.301	0.308	0.316	0.324	0.328	0.346	0.356
1½	½	0.502	0.514	0.526	0.539	0.546	0.577	0.595
	¾	0.450	0.461	0.473	0.484	0.490	0.517	0.532
	1	0.405	0.415	0.426	0.436	0.442	0.466	0.480
	1¼	0.369	0.378	0.387	0.396	0.401	0.423	0.435
	1½	0.343	0.352	0.361	0.370	0.375	0.397	0.409
	2	0.321	0.329	0.337	0.345	0.350	0.369	0.380
	2½	0.301	0.309	0.317	0.324	0.330	0.348	0.358
	3	0.286	0.293	0.301	0.308	0.313	0.330	0.340
	3½	0.274	0.281	0.288	0.295	0.300	0.316	0.326
	4	0.267	0.273	0.280	0.287	0.291	0.307	0.317
	4½	0.259	0.266	0.272	0.279	0.283	0.299	0.308
	5	0.253	0.260	0.266	0.272	0.276	0.291	0.300
	6	0.247	0.253	0.260	0.266	0.269	0.284	0.293
	8	0.234	0.240	0.246	0.252	0.255	0.270	0.279
	10	0.223	0.229	0.235	0.241	0.245	0.258	0.266
	12	0.221	0.227	0.232	0.238	0.241	0.255	0.263
2	½	0.442	0.453	0.464	0.475	0.481	0.508	0.523
	¾	0.392	0.402	0.412	0.422	0.428	0.452	0.465
	1	0.352	0.360	0.369	0.378	0.383	0.405	0.417
	1¼	0.319	0.327	0.335	0.343	0.348	0.367	0.379
	1½	0.297	0.304	0.311	0.319	0.323	0.341	0.352
	2	0.274	0.280	0.287	0.294	0.298	0.314	0.324
	2½	0.256	0.262	0.269	0.275	0.279	0.293	0.302
	3	0.243	0.249	0.254	0.260	0.264	0.277	0.285
	3½	0.231	0.236	0.242	0.248	0.251	0.265	0.273
	4	0.223	0.228	0.234	0.240	0.243	0.257	0.265
	4½	0.216	0.222	0.227	0.233	0.236	0.249	0.256
	5	0.210	0.215	0.220	0.225	0.228	0.241	0.248
	6	0.203	0.208	0.213	0.218	0.221	0.233	0.240
	8	0.191	0.196	0.201	0.206	0.209	0.220	0.227
	10	0.182	0.187	0.192	0.196	0.199	0.210	0.215
	12	0.178	0.183	0.187	0.192	0.195	0.205	0.210

Table 10. Coefficients of Transmission (U) for Pipes Insulated with Laminated Asbestos Type Insulation (Approximately 20 Laminations per Inch Thickness)

These coefficients are expressed in Btu per hour per square foot of pipe surface per degree Fahrenheit difference in temperature between pipe and surrounding still air at 70 F

Thickness of Insulation (Inches)	Nominal Pipe Size (Inches)	Hot Water				Steam		
		120 F	150 F	180 F	210 F	227.1 F (5 Lb)	297.7 F (50 Lb)	337.9 F (100 Lb)
		Temperature Difference						
		50 F	80 F	110 F	140 F	157.1 F	227.7 F	267.9 F
1	½	0.910	0.925	0.940	0.956	0.964	1.001	1.022
	¾	0.823	0.836	0.850	0.863	0.871	0.902	0.921
	1	0.748	0.760	0.773	0.785	0.792	0.823	0.840
	1¼	0.686	0.698	0.710	0.721	0.728	0.756	0.771
	1½	0.649	0.659	0.671	0.682	0.688	0.716	0.731
	2	0.610	0.620	0.630	0.640	0.647	0.671	0.685
	2½	0.581	0.590	0.600	0.609	0.615	0.638	0.651
	3	0.558	0.567	0.576	0.585	0.591	0.613	0.626
	3½	0.539	0.548	0.557	0.566	0.571	0.592	0.604
	4	0.524	0.532	0.541	0.551	0.556	0.577	0.589
	4½	0.514	0.522	0.530	0.539	0.544	0.564	0.575
	5	0.503	0.511	0.519	0.528	0.533	0.553	0.565
	6	0.492	0.500	0.509	0.517	0.522	0.542	0.553
	8	0.473	0.480	0.488	0.497	0.502	0.521	0.532
	10	0.458	0.465	0.473	0.481	0.485	0.504	0.514
	12	0.452	0.459	0.467	0.475	0.478	0.497	0.507
1½	½	0.755	0.767	0.780	0.793	0.800	0.831	0.848
	¾	0.674	0.685	0.697	0.708	0.715	0.743	0.759
	1	0.607	0.618	0.628	0.639	0.645	0.670	0.684
	1¼	0.553	0.562	0.572	0.581	0.587	0.610	0.622
	1½	0.517	0.527	0.536	0.545	0.550	0.572	0.584
	2	0.481	0.490	0.499	0.508	0.513	0.535	0.547
	2½	0.453	0.460	0.469	0.477	0.481	0.500	0.511
	3	0.429	0.436	0.444	0.452	0.456	0.475	0.485
	3½	0.412	0.419	0.427	0.434	0.438	0.456	0.465
	4	0.400	0.407	0.415	0.422	0.426	0.443	0.453
	4½	0.390	0.396	0.402	0.409	0.413	0.429	0.437
	5	0.380	0.386	0.393	0.400	0.403	0.418	0.427
	6	0.369	0.375	0.382	0.389	0.392	0.408	0.417
	8	0.351	0.358	0.364	0.370	0.374	0.388	0.397
	10	0.337	0.344	0.350	0.356	0.359	0.373	0.382
	12	0.332	0.338	0.344	0.350	0.353	0.367	0.375
2	½	0.664	0.675	0.687	0.698	0.704	0.732	0.747
	¾	0.591	0.601	0.611	0.621	0.627	0.652	0.665
	1	0.529	0.538	0.547	0.557	0.562	0.584	0.597
	1¼	0.480	0.488	0.497	0.505	0.510	0.529	0.540
	1½	0.445	0.453	0.462	0.470	0.475	0.494	0.504
	2	0.412	0.420	0.427	0.434	0.438	0.455	0.464
	2½	0.385	0.392	0.398	0.405	0.409	0.425	0.434
	3	0.364	0.370	0.376	0.382	0.385	0.400	0.408
	3½	0.346	0.352	0.358	0.365	0.368	0.382	0.390
	4	0.336	0.342	0.348	0.354	0.357	0.371	0.378
	4½	0.325	0.332	0.338	0.343	0.346	0.360	0.367
	5	0.316	0.322	0.327	0.333	0.336	0.349	0.356
	6	0.306	0.312	0.317	0.323	0.326	0.338	0.345
	8	0.288	0.293	0.298	0.303	0.306	0.317	0.324
	10	0.275	0.279	0.284	0.289	0.292	0.302	0.308
	12	0.269	0.274	0.278	0.283	0.286	0.296	0.302

Table 11. Coefficients of Transmission (*U*) for Pipes Insulated with Rock Wool Type Insulation

These coefficients are expressed in Btu per hour per square foot of pipe surface per degree Fahrenheit difference in temperature between pipe and surrounding still air at 70 F

Thickness of Insulation (Inches)	Nominal Pipe Size (Inches)	Hot Water				Steam		
		120 F	150 F	180 F	210 F	227 1 F (5 Lb)	297 7 F (50 Lb)	337 9 F (100 Lb)
		Temperature Difference						
		50 F	80 F	110 F	140 F	157 1 F	227 7 F	267 9 F
	½	0.631	0 644	0.658	0 672	0.680	0 712	0 730
	¾	0 569	0.581	0.593	0 606	0.613	0.642	0 659
	1	0.518	0 529	0.541	0.552	0.559	0.585	0 600
	1¼	0.476	0.486	0.497	0.507	0.513	0.537	0.551
	1½	0.450	0.460	0.470	0.480	0.485	0.508	0.522
	2	0 422	0.431	0.441	0.450	0.456	0.478	0.490
	2½	0 402	0.411	0.420	0 428	0.434	0.455	0.466
	3	0 385	0.394	0.402	0.411	0.415	0.435	0.446
1	3½	0.373	0.381	0.389	0.398	0.402	0.421	0.432
	4	0 363	0.371	0.379	0 387	0 392	0.411	0.422
	4½	0.355	0.363	0.371	0.379	0 383	0.402	0.413
	5	0 348	0.356	0.364	0.371	0.376	0.394	0.404
	6	0 341	0 348	0.356	0.363	0.368	0.386	0.396
	8	0.327	0.335	0.342	0.349	0.353	0.372	0.381
	10	0.317	0.324	0.331	0.338	0.343	0 360	0.369
	12	0 313	0.320	0.327	0.334	0.338	0.355	0.364
	½	0.523	0.534	0.545	0.556	0.563	0 590	0 606
	¾	0 468	0 477	0.487	0.497	0.503	0.528	0.542
	1	0.421	0.430	0.440	0.449	0 455	0.477	0.490
	1¼	0 383	0.391	0.399	0.407	0.412	0 433	0.444
	1½	0.359	0.366	0.375	0.383	0 387	0.407	0.419
	2	0.333	0.340	0.348	0.356	0 360	0 378	0.389
	2½	0.314	0.320	0.327	0.335	0.339	0.355	0.365
	3	0.296	0.302	0.310	0.317	0 321	0.337	0.347
1½	3½	0 286	0.291	0.298	0.304	0.307	0.323	0.332
	4	0.278	0.284	0.290	0.296	0 300	0.315	0.323
	4½	0.270	0 276	0.282	0.287	0 291	0.305	0.313
	5	0.263	0.269	0.275	0.280	0.284	0.298	0.305
	6	0.257	0.262	0.267	0.273	0.277	0.290	0.297
	8	0.244	0.249	0.254	0.260	0 263	0 276	0.283
	10	0 235	0 240	0.245	0.250	0.253	0.265	0.272
	12	0.230	0.234	0.239	0 245	0 247	0.260	0.267
	½	0.461	0.471	0.481	0.491	0 496	0.520	0.534
	¾	0 409	0.418	0.427	0.436	0 441	0.463	0.475
	1	0.366	0.374	0.382	0.390	0.395	0 415	0.427
	1¼	0.333	0.340	0.347	0.355	0.359	0.377	0.387
	1½	0.310	0.316	0.323	0.330	0.334	0.351	0.360
	2	0.286	0.292	0.298	0.304	0.308	0.323	0.331
	2½	0.268	0.274	0.279	0.285	0 289	0.302	0.310
	3	0.252	0.257	0.262	0.268	0.272	0.284	0.292
2	3½	0.241	0.246	0.251	0.257	0.260	0.272	0.280
	4	0.232	0.237	0.242	0.247	0 250	0.262	0.269
	4½	0.225	0 230	0.235	0.240	0.243	0.255	0.262
	5	0.218	0.223	0.228	0.233	0.236	0.247	0.253
	6	0.213	0.217	0.221	0.226	0 228	0.239	0.245
	8	0.200	0.204	0.208	0.213	0.215	0.225	0.231
	10	0.189	0.193	0.197	0.201	0.204	0.214	0.220
	12	0.185	0.190	0.194	0.198	0 200	0.210	0.216

inclusive. The loss through other thicknesses of the materials, and for other hot water or steam temperature conditions may be obtained by interpolation. The heat loss coefficients given in Tables 6 to 11 are based on the conductivities in Table 5 and were computed from data given in Chapter 22, THE GUIDE 1931.

LOW TEMPERATURE PIPE INSULATION

Surfaces maintained at low temperatures should be insulated so as to retard the flow of heat from the outside into the low temperature area and to prevent the formation of condensation and of frost if the temperatures are low enough, as well as to prevent corrosion induced by the presence of condensed moisture on metal surfaces. Materials commonly used for insulating pipes and surfaces at low temperatures are cork, rock cork, hair felt and other felted or fibrous non-absorbent materials. Thermal conductivities of low temperature insulating materials are given in Chapter 5.

Insulating materials are available commercially to meet varying temperature gradients. For example, the thickness of insulation for ice water is approximately 1½ in. if the temperature in the line is not lower than 25 F; the thickness of insulation for brine is approximately 2½ in. where the temperature ranges from 0 deg to 25 F; and the thickness of insulation where the brine temperature ranges from −30 F to 0 deg is approximately 4 in.

Insulation to Prevent Freezing

If the surrounding air temperature remains sufficiently low for an ample period of time, insulation cannot prevent the freezing of still water, or of water flowing at such a velocity that the quantity of heat carried in the water is not sufficient to take care of the heat losses which will result and cause the temperature of the water to be lowered to the freezing point. Insulation can materially prolong the time required for the water to give up its heat, and if the velocity of the water flowing in the pipe is maintained at a sufficiently high rate, freezing may be prevented.

Table 12 may be used for making estimates of the thickness of insulation necessary to take care of still water in pipes at various water and surrounding air temperature conditions. Because of the damage and service interruptions which may result from frozen water in pipes, it is essential that the most efficient insulation be utilized. This table is based on the use of hair felt or cork, having a conductivity of 0.30. The initial water temperature is assumed to be 10 deg above, and the surrounding air temperature 50 deg below the freezing point of water (temperature difference, 60 F).

The last column of Table 12 gives the minimum quantity of water at initial temperature of 42 F which should be supplied every hour for each linear foot of pipe, in order to prevent the temperature of the water from being lowered to the freezing point. The weights given in this column should be multiplied by the total length of the exposed pipe line expressed in feet. As an additional factor of safety, and in order to provide against temporary reductions in flow occasioned by reduced pressure, it is advisable to double the rates of flow listed in the table. It must be

emphasized that the flow rates and periods of time designated apply only for the conditions stated. To estimate for other service conditions the following method of procedure may be used.

If water enters the pipe at 52 F instead of 42 F, the time required to cool it to the freezing point will be prolonged to twice that given in the table, or the rate of flow of water may be reduced so that the quantity required will be one-half that shown in the last column of Table 12. However, if the water enters the pipe at 34 F it will be cooled to 32 F in one-fifth of the time given in the table. It will then be necessary to increase the rate of flow so that five times the specified quantity of water will have to be supplied in order to prevent freezing.

If the minimum air temperature is −38 F (temperature difference, 80 F), instead of −18 F, the time required to cool the water to the freezing point will be 60/80 of the time given in the table, or the necessary quantity of water to be supplied will be 80/60 of that given.

TABLE 12. DATA FOR ESTIMATING REQUIREMENTS TO PREVENT FREEZING OF WATER IN PIPES

Nominal Pipe Size (Inches)	Number of Hours to Cool Water to Freezing Point			Water Required to Flow to Prevent Freezing, Pounds per Linear Foot of Pipe per Hour		
	Thickness of Insulation in Inches					
	1	2	3	1	2	3
½	0.42	0.50	0.57	0.54	0.45	0.40
1	0.83	1.02	1.16	0.68	0.55	0.48
1½	1.40	1.74	2.02	0.84	0.68	0.58
2	1.94	2.48	2.90	0.95	0.75	0.64
3	3.25	4.27	5.08	1.24	0 94	0.79
4	4.55	6.02	7.20	1.47	1.11	0.93
5	5 92	7.96	9.69	1.73	1.29	1 06
6	7.35	9.88	12.20	1.98	1.46	1.19
8	10.05	13.90	17.25	2.46	1.78	1.44
10	13.00	18.10	22.70	2.96	2.12	1.70
12	15.80	22.20	28.10	3.43	2.46	1.93

In making calculations to arrive at the values given in Table 12, the loss of heat stored in the insulation, the effect of a varying temperature difference due to the cooling of pipe and water, and the resistance of the outer surface of the insulation to the transfer of heat to the air have all been neglected. When these factors enter into the computations it is necessary to enlarge the factor of safety. Also as stated, the time shown in the table is that required to lower the water to the freezing point. A longer period would be required to freeze the water, but the danger point is reached when freezing starts. The flow of water will stop and the entire line will be in danger as soon as the water freezes across the section of the pipe at any point.

When water must remain stationary longer than the times designated in Table 12, the only safe way to insure against freezing is to install a steam or hot water line, or to place an electric resistance heater along the side of the exposed water line. The heating system and the water line are then

insulated so that the heat losses from the heating system are not excessive, and the heating effect is concentrated against the water pipe where it is needed. For this form of protection 2 in. of an efficient insulation may be applied.

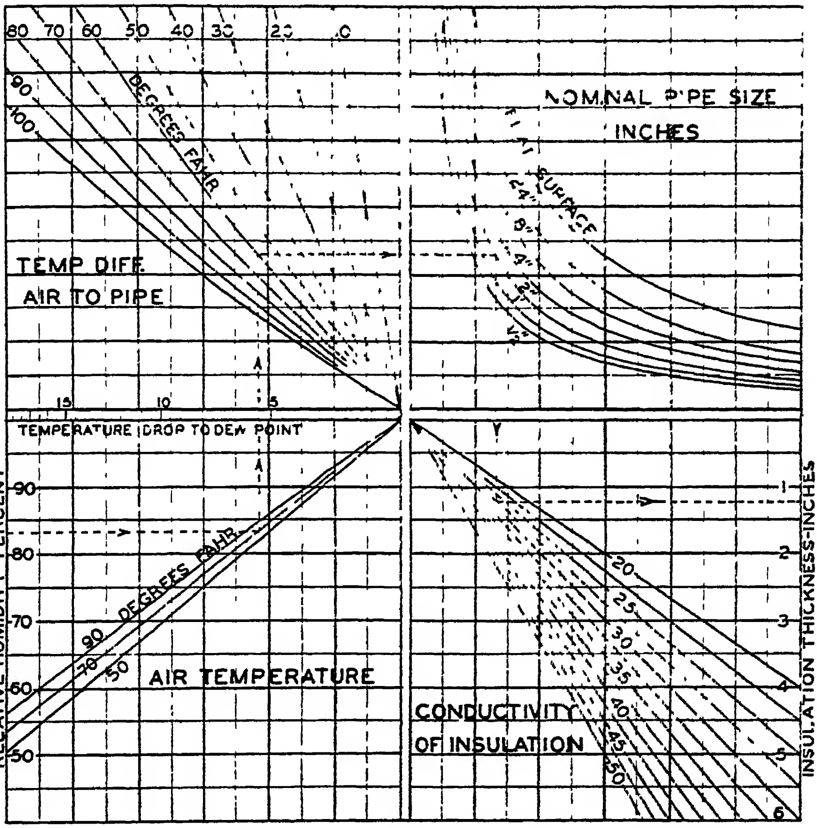
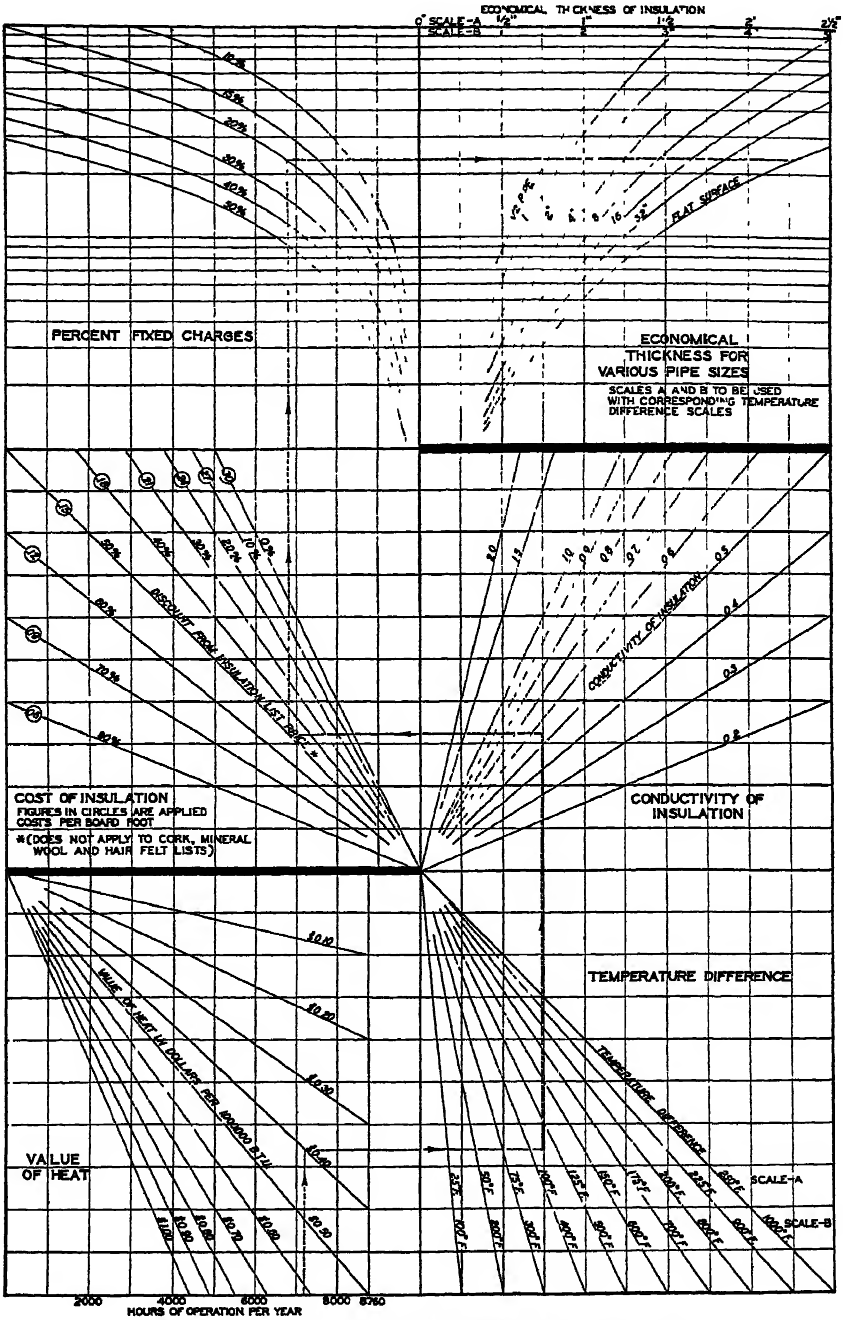


FIG. 2. THICKNESS OF PIPE INSULATION TO PREVENT SWEATING[a]

[a]Solve problems by drawing lines as indicated by dotted line, entering chart at lower left hand scale.

Pipe Sweating

In some cases the prevention of condensation rather than the conservation of heat is the governing factor in determining the thickness of insulation required. Fig. 2 may be used for determining the thickness of any material of known conductivity which should be used to prevent condensation on pipes and flat metallic surfaces. The surface resistances used for calculating the family of curves in Fig. 2 are based on the results of tests made on canvas-covered pipe insulation surfaces at Mellon Institute



(L. B. McMillan, *Proc. National Dist. Heating A'ss'n*, Vol 18, p 138)

FIG. 3. CHART FOR DETERMINING ECONOMICAL THICKNESS OF INSULATION

However, it has been found that the resistance for asphaltic and roofing surfaces is practically the same as for canvas surfaces, so that the curves given may be followed with no alteration for surfaces commonly used.

Moisture will be deposited on a surface whenever its temperature falls to that of the dew point. The maximum permissible temperature drop is indicated on Fig. 2 at the point where the guide line passes through the horizontal scale at the left center of the chart. This temperature drop represents the difference between the dry-bulb temperature and the dew-point temperature for the conditions involved. (See discussion of condensation in Chapter 7.)

The rate of heat loss from a surface maintained at constant temperature is greatly increased by air circulation over the surface. In the case of well-insulated surfaces the increases in losses due to air velocity are very small as compared with increases shown for bare surfaces, because of the fact that air flowing over the surface of the insulation can increase only the rate of heat transfer from surface to air, and cannot change the internal resistance to heat flow inherent in the insulation itself. The maximum increase in loss due to air velocity ranges from about 30 per cent in the case of 1-in. thick insulation, to about 10 per cent in the case of 3-in. thick insulation, provided that the insulation is thoroughly sealed so that air can flow only over the surface.

If the conditions are such that the air may circulate through cracks and crevices in the insulation, the increases may be far greater than those given. Therefore, it is essential that insulation be sealed as tightly as possible. Pipe insulation out-of-doors should be provided with a waterproof jacket, and other outdoor insulation should be thoroughly weatherproofed.

ECONOMICAL PIPE THICKNESS

Table 13 shows the thicknesses of insulation which ordinarily are used for various temperature conditions. Where a thorough analysis of economic thickness is desired, this may be accomplished through the use of the chart, Fig. 3.

The dotted line on the chart illustrates its use in solving a typical example. In using the chart, start with the scale at the left bottom margin representing the given number of hours of operation per year; then proceed vertically to the line representing the given value of heat; thence horizontally, to the right, to the line representing the given temperature difference; thence vertically to the line representing the conductivity of the given material; thence horizontally, to the left, to the line representing the given discount on that material; thence vertically to the curve representing the required per cent return on the investment; thence horizontally, to the right, to the curve representing the given pipe size; thence vertically to the scale at the top right margin where the economical thickness may be read off directly.

Underground Insulation

Underground steam distribution lines are carried in protective structures of various types, sizes and shapes. (See Chapter 37.) Detailed

TABLE 13 THICKNESSES OF INSULATION ORDINARILY USED INDOORS[a]

Steam Pressures (Lb Gage) or Conditions	Steam Temperatures Degrees Fahrenheit	Thickness of Insulation		
		Pipes Larger Than 4 In.	Pipes 2 In. to 4 In.	Pipes ½ In. to 1½ In.
0 to 25	212 to 267	1 in.	1 in.	1 in
25 to 100	267 to 338	1½ in.	1 in.	1 in
100 to 200	338 to 388	2 in	1½ in	1 in.
Low Superheat	388 to 500	2½ in.	2 in.	1½ in.
Medium Superheat	500 to 600	3 in.	2½ in.	2 in.
High Superheat	600 to 700	3½ in.	3 in	2 in

[a]All piping located outdoors or exposed to weather is ordinarily insulated to a thickness ½ in greater than shown in this table, and covered with a waterproof jacket

TABLE 14. THICKNESS OF LOOSE INSULATION FOR USE AS FILL IN UNDERGROUND CONDUIT SYSTEMS

Steam Pressures (Lb Gage) or Conditions	Steam Temperatures Degrees Fahrenheit	Minimum Thickness of Insulation in Inches					Minimum Distance Between Steam and Return
		Steam Lines			Return Lines		
		Pipes Less than 4 In	Pipes 4 In to 10 In.	Pipes Larger than 12 In	Pipes Less than 4 In.	Pipes 4 In and Larger	
Hot Water, or 0 to 25	212 to 267	1½	2	2½	1¼	1½	1
25 to 125	267 to 352	2	2½	3	1¼	1½	1¼
Above 125, or superheat	352 to 500	2½	3	3½	1¼	1½	1½

data on commonly used forms of tunnels and conduit systems have been published by the *National District Heating Association*[2].

Pipes in tunnels are covered with sectional insulation to provide maximum thermal efficiency and are also finished with good mechanical protection in the form of metal or waterproofing membrane outer jackets. Conduit systems are in more general use than tunnels. Pipes carried in conduits may be insulated with sectional insulation; however, the more usual practice is to fill the entire section of the conduit around the pipes with high quality, loose insulating material. The insulation must be kept dry at all times, and for this purpose effective waterproofing membranes enclose the insulation. A drainage system is also provided to divert water which may tend to enter the conduit.

The economical thickness of insulation for underground work is difficult of accurate determination due to the many variables which have to be considered. As a result of theories[3] previously developed, together with other experimental data which has been presented, the usual endeavor is to secure not less than 90 per cent efficiency for underground piping.

[2]*Handbook of the National District Heating Association*, Second Edition, 1932

[3]Theory of Heat Losses from Pipes Buried in the Ground, by J. R. Allen (A S H V.E. TRANSACTIONS, Vol 26, 1920)

Table 14 can be used as a guide in arriving at the minimum thickness of loose insulation fills to use for laying out conduit systems. Other factors such as the number of pipes and their combination of sizes, as well as the standard conduit sizes, are primary controlling factors in the amount and thickness of insulation for use.

When sectional insulation is applied to lines in tunnels or conduits, usual practice is to apply the most efficient materials ½ in. less in thickness than that determined by the use of Fig. 3. Fig. 3 is based on conditions of insulation exposed to the air, whereas normal ground temperature is substituted for air temperature in determining the temperature difference for use with the chart when applying it for underground pipe line estimates.

PROBLEMS IN PRACTICE

1 ● What precautions must be taken in selecting insulation used for covering pipe lines carrying materials at temperatures lower than the dew point?

Materials intended for this service should be as moisture proof as possible and in addition an outer covering should be applied which is proof against diffusion of air and water vapor. If the material permits the diffusion of air, the air will reach a point in the covering where the temperature is below the dew point. The condensed water will gradually accumulate until the covering becomes saturated, which will increase the conductivity and perhaps lower the mechanical strength of the covering until it becomes worthless.

2 ● Compute the total annual heat loss from 165 ft of 2-in. bare pipe in service 4000 hours per year. The pipe is carrying steam at 10 lb pressure and is exposed to an average air temperature of 70 F.

The pipe temperature is taken as the steam temperature, which is 239.4 F, obtained from Table 8, Chapter 1. The temperature difference between the pipe and air = 239.4 − 70 = 169.4 deg. By interpolation of Table 1 between temperature differences of 157.1 F and 227.7 F, the heat loss from a 2-in. pipe at a temperature difference of 169.4 deg is found to be 1.677 Btu per hour per linear foot per degree temperature difference. The total annual heat loss from the entire = 1.677 × 169.4 × 165 (linear feet) × 4000 (hours) = 188,000,000 Btu.

3 ● Coal costing $11.50 per ton and having a calorific value of 13,000 Btu per pound is being burned in the furnace supplying steam to the pipe line given in Question 2. If the system is operating at an over-all efficiency of 55 per cent determine the monetary value of the annual heat loss from the line.

The cost of heat per 1 million Btu supplied to the system = 1,000,000 × 11.5 (dollars) ÷ 13,000 (Btu) × 2000 (lb) × 0.55 (efficiency) = $0.804. The total cost of heat lost per year = 0.804 × 188 (million Btu) = $151.15.[4]

4 ● If the steam line given in Question 2 is covered with 1-in. thick 85 per cent magnesia, determine the resulting total annual heat loss through the insulation. Also compute the monetary value of the annual saving and the percentage of saving over the heat loss from the bare pipe.

By interpolation of Table 6 between temperature differences of 157.1 F and 227.7 F, the coefficient of transmission for 1-in. magnesia on a 2-in. pipe is found to be 0.525 Btu per hour per square foot of pipe surface per degree temperature difference at a temperature difference of 169.4 deg. The total hourly loss per square foot of insulated pipe will then be 0.525 × 169.4 = 89.04 Btu. From Table 3 the area per linear foot of 2-in. pipe is

[4]A closely approximate solution of this problem may be quickly made by use of the estimating chart given in Fig. 1.

found to be 0.622 sq ft The total annual loss through the insulation = 89 04 × 0 622 × 165 (linear feet) × 4000 (hours) = 36,550,000 Btu The annual bare pipe loss as determined in the solution of Question 2 was found to be 188 000.000 Btu The saving due to insulation is then 188,000,000 − 36,550,000 = 151,350,000 Btu per year.

From the solution of Question 3 it was found that the heat supplied to the system cost $0 804 per million Btu, therefore, the monetary value of the saving = 0 804 (dollars) × 151.35 (million Btu) = $121.69, or 81 2 per cent of the cost when using uninsulated pipe.

5 ● The manufacturer's list price for 85 per cent magnesia insulation is $0.36 per linear foot for 1-in. (standard thick) material to cover a 2-in. pipe. Determine the period of time required for the saving found in Question 4 to pay for the cost of the insulation if it can be purchased and applied at 80 per cent of list price (20 per cent discount).

The applied cost of insulation = 165 (linear feet) × 0 36 (dollars) × 0 80 (net) = 47 52. Since the annual saving as found in Question 4 amounts to $121.69, the insulation will pay for its cost in 47.52 ÷ 121.69 = 0 3905 years, in other words, the cost will be repaid 2 56 times by the saving obtained in one heating season.

6 ● The conductivity of magnesia insulation is 0.455 at the mean temperature which will result under the conditions of Question 4. Estimate the most economical thickness of magnesia for application on the pipe when operating under the conditions which are given in the foregoing problems and when a 20 per cent return is required on the investment for insulation.

Use chart given in Fig 3. Begin at the left bottom margin and proceed successively as shown by the dotted line example to the following essential data which are collected from the problems previously given:

4000 hours operation per year.
$0.804 value of heat, dollars per million Btu.
169.4 deg temperature difference.
0 455 conductivity of insulation.
20 per cent discount from list, cost of insulation
20 per cent fixed charges, return on investment
2-in. pipe size.

Solution of the problem by use of Fig 3 results in a required thickness of approximately 1.05 in. The nearest commercial thickness procurable is standard thick (1 1/32 in) magnesia.

(It is of interest to note that the use of Fig. 3 will generally result in solutions which, for all practical purposes, agree closely with the specifications for thicknesses given in Table 13)

7 ● Determine the minimum thickness of wool felt insulation having a conductivity of 0.30 necessary to prevent condensation of moisture on a 4-in. pipe carrying cold water at a temperature of 40 F when the surrounding air reaches maximum conditions of 90 F with a relative humidity of 90 per cent.

The difference between the temperature of the pipe and the surrounding air is 90 − 40 = 50 deg. For quick estimating purposes use the chart given in Fig. 2 Enter this chart at the lower left margin on the 90 per cent relative humidity line and proceed horizontally to the right to intersect the 90 deg air temperature line Project a line up to the 50 deg temperature difference line, and then horizontally to the right to the intersection with the 4-in. pipe size line. From this point proceed down to intersect the 0.30 line which denotes the conductivity of the insulation. Directly opposite this point of intersection the correct thickness of insulation is read from the scale on the lower right margin This chart solution denotes that wool felt 2 4-in. thick is sufficient to prevent condensation. The nearest commercial thickness procurable is 2½ in.

For prevention of condensation as well as for protection against freezing, if the thickness determined theoretically cannot be had, it is better to apply the next greater thickness procurable rather than to use any lesser thickness because an additional factor of safety is thus obtained.

Chapter 37

DISTRICT HEATING

Piping Distribution, Selection of Pipe Sizes, Provision for Expansion, Capacity of Returns with Various Grades, Conduits for Piping, Pipe Tunnels, Building Service Connections, Steam Consumption, Fluid Meters and Metering, Rates

THOSE phases of district heating which frequently fall within the province of the heating engineer are outlined here with data and information for solving incidental problems in connection with institutions and factories and for the design of heating systems for buildings which are to be supplied with purchased steam. A complete district heating installation should not be attempted without a thorough study of the entire problem by men competent and experienced in that industry.

PIPING DISTRIBUTION

The methods used in district heating work for the distribution of steam are applicable to any problem involving the supply of steam to a group of buildings. The first step is to establish the route of the pipes, and in this matter the local conditions so fully control the layout that little can be said regarding it.

Having established the route of the pipes, the next step is to calculate the pipe sizes. In district heating work it is common practice to design the piping system on the basis of pressure drop. The initial pressure and the minimum permissible terminal pressure are specified and the pipe sizes are so chosen that the required amount of steam, with suitable allowances for future increases, will be transmitted without exceeding this pressure drop. The steam velocity is therefore almost disregarded and may reach a very high figure. Velocities of 35,000 fpm are not considered high. By the use of this method the pipe sizes are kept to a minimum with consequent savings in investment.

The steam flowing through any section of the piping can be computed from a study of the requirements of the several buildings served. In general a condensation rate of 0.25 lb per hour per square foot of equivalent heating surface is a safe figure. This allows for line condensation which, however, is a small part of the total at times of maximum load. Any unusual requirements such as those for process steam should be individually calculated.

The steam requirements for water heating should be taken into account, but in most types of buildings this load will be relatively small compared with the heating load and will seldom occur at the time of the heating

peak. Unusual features such as large heaters for swimming pools should not be overlooked.

The pressure at which the steam is to be distributed will depend, in part, upon whether or not it has been passed through electrical generating units. If it has, the pressure will be considerably lower than if live steam, direct from the boilers, is used. The advantages of low pressure distribution (2 to 30 lb per square inch) are (1) smaller heat loss from the pipes, (2) less trouble with traps and valves, and (3) simpler problems in pressure reduction at the buildings. With distribution pressures not exceeding 40 lb per square inch there is little danger even if the full distribution pressure should build up in the radiators through the faulty operation of a reducing valve; but with pressures higher than this a second reducing valve or some form of emergency relief is usually desirable to prevent excessive pressures in the radiators. The advantages of high pressure distribution are (1) smaller pipe sizes and (2) greater adaptability of the steam to various operations other than building heating.

The different kinds of apparatus which frequently must be served require various minimum pressures. Kitchen equipment requires from 5 to 15 lb per square inch, the higher pressures being necessary for apparatus in which water is boiled, such as stock kettles and coffee urns. An increased amount of heating surface, which is easily obtained in some kinds of apparatus, results in quicker and more satisfactory operation at low pressures. For laundry equipment, particularly the mangle, a pressure of 75 lb per square inch is usually demanded although 30 lb per square inch is sufficient if the mangle is equipped with a large number of rolls and if a slow rate of operation is permissible. Pressing machines and hospital sterilizers require about 50 lb per square inch.

PIPE SIZES

The lengths of pipe, steam quantities, and initial and terminal pressures having been chosen, the pipe sizes can readily be calculated by means of the Unwin pressure drop formula. This formula, which gives pressure drops slightly larger than actual test results, is as follows:

$$P = \frac{0.0001306\ W^2L\left(1 + \frac{3.6}{D}\right)}{dD^5} \qquad (1)$$

where

P = pressure drop, pounds per square inch.
W = weight of steam flowing, pounds per minute.
L = length of pipe, feet.
D = inside diameter of pipe, inches
d = average density of steam, pounds per cubic foot.

This formula is similar to the Babcock formula given in Chapter 32.

Information on provision for expansion will be found in Chapter 34.

In general, return lines when installed follow the contour of the land, and Table 1 gives sizes of return pipes for various grades. It is evident that at points where the grade is great, smaller pipes can be installed.

CONDUITS FOR PIPING

Conduits for steam pipes buried underground should be reasonably waterproof, able to withstand earth loads and to take care of the expansion and contraction of the piping without strain or stress on the couplings, or without affecting the insulation or conduit. Expansion of the piping must be carefully controlled by means of anchors and expansion joints or bends so that the pipes can never come in contact with the conduit. Anchors can be anchor fittings or U-shaped steel straps which partially encircle the pipes and are firmly bolted to a short length of structural steel set in concrete.

TABLE 1. CAPACITY OF RETURNS FOR UNDERGROUND DISTRIBUTION SYSTEMS IN POUNDS OF CONDENSATE PER HOUR

Size[a] of Pipe In.	Pitch of Pipe per 100 Ft						
	6"	1"	2"	3"	5"	10"	20"
1	448	998	1890	2240	3490	5490	7490
1¼	1740	2490	3990	4880	6480	9480	13500
1½	2700	4190	5740	7480	9480	14500	20900
2	4980	7380	10700	13900	16900	24900	36900
3	13900	22500	30900	37400	50400	74800	105000
4	30900	44800	64800	79700	105000	154000	229000
5	54800	79800	120000	144800	195000	294000	418000
6	90000	138000	187000	237000	312000	449000	....
8	190000	277000	404000	508000	660000	938000	....
10	344000	498000	724000	900000	1190000	....	....
12	555000	798000	1148000	1499000	1990000	....	....

[a]Size of pipe should be increased if it carries any steam

In laying out conduits of this type the following points should be borne in mind:

1 An expansion joint, offset, or bend should be placed between each two anchors.

2. If the distance between buildings is 150 ft or less and the steam line contains high-pressure steam, the line may be anchored in the basement of one building and allowed to expand into the basement of the second building. If the steam line contains low-pressure steam (up to 4-lb pressure), this method may be used if buildings are 250 ft or less apart

3. If the distance between buildings is between 150 ft and 300 ft and the steam line contains high-pressure steam, the lines should be anchored midway between the buildings and allowed to expand into the basements of both buildings. If the steam line contains low-pressure steam this method may be used if buildings are between 250 ft and 600 ft apart. No manhole is required at the anchor, and a blind pit is all that is necessary.

4. For longer lines, manholes must be located according to judgment and depending upon the expansion value of the type of expansion joint or bend that is used. The minimum number of manholes will be required when an expansion bend or an anchor with double expansion joint is placed in each manhole and the pipes are anchored midway between manholes

5. A proper hydrostatic test should be made on the assembled line before the insulation and the top of the conduit are applied. The hydrostatic pressure should be one-and-one-half times the maximum allowable pressure and it should be held for a period of at least two hours without evidence of leakage. In any case the pressure should be no less than 100 lb per square inch.

The styles and construction of conduits commonly used may be classified as follows. Some of the more common forms are illustrated in Fig. 1.

Wood Casing: The pipe is enclosed in a cylindrical casing usually having a wall 4 in. thick and built of segments which are bound together by a wire wrapped spirally around

the casing. The casing is lined with bright tin and coated with asphaltum. The pipe is supported on rollers carried in a bracket which fits into the casing The lengths of casing are tightly fitted together with a male and female joint This form of conduit is illustrated in Fig. 1 at *A* The casing rests on a bed of crushed stone with tile drains laid below. The tile drains are of 4-in field tile or vitrified sewer tile, laid with open joints.

Filler Type: The pipes are supported on expansion rollers properly supported from the conduit or independent masonry base. The pipes are protected by a split-tile conduit, and the entire space between the pipes and the tile is filled with an insulating filler. Thus the pipes are nested and the insulation between them and the tile effectively prevents circulation of air. The conduit is placed on a bed of gravel or crushed rock from 4 to 6 in thick, which is extended upward so as to come about 2 in. above the parting lines of the tile. A tile underdrain is placed beneath the conduit throughout the entire length and is connected to sewers or to some other point of free discharge At *B* and *D* in Fig 1 are shown two forms of tile conduit of the filler type.

Circular Tile or Cast-Iron Conduit: The pipes are carried on expansion rollers supported on a frame which rests entirely on the side shoulders of the base drain foundation.

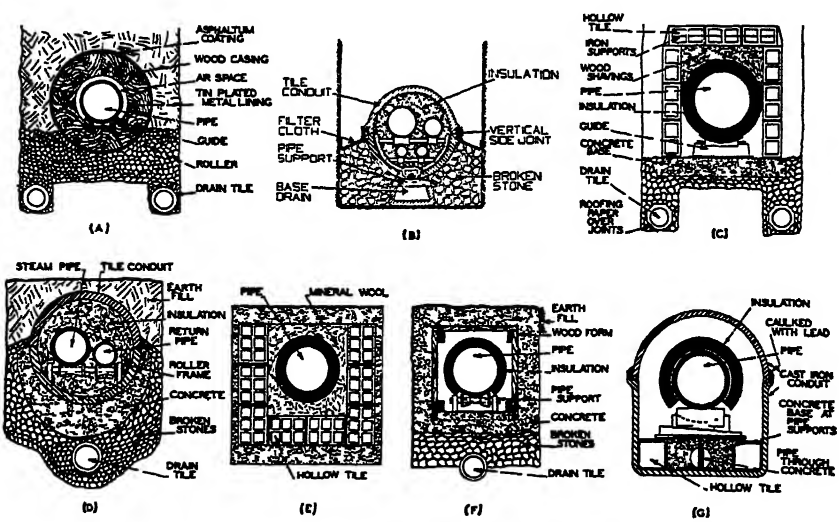


Fig. 1. Construction Details of Conduits Commonly Used

The pipes are protected by a sectional tile conduit, scored for splitting, or a cast-iron conduit, both being of the bell and spigot type. The conduit has a longitudinal side joint for cementing, after the upper half of conduit is in place, so shaped that the cement is keyed in place while locking the top and bottom half of the conduit together with a water-tight vertical side joint. The cast-iron conduit has special side locking clamps in addition to the vertical side joint. The entire space between the conduit and the pipes is filled with a water-proofed asbestos insulation. The conduit is supported on the base drain foundation, each section resting on two sections of the base drain, thus interlocking. The base drain is so shaped that it provides a cradle for the conduit, resting solidly on the trench bottom and providing adequate drainage area immediately under the conduit. The underdrain is connected to sewers or some other point of free discharge. For tile conduit the base drain is vitrified salt glazed tile and for cast-iron conduit it is either extra heavy tile or cast-iron. A free internal drainage area is also provided to carry away any water that may collect on the inside of the conduit from a leaky pipe or joint in the conduit. Broken stone is filled in around the base drain and up to the vertical side joint. The broken stone is covered with an asphalted filter cloth to prevent sand from sifting through the broken stone and clogging the drainage area of the base drain. The tile conduit is made in 2-ft lengths and the cast-iron conduit in 4-ft lengths, cast in

separate top and bottom halves. Special reinforcing ribs give the cast-iron conduit ample strength with minimum weight

Insulated Tile Type: The insulating material, diatomaceous earth, is molded to the inside of the sectional tile conduit. The space between the pipes and the insulating conduit lining may also be filled with insulation. The pipes are carried on expansion rollers supported on a frame which rests on the side shoulders of the base drain foundation. This type of conduit has the same mechanical features as those described under the heading Circular Tile or Cast-Iron Conduit.

Sectional Insulation Type (Tile or Cast-Iron). Each pipe is insulated in the usual way with any desired type of sectional pipe insulation over which is placed a standard waterproof jacket with cemented joints. The pipes are enclosed in a sectional tile or cast-iron conduit as described under the heading Circular Tile or Cast-Iron Conduits.

Sectional Insulation Type (Tile or Concrete Trench): A type of construction frequently used in city streets, where service connections are required at frequent intervals, the pipes are insulated as described in the preceding paragraph, and are enclosed in a box or trench made either entirely of concrete or with concrete bottom and specially constructed tile sides and tops. The pipes are supported on roller frames secured in the concrete. At *C* and *E*, Fig. 1, are shown two tile conduits using sectional insulation. In these particular designs the space surrounding the pipe is filled partially or wholly with a loose insulating material. The use of loose material in addition to the sectional insulation is, of course, optional and is only justifiable where high pressure steam is used. The conduit shown at *F* is of a similar type and has the advantage of being made entirely of concrete and other common materials.

Sectional Insulation Type (Bituminized Fibre Conduit): Each pipe is individually insulated and encased in a bituminized fibre conduit. The insulating material is 85 per cent carbonate of magnesia sectional pipe covering, applied in the usual manner as on overhead pipes, except that bands are omitted. After every fifth section of magnesia covering there is applied a short, hollow section of very hard asbestos material in the bottom portion of which rests a grooved-iron plate carrying ball-bearings upon which the pipe rides when expanding or contracting. This short expansion section is of the same outside diameter as the adjacent 85 per cent magnesia covering. Over the pipe covering and expansion device there are placed two layers of bituminized fibre conduit with all joints staggered, and the surface of each conduit is finished with liquid cement. Conduits are placed on a bed of crushed rock or gravel, approximately 6 in. deep, and this is extended upward to about the center line of the conduit when trench is backfilled. Underdrains leading to points of free discharge are placed in the gravel or crushed rock beds.

Special Water-Tight Designs: It is occasionally necessary to install pipes in a very wet ground, which calls for special construction. The ordinary tile or concrete conduit is not absolutely water tight even when laid with the utmost care. The conduit shown at *G*, Fig. 1, is of cast-iron with lead-calked joints and is water tight if properly laid. It is obviously expensive and is justified only in exceptional cases. A reasonably satisfactory construction in wet ground is the concrete or tile conduit with a waterproof jacket enclosing the pipe and its insulation, and with the interior of the conduit carefully drained to a manhole or sump having an automatic pump. It is useless to install external drain tile when the conduit is actually submerged.

PIPE TUNNELS

Where steam heating lines are installed in tunnels large enough to provide walking space, the pipes are supported by means of hangers or roller frames on brackets or frame racks at the side or sides of the tunnel. The pipes are insulated with sectional pipe insulation over which is placed a sewed-on, painted canvas jacket or a jacket of asphalt-saturated asbestos water-proofing felt. The tunnel itself is usually built of concrete or brick and water-proofed on the outside with membrane water-proofing.

On account of their relatively high first cost as compared with smaller conduits, walking tunnels are sometimes not installed where provision for the heating lines is the only consideration, but only where they are required

to accommodate miscellaneous other services or provide underground passage between buildings.

BUILDING SERVICE CONNECTIONS

Most district heating companies enforce certain regulations regarding the consumer's installation, partly to safeguard their own interests but principally to insure satisfactory and economical service to the consumer.

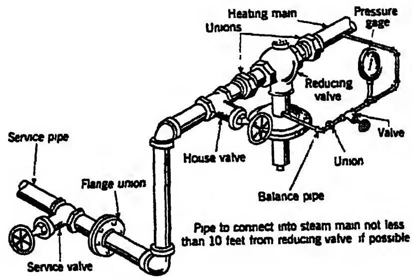


Fig. 2. Connections for Reducing Valves of Size Less than 4 Inches

There are certain fundamental principles that should be followed in the design of a building heating system which is to be supplied from street mains. Although some of these apply to any building, they have been demonstrated to be especially important when steam is purchased.

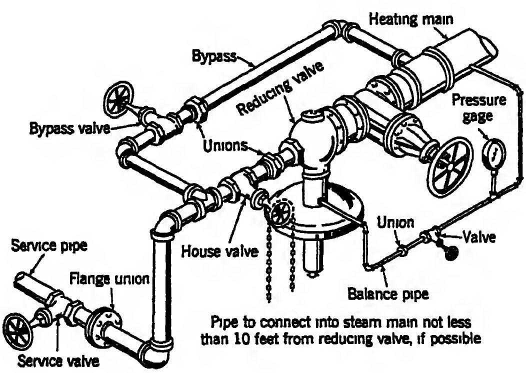


Fig. 3. Connections for Reducing Valves of Size 4 Inches and Larger, and for Expanded Valves

Figs. 2 and 3 show typical service connections used for low pressure steam service. As shown in Fig. 2, no by-pass is used around the reducing valve on sizes less than 4 in. Fig. 3 illustrates the use of a by-pass around reducing valves 4 in. and larger. This latter construction permits the

operation of the line in case of failure in the reducing valve. In the smaller sizes, the reducing valve can be removed, a filler installed, and the house valve used to throttle the flow of steam.

Fig. 4 shows a typical installation used for high pressure steam service. The first reducing valve, usually furnished by the utility company,

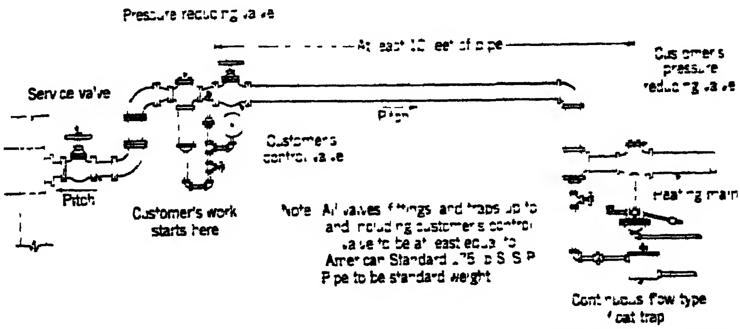


FIG 4. STEAM SUPPLY CONNECTION WHEN USING CONDENSATION METER

effects the initial pressure reduction. The second reducing valve, usually furnished by the customer, reduces the steam pressure to that required.

1. Provision should be made for conveniently shutting off the steam supply at night and at other times when heat is not needed.

It has been thoroughly demonstrated that a considerable amount of heat can be saved by shutting off steam at night. Although there is, in

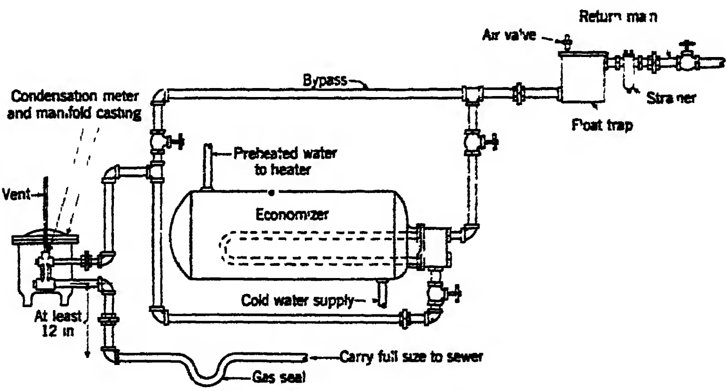


FIG 5. RETURN PIPING FOR CONDENSATION METER

some cases, an increased consumption of heat when steam is again turned on in the morning, there is a large net saving which may be explained by the fact that the lower inside temperature maintained during the night obviously results in lower heat loss from the building, and less heat need therefore be supplied.

Steam can be entirely shut off at night in most buildings even in very

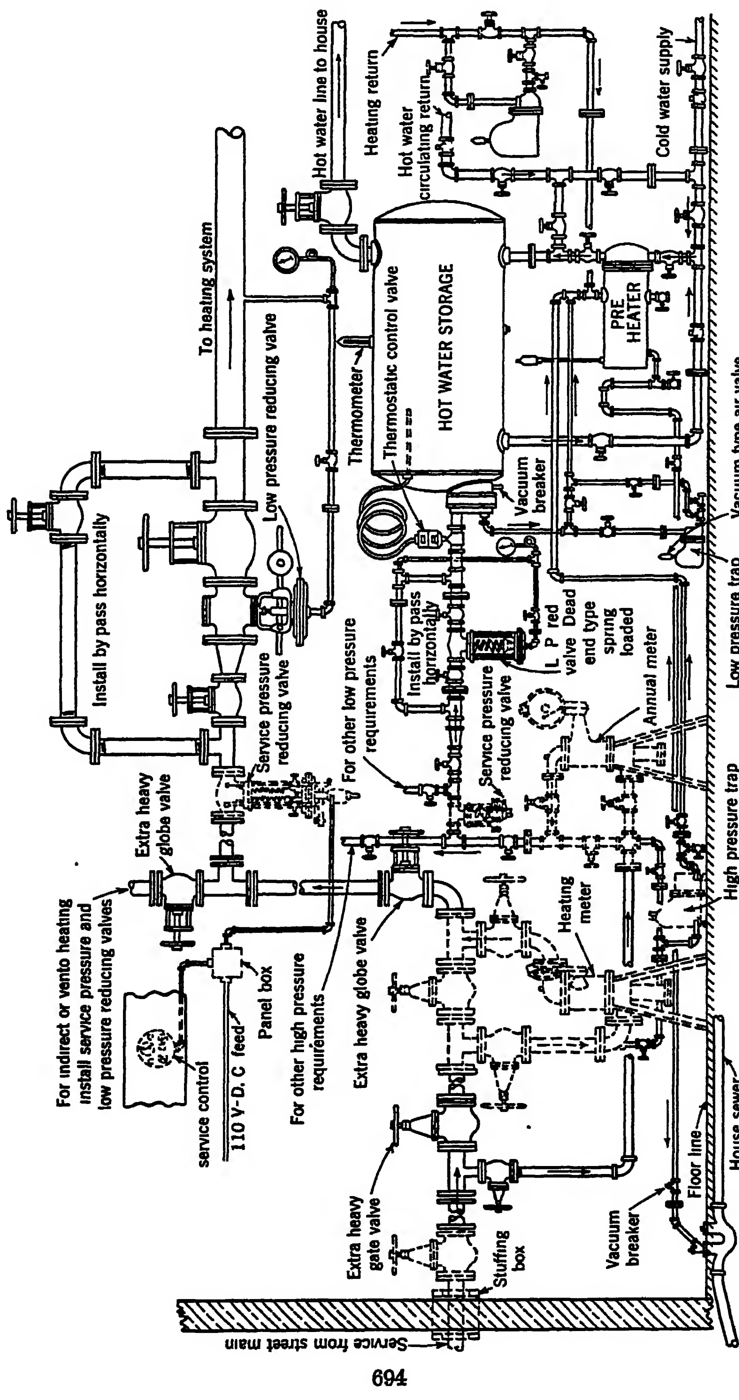


Fig. 6. Typical Service Installation

cold weather without endangering plumbing. It is necessary, however, to have an ample amount of heating surface so that the building can be quickly warmed in the morning. Where the hours of occupancy differ in various parts of the building, it is good practice to install separate supply pipes to the different parts. For example, in an office building with stores or restaurants on the first floor which are open in the evening, a separate main supplying the first floor will permit the steam to be shut off from the remainder of the building in the late afternoon. The division of the building into zones each with a separately controlled heat supply is sometimes desirable, as it permits the heat to be adjusted according to variations in sunshine and wind.

2. Residual heat in the condensate should be salvaged.

This heat may be salvaged by means of a cooling radiator, or as is more frequently done, by a water heating economizer (see Fig. 5) which preheats the hot water supply to the building. Fig. 6 shows a typical steam service installation for high pressure steam, complete for steam flow metering, water heating, preheating, automatic heating control, and for using steam for other purposes.

The condensation from the heating system, after leaving the trap, passes through the preheater on its way to the meter. The supply to the hot water heater passes through the preheater, absorbing heat from the condensation. If the hot water system in the building is of the recirculating type, the recirculating connection should be tied in *between* the preheater and the water heater proper, not at the preheater inlet, because the recirculated hot water is itself at a high temperature. The number of square feet of heating surface in the preheater should be approximately equal to one per cent of the equivalent square feet of heating surface in the building.

Because of the lack of coincidence between the heating system load and the hot water demand, a greater amount of heat can be extracted from the condensation if storage capacity is provided for the preheated water. Frequently a type of preheater is used in which the coils are submerged in a storage tank.

3. Heat supply should be graduated according to variations in the outside temperature.

This may be done in several ways, as by the use of thermostats of various types or by orifice systems. Another method which is very simple is the use of an ordinary vacuum return line system in which the pressure in the radiators is varied between a high vacuum and a few pounds pressure, thus producing some control over the heat output. One form of control which appears to be well suited for controlling district steam service to a building is the weather compensating thermostat. It regulates the steam supply automatically according to the outdoor temperature, and gives frequent short intervals of intermittent steam supply, and at the same time insures delivery of steam to all the radiators.

Another form of regulation, known as the time-limit control, is sometimes employed for regulating the steam supply from the central station main to the building. Such a control provides an intermittent supply of steam to the radiators either throughout the 24 hours of the day or during the day-

time hours only. The setting of a switch may provide no service, continuous service, or periodic service. For the latter, by means of several intermittent settings, steam will be supplied during each period in increments of a certain number of minutes for each successive setting of the switch, steam being shut off during the balance of the period. These settings afford from 15 to 80 per cent of the maximum heating effect required on days of zero temperature. A night switch with a variety of settings may be adjusted so as to maintain throughout the night the intermittent supply called for by the day switch setting, or may be set to interrupt the operation of the day switch and entirely cut off the supply of steam to the radiation at night during certain hours which are selected by the operating engineer.

FLUID METERS

No one thing has contributed more to the advancement of district heating than the perfection of fluid meters, which may be classified as follows:

1. *Positive Meters:* The fluid passes in successive isolated quantities—either weights or volumes. These quantities are separated from the stream and isolated by alternately filling and emptying containers of known capacity.

2. *Differential Meters:* The fluid does not pass in isolated separately-counted quantities but in a continuous stream which may flow through the line without actuating the primary device of the meter. In the differential meter, the quantity of flow is not determined by simple counting, as with the positive meter, but is determined from the action of the steam on the primary element.

Additional subdivisions of these two general classifications can be made as follows:

- Fluid Meters
  - Positive - quantity
    - Weighing
      - Weighers
      - Tilting trap
    - Volumetric
      - Rotary
      - Bellows
  - Differential
    - Quantity - Current - Turbine
    - Rate of flow
      - Head (Kinetic)
        - Venturi
        - Flow nozzle
        - Orifice
        - Pitot tube
      - Area (Geometric)
        - Orifice and plug
        - Cylinder and piston
      - Head area (Weir)
        - V-notch
        - Special notch

In selecting a meter for a particular installation, the number of different makes and types of meters suitable for the job is usually limited by one or more of the following considerations:

1. Its use in a new or an old installation.
2. Method to be used in charging for the service.
3. Location of the meter.
4. Large or small quantity to be measured.
5. Temporary or permanent installation.

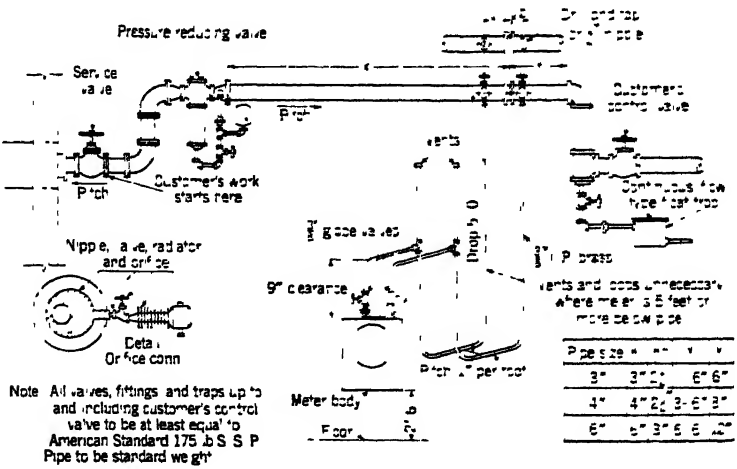


FIG 7 ORIFICE METER STEAM SUPPLY CONNECTION

6. Cleanliness of the fluid to be measured
7. Temperature of the fluid to be measured.
8. Accuracy expected.
9. Nature of flow turbulent, pulsating, or steady.
10. Cost.
    (*a*) Purchase price.
    (*b*) Installation cost.
    (*c*) Calibration cost.
    (*d*) Maintenance cost.
11. Servicing facilities of the manufacturer
12. Pressure at which fluid is to be metered.
13. Type of record desired as to indicating, recording or totalizing
14. Stocking of repair parts.
15. Use of open jets where steam is to be metered.
16. Metering to be done by one meter or by a combination of meters.
17. Use as a check meter.
18. Its facilities for determining or recording information other than flow.

Condensation Meters

The majority of the meters used by district heating companies in the sale of steam to their customers are of the condensation or flow types.

The condensation meter is a popular type for use on small and medium sized installations, where all of the condensate can be brought to a common point for metering purposes. Its simplicity of design, ease in testing, accuracy at all loads, low cost, and adaptability to low pressure distribution has made it standard equipment with many heating companies.

Two types of condensation meters are in general use: the *tilting bucket* meter and the *revolving drum* or *rotor* meter of which there are several makes on the market. Condensation meters should not be operated under

pressure; they are made for either gravity or vacuum installation. Continuous flow traps are necessary ahead of the meter if a vented receiver is not used. Where bucket traps are used, a vented receiver before the meter is essential. If desirable a receiver may be used with a continuous flow trap, but this is not necessary.

Steam flow meters are available in many types and combinations, as indicated in the subdivision covering fluid meters on page 680.

The *orifice and plug* meter is one in which the steam flow varies directly as the area of the orifice. The vertical lift of the plug, which is proportional to the flow, is transmitted by means of a lever to an indicator and to a

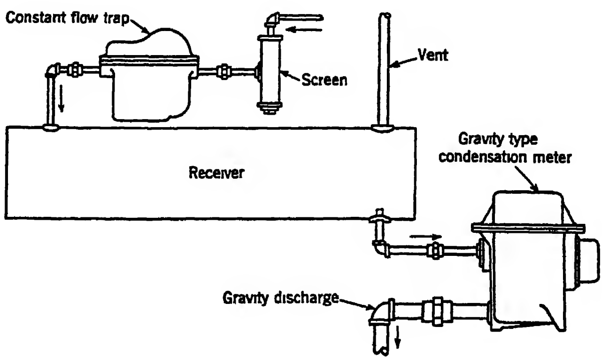


FIG. 8. GRAVITY INSTALLATION FOR CONDENSATION METER USING VENTED RECEIVERS

pencil arm which records the flow on a strip chart. The total flow over a given period is obtained by measuring the area by using a planimeter on the chart and applying the meter constant.

Fig. 7 shows a typical orifice type meter connection and indicates typical requirements in the installation of this type of meter. Fig. 8 illustrates a gravity installation using a vented receiver ahead of the meter, while Fig. 9 shows a vacuum installation without a master trap.

Flow meters using an orifice, Venturi tube, flow nozzle, or Pitot tube as the primary device are made by a number of manufacturers and can be obtained in either the mechanically or electrically operated type. The electric flow meter makes it possible to locate the instruments at some distance from the primary element.

Flow meters employing the orifice, Venturi tube, flow nozzle or Pitot tube should be so selected as to keep the lower operating range of the load above 20 per cent of the capacity of the meter. This is desirable for accuracy as the differential pressure at light loads is too small to properly actuate the meter. A few general points to be considered in installing a meter of this type are:

1. It is desirable to place the differential medium in a horizontal pipe in preference to a vertical one, where either location is available.

2. Reservoirs should always be on the same level and installed in accordance with the instructions of the meter company.

3. The meter body should be placed at a lower level than that of the pressure differential medium. Special instructions are furnished where the meter body is above.
4. Meter piping should be kept free from leaks.
5. Sludge should not be permitted to collect in the meter body.
6. The meter body and meter piping should be kept above freezing temperatures
7. It is best not to connect a meter body to more than one service.
8. Special instructions are furnished for metering a turbulent or pulsating flow.

STEAM CONSUMPTION

The following factors are used in New York City for the different classes of buildings listed. The factors are based on maintaining an inside tem-

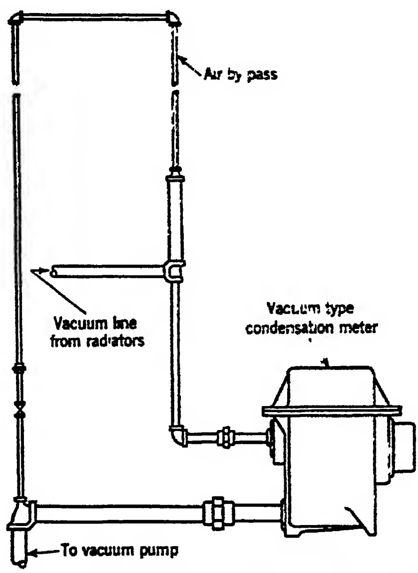


FIG. 9. VACUUM CONDENSATION METER INSTALLATION WITHOUT MASTER TRAP

perature of 70 F for certain hours, with a minimum outside temperature of 0 F and an average of 43 F for the heating season of eight months (October 1 to June 1). In this group are six types of buildings.

Manufacturing or commercial loft type where steam is used to heat the premises during the day hours to maintain 65 to 68 F from 9 a.m. to 5 p m. No Sunday or holiday use and no night use. *Factor:* 325 lb per square foot of heating surface per season.

Office buildings using steam during daylight hours to maintain 70 F from 9 a.m. to 6 p.m. for approximately 240 days (heating season). No night use. *Factor:* 400 lb per square foot of heating surface per season.

Office buildings using steam during day hours and at night when required to 7, 8 and 9 p m. (customary where there are stock brokers or banking offices), 240 days. *Factor:* 500 lb per square foot of heating surface per season.

Residences of the block type (not detached) where high-class heating service is required; somewhat similar to apartment buildings. *Factor:* 550 lb per square foot of heating surface per season.

Apartment houses where high-class heating service is required (Steam off at midnight.) *Factor:* 650 lb per square foot of heating surface per season.

Hotels (commercial type) where very high-class service is required for 24 hours *Factor:* 800 lb per square foot of heating surface per season.

By assuming one square foot of equivalent heating surface for each 100 cu ft of space heated, which seems a fair ratio in New York City, it is possible roughly to estimate the steam required per cubic foot of space, information which is often more easily obtained than the square feet of heating surface. Additional data on the heating requirements of various types of buildings in a number of cities may be found in the Handbook of the *National District Heating Association.*

RATES

Fundamentally, district heating rates are based upon the same principles as those recognized in the electric light and power industry, the main object being a reasonable return on the investment. However, there are other requirements to be met; the rate for each class of service should be based upon the cost to the utility company of the service supplied and upon the value of the service to the consumer, and it must be between these two limits. The profit need not be divided proportionately among the rate groups, but should be established from a competitive standpoint. District heating rates should be designed to produce a sufficient return on the investment regardless of weather conditions, although existing rate schedules do not conform with this principle. Lastly, the rate schedule must be reasonably easy for the intelligent layman to comprehend.

Depreciation should be based on a careful estimate of the life of various elements of the property. Appropriations to reserves should be made, with generosity in good years and with discretion in less favorable years.

Glossary of Terms

Load Factor. The ratio, in per cent, of the average load to the maximum load. This is usually based on a one year period but may be applied to any specified period.

Demand Factor. The relation between the connected radiator surface or required radiator surface and the demand of the particular installation. It varies from 0.25 to 0.3 lb per hour per square foot of surface.

Diversity Factor. The ratio of the sum of the individual demands of a number of buildings to the actual composite demand of the group.

Types of Rates

A. Flat Rates.
- 1 Radiator surface charge. *Obsolescent*

B Meter Rates.
- 1. Straight-line.
- 2 Step. *Obsolescent*
- 3. Block.
  - (*a*) Class rates

C. Demand Rates
- 1. Flat demand.
- 2. Wright.
- 3. Hopkinson.
- 4. Doherty (or Three charge)

Straight-Line Meter Rate The price charged per unit is constant and the consumer pays in direct proportion to his consumption without regard to the difference in costs of supplying the individual customers

Block Meter Rate The pounds of steam consumed by a customer are divided into blocks of M lb each, and lower rates are charged for each successive block consumed. This type of charge predominates in steam heating rate schedules for it has the advantage of proportioning the bill according to the consumption and the cost of service. It has the disadvantage of not discriminating between customers having a high load factor (relatively low demand) and those having a low load factor (relatively high demand). The utility company must maintain sufficient capacity to serve the high demand customers and the cost of the increased plant investment is divided equally among the users, so the high demand customers are benefited at the expense of the others

Demand Rates. These refer to any method of charge based on a measured maximum load during a specified period of time

The *flat demand rate* is usually expressed in dollars per M lb of demand per month or per annum. It is based on the size of a customer's installation and is seldom used except where a flow meter is not practicable.

The *Wright demand rate* is similar in calculation to the block rate except that it is expressed in terms of hours' use of the maximum demand It is seldom used but forms the basis for other forms of rates.

The *Hopkinson demand rate* is divided into two elements·

(*a*) A charge based upon the demand, either estimated or measured,
(*b*) A charge based upon the amount of steam consumed.

This rate may be modified by dividing the quantities of steam demanded and consumed into blocks charged for at different rates.

Demand rates are comparatively new and are not yet widely used, though they are equitable and competitive they are difficult for the average layman to understand They are of benefit to utility companies and to consumers because the investment and operating costs can be divided to suit the particular circumstances into demand, customer and consumption groups through the use of some modification of the Hopkinson rate.

Fuel Price Surcharge It is usually desirable to establish a rate upon a specified basic cost of fuel to the utility company. Where there are wide variations in the price of fuel, it is also desirable to add a definite charge per M lb of steam sold for each increment of increase in the price of fuel. This surcharge automatically compensates for the variations without necessitating frequent changing of the whole rate structure

REFERENCES

Pipe Line Design for Central Station Heating, by B. T Gifford (A.S H.V.E TRANSACTIONS, Vol. 17, 1911).

Engineering and Cost Data Relative to the Installation of Steam Distributing Systems in a Large City, by F. H. Valentine (A S H.V.E. TRANSACTIONS, Vol. 22, 1916).

Transmission of Steam in a Central Heating System, by J. H. Walker (A.S.H V E. TRANSACTIONS, Vol. 23, 1917).

Efficiency of Underground Conduit, by G B. Nichols (A S.H.V.E. TRANSACTIONS, Vol. 23, 1917).

Economical Utilization of Heat from Central Plants, by N. W. Calvert and J. E. Seiter (A.S.H.V.E. TRANSACTIONS, Vol 30, 1924).

Standard Connections for Condensation Meters, (*N.D.H.A.* Proceedings, Vol XII, pp. 63-76).

Installation and Maintenance of Steam Meters, (*N.D.H.A.* Proceedings, Vol. XIII, pp. 177-183).

Inaccuracy in Flow Meter Calculations, (*N.D.H.A.* Proceedings, Vol. XIII, pp 183-193).

Testing of Steam Meters, (*N.D.H.A.* Proceedings, Vol. XIV pp. 272-276).

Meter Accuracy Guarantees, (*N.D.H.A.* Proceedings, Vol. XIV, pp. 276-277).

Effect of Pulsations on the Flow of Gases, (*N.D.H.A.* Proceedings, Vol. XIV, pp. 277-281).

Meter Connections, (*N.D.H.A.* Proceedings, Vol. XX, pp. 126-143).

Layout for Testing Meters, (*N.D.H.A.* Proceedings, Vol. XX, pp. 391-392).

Characteristic Meter Calibration Curves, (*N.D.H.A.* Proceedings, Vol. XX, pp. 444-453).

Rates, (*N.D.H.A.* Handbook, 1932, Chapter 10).

PROBLEMS IN PRACTICE

1 ● What is the common method of determining the size of mains in a distribution system?

On the basis of pressure drop: The initial pressure and the minimum permissible terminal pressure are specified, and the pipe sizes are so chosen that the maximum estimated amount of steam may be transmitted without exceeding this pressure difference. The steam's velocity is disregarded and it may reach a magnitude in excess of 35,000 fpm which is not considered high.

2 ● a. What are the advantages and disadvantages of a low pressure distribution system?

b. High pressure?

a. The advantages of a low pressure distribution system include:

1. Smaller heat loss from the pipes
2. Less trouble with traps and valves.
3. Simpler problems with pressure reducing equipment at the buildings.
4. No danger to building heating equipment from high pressure through failure of the reducing valves.

The disadvantages of a low pressure system are:

1. Larger pipe sizes.
2. Decreased field of usefulness owing to small pressure range.

b. The advantages of a high pressure system are:

1. Smaller pipe sizes.
2. Greater adaptability of the steam to various uses other than building heating.

The disadvantages of a high pressure system are:

1. Large heat loss from the pipes.
2. The high pressure traps and valves required often give more trouble than low pressure traps and valves do.
3. Extra heavy fittings are required.
4. Usually two reducing valves or some form of emergency relief is necessary to protect the building piping system.

3 ● Determine the size of pipe from the following data using Unwin's formula:
Length of pipe, 600 ft.
Steam to be carried, 90,000 lb per hour, dry saturated.
Initial pressure, 100 lb per square inch, gage.
Final pressure, 40 lb per square inch, gage.

Using the formula:

$$P = \frac{0.0001306\ W^2L\left(1 + \frac{3.6}{D}\right)}{dD^5}$$

The pressure drop $P = 100 - 40 = 60$ lb per square inch.

The weight of steam per minute $W = \dfrac{90{,}000}{60} = 1500$

The length of pipe in feet $L = 600$

The average density of steam d in pounds per cubic foot, taken from Keenan's Table:

$$\begin{aligned} \text{At 100-lb gage, } d &= 0.2575 \\ \text{At 40-lb gage, } d &= 0.1289 \\ \text{Average } d &= 0.1932 \end{aligned}$$

The diameter of the pipe in inches $= D$.

Substituting the values in the formula:

$$60 = \frac{0.0001306 \times 1500^2 \times 600 \left(1 + \dfrac{3.6}{D}\right)}{0.1932 \times D^5}$$

$$D = 7.35 \text{ in.}$$

Therefore, an 8-in. pipe should be used.

4 ● What points should be borne in mind when laying out an underground steam conduit?

The conduit should be reasonably waterproof, able to withstand earth loads and to take care of the expansion and contraction of the piping without strain or stress on the couplings, or without affecting the insulation or the conduit. Expansion of the piping must be carefully controlled by means of anchors and expansion joints or bends so that the pipes can never come in contact with the conduit.

5 ● What is considered the proper pressure for a hydrostatic test before completing the conduit?

In the case of any underground piping which is to be buried or otherwise made inaccessible, the assembled lines shall first be tested hydrostatically at a pressure of one and one-half times the maximum allowable service pressure and held for a period of at least two hours without evidence of leakage. In any case the hydrostatic pressure should not be less than 100 lb per square inch

6 ● What factors should be considered before determining the route of a steam line?

1. The line should be so located that it will bring in the greatest revenue (or supply the most steam) with the least cost.
2. The ultimate length and size of services and branches necessary with each possible location should be estimated, for mains should be run near to the big loads.
3. The location of the boiler room or piping center of present and future buildings to be served should be considered.
4. Where possible, make the lines straight between manholes.
5. Avoid such obstructions as other lines, sewers, ducts, curb drains, manholes, valve boxes, catch basins, fire hydrants, and poles; especially avoid electric ducts and water lines.
6. Avoid locating lines near where pile driving and foundation construction for new buildings will take place.
7. Consider construction difficulties such as traffic, hard rock, and wet earth, which increase time and labor
8. Consider the economies of using available sidewalk vaults of buildings. Weigh the advantage of less excavation against the cost of obstruction removal
9. Consider all operating difficulties.
10. Consider the difficulties of negotiating agreements for lines on private property where public and private rights-of-way are available.
11. Consider the effect of proposed municipal and other improvements.
12. Consider municipal regulations.

7. ● State the advantages and disadvantages of tunnels over conduits.

The advantages of pipe tunnels over conduits are

1. Accommodation for miscellaneous services other than steam.
2. Provision of an underground passage between buildings
3. Easy installation of additional pipes and easy replacement of existing pipes with larger sizes.
4. Easy inspection and maintenance of pipes.

The disadvantages of pipe tunnels over conduits are:

1. Higher first cost
2. Higher maintenance cost in general

8 ● Is the steam consumption less in a building that shuts off its steam at night than in one that does not? Why?

It has been thoroughly demonstrated that the steam consumption is less in a building where the steam is shut off at night Although there is, in some cases, an increased consumption of heat when steam is again turned on in the morning, there is a large net saving which may be explained by the fact that the lower inside temperature maintained during the night obviously results in lower heat loss from the building, and less heat need therefore be supplied.

9 ● Is the condensate from a building supplied with purchased steam always discharged to the sewer?

No. In some cities where the customers are not spread over too wide a territory and where natural water conditions make the treatment of boiler feed water expensive, the steam company provides mains for the return of the condensate to the boilers

10 ● What are the common methods for salvaging heat in condensate?

The most common methods are·

1. The use of a water heating economizer for preheating the hot water supply to the building
2 The use of a cooling radiator.

11 ● What are the common means used to graduate the heat supply according to variations in outside temperature?

a. A weather compensating thermostat regulates the steam supply automatically according to the outdoor temperature, and gives frequent short intervals of intermittent steam supply, at the same time it insures delivery of steam to all the radiators

b. Another method which is very simple is the use of an ordinary vacuum return line system in which the pressure in the radiators is varied between a high vacuum and a few pounds to produce some control over the heat output

c. The use of an orifice system graduates heat supply.

d The time-limit control which may be set to provide no service, continuous service, or periodic service, is also used For periodic service, steam may be supplied during each period in increments of a certain number of minutes for each successive setting of the switch, steam being shut off during the balance of the period. This type of service is provided by several intermittent settings A night switch will maintain the intermittent day setting, or interrupt the day operation and cut off the supply of steam at night during any desired hours.

Chapter 38

RADIANT HEATING

Physical and Physiological Factors, British Equivalent Temperature, Control of Heat Losses, Application Methods, Calculation Principles, Mean Radiant Temperature, Measurement of Radiant Heating

HEATING for comfort is generally understood to mean that heat must be supplied to control the rate of heat loss from the human body so that the physiological reactions are conducive to a feeling of comfort in the individual. While in convection heating, as described in Chapter 30, heat is transferred from a heating unit to the air and thence to the occupant, the primary object of radiant heating is to warm the occupant directly without heating the air to any extent. Thus, the difference between convection heating and radiant heating is partly physical and partly physiological.

Comfort requires that heat be removed from the body at the same rate as it is generated by the oxidation of the foodstuffs in the body tissues. The normal rate of heat production in a sedentary individual is about 400 Btu per hour[1], or, since the entire surface area of an average adult is 19.5 sq ft, about 20.5 Btu per square foot per hour. Conditions should be such as to remove heat at this rate if the surface is to be maintained at the mean normal surface temperature of the human body.

Heat is transferred from any warm dry body to cooler surroundings principally by convection and by radiation, the approximate total rate of heat loss being the sum of the two. Where the body surface is moist there is additional loss of heat through evaporation from both the body surface and the respiratory tract.

The rate of heat loss by convection depends upon the difference between the temperature of the body and that of the surrounding air, and on the rate of air motion over the body. The loss by radiation depends entirely upon the difference between the temperature of the body and the mean surface temperature of the surrounding walls and objects. This latter temperature is called the *mean radiant temperature* (MRT). Because these two types of heat loss act in a supplementary manner toward each other, a required rate of heat loss can be secured by having a relatively low air temperature and a relatively high MRT, or *vice versa*. Thus, if the air is reduced from a given temperature to a lower temperature, the amount of heat lost from the body by convection is increased, and this

[1]Heat and Moisture Losses from the Human Body and Their Relation to Air Conditioning Problems, by F. C. Houghten, W. W. Teague, W. E. Miller, and W. P. Yant (A.S.H.V.E. TRANSACTIONS, Vol 35, 1929)

increase can be compensated for by raising the MRT. Similarly, with a higher air temperature the same total heat loss will be maintained by a correspondingly lower MRT.

The loss by evaporation depends on the air temperature, air movement, and humidity; it is increased if the humidity is reduced. For the usual conditions of heating by radiators or convectors, where the air temperature ranges from 70 F to 73 F, approximately 75 per cent of the total heat loss of 400 Btu per hour occurs by radiation and convection, and the balance, or 100 Btu per hour, occurs by evaporation. In the case of radiant heating, if the air temperature is reduced to 60 F, 84 per cent of the 400 Btu per hour, or 336 Btu per hour, is lost by radiation and convection, and 64 Btu per hour are lost by evaporation.

The mean normal surface temperature of the human body, taken over the whole area, including not only the exposed skin surface but also surfaces of the clothes and the hair, has been very extensively used as 75 F, particularly in British literature. However, results obtained by Aldrich[2] in rooms in which the air and wall surface temperatures were approximately 72 F gave mean values nearer to 83 F than to 75 F.

The mean body surface temperature which will maintain the optimum heat loss by radiation and convection in a uniform environment of 72 F may be calculated from fundamental equations for radiation and natural convection by substituting a comparable cylinder for the body. Heilman[3] gives the following equations:

$$H_r = 0.1723\, e \left[\left(\frac{T_s}{100}\right)^4 - \left(\frac{T_w}{100}\right)^4 \right] \tag{1}$$

$$H_c = 1.235 \left(\frac{1}{D}\right)^{0.2} \times \left(\frac{1}{T_m}\right)^{0.181} \times \left(T_s - T_a\right)^{1.266} \tag{2}$$

where

H_r = heat loss by radiation, Btu per square foot per hour

H_c = heat loss by convection, Btu per square foot per hour.

T_s = absolute temperature of the body surface, degrees Fahrenheit.

T_w = absolute temperature of the walls, degrees Fahrenheit.

T_a = absolute temperature of the air, degrees Fahrenheit.

$T_m = \frac{T_s + T_a}{2}$

D = diameter of cylinder, inches.

e = the ratio of actual emission to black body emission.

If it be assumed that a normal adult has an average height of 5 ft 8 in. and an average body surface area of 19.5 sq ft, the surface of his body will have the same area as that of a cylinder 5 ft 8 in. long with a diameter of 13.15 in. The value of e for skin and clothing is practically 0.95. T_a and T_w are each taken as 72 F, or 532 Absolute. The sum of H_r and H_c is taken to be 15.4 Btu per square foot per hour, which is derived as the normal rate of heat loss due to convection and radiation from a sedentary individual by dividing his total sensible heat loss by his area. Solution of

[2] A Study of Body Radiation, by L. B Aldrich (Smithsonian Miscellaneous Collections, Vol. 81, No 6, December, 1928).

[3] Surface Heat Transmission, by R. H Heilman (*Trans A.S.M.E., Fuels and Steam Power Section*, Vol 51, No 22, September-December, 1929).

Equations 1 and 2, using average figures as outlined, gives a value of approximately 83 F for the normal temperature of the body surface. This agrees more closely with the values obtained by Aldrich than with the 75 F used by British investigators.

British Equivalent Temperature

The British Equivalent Temperature (BET) is the temperature of an environment which is effective in controlling the rate of sensible heat loss from a sizable black body in still air when the body has a maintained surface temperature of 83 F. The BET is, therefore, a function of both the air temperature and the mean radiant temperature. Its numerical value in a uniform environment (walls and air at the same temperature) is equal to the temperature of the walls and air. In a non-uniform environment (walls and air at different temperatures) the BET is equivalent to that of a uniform environment in which an 83 F surface loses sensible heat at the same rate as it does in the non-uniform environment. As originally defined, the BET was based on a body surface temperature of 75 F, but 83 F has been accepted as giving results more nearly conforming with American practice[4]. The higher the BET the less the heat loss from the body, the rate of loss in still air being approximately proportional to the difference between the BET and the mean body surface temperature.

If the BET were 83 F, there could be no sensible heat loss from a surface at that temperature, so the temperature of a normal body surface would have to rise to a point where the heat generated in the tissues could be dissipated.

When convected heat is used, the temperatures of the air and walls are nearly the same, and the optimum value of the BET from the physiological point of view is 72 F. Under these conditions the mean surface temperature of a normal body would have the optimum value of 83 F because the rate of heat loss by radiation and convection would be 15.4 Btu per square foot per hour and that by evaporation 5.1 Btu per square foot per hour, which would just balance the rate of heat production of 20.5 Btu per square foot per hour. This BET of 72 F in a uniform environment is exactly equivalent to the *effective temperature* of 66 F as defined by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS (see Chapter 3), because, in a uniform environment, a dry-bulb temperature of 72 F in still air with a relative humidity of 30 per cent gives an *effective temperature* of 66 F, which has been determined to be the optimum.

APPLICATION METHODS

There are two general methods of applying radiant heating, as follow:

1. *By warming the interior surfaces of the building.* Pipe coils are embedded in the concrete or plaster of the walls, ceiling or floors, the heating medium being hot water or, in some cases, steam. This has the effect of warming the entire concrete or plaster surface in which the pipes are embedded. Since the temperature of the heating medium should not exceed about 120 F on account of the possibility of cracking the plaster, the

[4]Application of the Eupatheoscope for Measuring the Performance of Direct Radiators and Convectors in Terms of Equivalent Temperatures, by A. C. Willard, A. P. Kratz, and M. K. Fahnestock (A S H V E TRANSACTIONS, Vol 39, 1933)

area of the panel must be sufficient to supply the requisite quantity of heat at this low temperature When carefully designed, this method produces comfortable and economical results.

2. *By attaching separate heated plates or panels to the interior surfaces of the structure.* These plates or panels are placed either in an insulated recess flush with the surface of the walls or ceiling or bolted on its face They may be decorated as desired As it is difficult to make an invisible joint between the edge of such a plate and the plaster, it is common to use a frame of plaster, wood, metal or composition around the panel These plates may be placed either on the ceiling or the wall, or in some cases as a margin around the edge of the floor. If floor heating is required the temperature over the whole area should not exceed 70 F.

If the entire warm surface is installed at one end of the room there may be a marked difference between the BET on the two sides of a body in the room. It is usually desirable therefore that the heat be distributed at different points in the room so that no uncomfortable effects will be felt from unequal heating.

CALCULATION PRINCIPLES

The calculations for radiant heating are entirely different from those for convective heating. The purpose of the latter is to determine the rate of heat loss from the room by conduction, convection, and radiation when maintained in the desired condition; radiant heating involves the regulation of the rate of heat loss per square foot from the human body.

The first step in the calculations for radiant heating is to ascertain the necessary mean radiant temperature (MRT); next, the size, temperature, and disposition of the heating surfaces required in the room to produce this MRT are estimated; and after this the determination of the convective heat is made.

Mean Radiant Temperature

If the whole of the interior surface of a room were at the same temperature, this temperature would represent the MRT. Such a condition seldom exists, however, since the actual surface temperature in any heated space having surfaces exposed to the outer air varies greatly for different sides of the enclosure. It is therefore necessary to ascertain by calculation the mean of these interior surface temperatures.

The mean temperature in this sense is not the arithmetic average of the actual thermometric temperatures of the surfaces, but the temperature corresponding to the average rate of heat emission per square foot of surface. The temperature corresponding to this mean emission can be taken from Table 1. Conversely, the emission at different temperatures and also the emissivity factors can be obtained from this table. For instance, 1 sq ft of surface at 50 F will emit 104.9 Btu per square foot per hour to surroundings at absolute zero if the emissivity of the surface is 0.9.

If the area in square feet of each part of the space is multiplied by the emission value corresponding to its actual temperature, and these products are added together, the gross amount of radiant heat discharged into the room by the wall surface per hour is obtained. This quantity, divided by the total interior surface, gives the average amount of heat coming into the room from the surface of the walls per square foot of surface per hour.

Interpolating in Table 1, the total radiation from a surface at 83 F for

TABLE 1 TOTAL BLACK BODY RADIATION TO SURROUNDINGS AT ABSOLUTE ZERO[a]

Body or Mean Radiant Temperature Deg Fahr	Radiation in Btu per square foot per hour emitted to surroundings with a temperature of absolute zero by bodies at various temperatures and with emissivity factor e				Body or Mean Radiant Temperature Deg Fahr	Radiation in Btu per square foot per hour emitted to surroundings with a temperature of absolute zero by bodies at various temperatures and with emissivity factor e			
	e 1 00	e 0 95	e 0 90	e 0 80		e 1 00	e 0 95	e 0 90	e 0 80
30	99.3	94 3	89 4	79 4	71	136 5	129 6	122 9	109 3
35	103 5	98.3	93 2	82 8	72	137.4	130 5	123.6	109.9
40	107 6	102 4	96 8	86 1	73	138 4	131.5	124 5	110 6
45	112 1	106 5	100 9	89 7	74	139 6	132 6	125 6	111 7
46	112.9	107.3	101 6	90 4	75	141 0	133 9	126.9	112.8
47	113 9	108 2	102.5	91 1	80	146 6	139 4	132 0	117.4
48	114.8	109.1	103.4	91 9	85	152 3	144 6	137.1	121 9
49	115.6	109 9	104.1	92 4	90	157 9	149.9	142.1	126 4
50	116 5	110 6	104 9	93 2	100	169 6	161 1	152.6	135.7
51	117.5	111 6	105 8	94 0	110	181.6	172 5	163.5	145.4
52	118.4	112.5	106 5	94 7	120	194.8	185 0	175.4	155.9
53	119 4	113 4	107 4	95 5	130	210 1	199 6	189.1	168.1
54	120.2	114 2	108 2	96 2	140	223.2	212 1	201.0	178.5
55	121.1	115 1	109 0	96 9	150	237.1	225 2	213.5	189.7
56	122.1	116.0	109 9	97 7	160	251.1	238 8	226.0	201.0
57	123.1	117.0	110.9	98 5	170	270.5	257.0	243.5	216.4
58	124 0	117.8	111.6	99 2	180	288.0	273 8	259 1	230.4
59	124.9	118 6	112.4	99.9	190	306 5	291 0	275.8	245.1
60	125.8	119.5	113.4	100 7	200	325.2	309 0	292.8	260.3
61	126 6	120 3	114.0	101.4	210	348.0	330.6	313 1	278.4
62	127 7	121 4	114 9	102.2	220	371.5	353 0	334 4	297.1
63	128 6	122.2	115.8	102.9	250	437.8	415.9	394.0	350.2
64	129.6	123 1	116 7	103 7	300	575.0	546.1	517.5	460.0
65	130.5	124 0	117.5	104 4	350	740 0	703.0	666.0	592.0
66	131.6	125 0	118.4	105.4	400	942.1	895 0	847.5	753.5
67	132.5	125.9	119 3	106.0	450	1176 0	1117.0	1059.0	941.0
68	133 5	126.8	120 1	106.8	500	1464 0	1390.0	1318.0	1171.0
69	134 5	127.8	121.1	107.6	550	1791 0	1701.0	1613.0	1434.0
70	135.5	128.8	121.9	108 4	600	2405 0	2284.0	2165.0	1925.0

[a]These factors are calculated from the formula

$$Q = e\left(\frac{0\,1723 \times T^4}{100{,}000{,}000}\right)$$

where

Q = total black body radiation, Btu per square foot per hour.
e = emissivity
T = absolute temperature, degrees Fahrenheit.

an emissivity of 0.95 is 142 Btu per square foot per hour. The difference between 142 Btu and the average amount of heat coming into the room is the amount which will be lost per square foot per hour by radiation from a body at 83 F. If a rate at which it is desired that heat be lost from the body by radiation and convection be assumed, the mean radiant emission from the walls required to give the desired result can be determined from Table 1, as can also the required air temperature for the corresponding convective effect.

The determination of the amount of radiant heating surface needed in a room requires knowledge of the climate, the type of structure, the type of heating, and the surface temperature of the walls. This problem can be solved only on an empirical basis. After some experience, however,

it is possible to estimate these variables with a considerable degree of accuracy for any climate or construction.

Assume that a mean radiant temperature of 65 F is desired. Table 1 shows that with all the walls at this temperature, and with an emissivity of 0.95, the gross heat emission is 124 Btu per square foot per hour. The total emission of radiation into the room from that surface would therefore be $A \times 124$, where A is the total inside area of the room. This is the *desired* emission.

If the whole area be divided into a number of different parts which are each at a uniform temperature—a_1, a_2, a_3,—and each is multiplied by the value of the heat emission corresponding to that temperature, and if all these products are added together, their sum will represent the total *actual* emission of radiation into the room at these temperatures without the aid of any hot surface.

The difference between the desired emission and the actual emission represents the additional heat which must be supplied by the hot surface. The temperature of the proposed hot surface must then be selected, and its emission per square foot at that temperature determined from Table 1. This emission is divided into the additional amount of heat needed, adjusted for the fact that the heating units will shield the walls behind them, and the quotient obtained will be the area of the required heating surface.

It is evident that this method of calculation is approximate, and depends for its accuracy on a correct estimate of the ultimate surface temperatures attained by the actual wall surfaces.

It is necessary also to calculate how much heat will be given off by the same surfaces by convection, and thereby to determine whether this amount of convected heat will warm entering ventilating air to the temperature maintained. If it will not, additional convection surfaces must be introduced to make up the deficiency.

MEASUREMENT OF RADIANT HEATING

Convection heating, having as its object the raising of the air temperature to a specified degree, must be measured by thermometric methods which indicate essentially the air temperature, and not the rate of heat loss from the human body. Radiant heating, having as its object the control of the rate of heat loss from the human body, can be measured only by methods which basically are calorimetric, that is, which measure directly the rate of heat loss from an object maintained at the temperature of the body, irrespective of air temperature.

The apparatus for this purpose consists essentially of a hollow sphere, or cylinder, containing a fluid which can be maintained accurately at 83 F (the accepted mean surface temperature of the human body), with an accurate means of measuring the rate of heat supply required to maintain the temperature at that exact point. The latter measurement can be made with sufficient accuracy by electrical methods. Although a BET of 72 F is desirable, the mean radiant and air temperatures may both vary, provided the heat loss by radiation and convection from a surface at 83 F is maintained at the rate of 15.4 Btu per square foot per hour,

which corresponds to $\frac{15.4}{3.415} = 4.5$ watts per square foot of exposed surface.

This instrument, the *eupatheoscope*, can readily be adapted as a thermostat by electrical control to shut off or turn on heat when the critical temperature of 83 F in the vessel is increased or decreased. A modification of the instrument is called the *eupatheostat*.

Another instrument for maintaining comfort conditions is at present available only in a model adapted to British practice as it is designed for a temperature of 75 F. It consists of a blackened copper sphere of approximately 6 in. diameter in which is housed a cylindrical sump containing a volatile liquid. In operation, a small electric heating coil drawing about 5 watts creates in the sphere a vapor pressure which is constant as long as the heat losses from the sphere are standard. If the temperature of the air or the MRT becomes too high for comfort, a greater pressure is created, owing to a smaller loss of heat from the sphere. This increase of pressure acts on a diaphragm and shuts off the supply of heat to the room.

For testing work, the *globe thermometer* is a very useful instrument. It consists of an ordinary mercury thermometer, with its bulb placed in the center of a sphere from 6 in. to 9 in. in diameter, usually made of thin copper and painted black. The temperature thus recorded is termed the *radiation-convection temperature*.

EXAMPLE

Example 1. The surface areas, temperatures, and emissions for a room having a volume of 5760 cu ft are given in Table 2. The figures for temperatures are fairly representative of American practice with well-built walls, and are based on an emissivity of 0 95 which approximates that of most paints and building materials.

TABLE 2 SURFACE AREAS, TEMPERATURES, AND EMISSIONS FOR A ROOM OF 5760 CU FT

	AREA SQ FT	ASSUMED SURFACE TEMPERATURE (DEG FAHR)	HEAT EMISSION (BTU PER SQ FT PER HOUR)	TOTAL HEAT EMISSION FROM AREA (BTU PER HOUR)
External Wall	297	50	110.6	32,850
Glass	279	45	106.5	29,710
Inner Wall	480	55	115.1	55,250
Ceiling	480	55	115.1	55,250
Floor	480	55	115.1	55,250
Total	2016			228,310

The mean radiant temperature of the room is $\frac{228,310}{2016} = 113.2$ Btu per square foot per hour which, as seen from Table 1, corresponds to an MRT of 53 F for an average emissivity of 0 95.

For an average individual having a body surface of 19.5 sq ft, under conditions of comfort with a body surface temperature of 83 F, the heat given off by radiation may be determined by means of Equation 1 as 217 Btu per hour, or 11.1 Btu per square foot per hour. This corresponds to an environmental emission of $142 - 11.1 = 130$ 9 Btu per square foot per hour, and, according to Table 1, to an MRT of 72 F.

If this body be placed in the room described, it will lose heat at the rate of $19.5\ (142 - 113.2) = 562$ Btu per hour. This loss is 345 Btu per hour, or 17.7 Btu per square foot per hour, more than the rate of heat loss for comfort, which is only $19.5\ (142 - 130.9) = 217$ Btu per hour.

In order to determine the amount of radiating surface necessary to maintain the MRT at 72 F, assume the surface temperature of the hot plates to be installed to be 200 F, which is approximately the temperature they would have if heated by steam

The 2016 sq ft total area of the surfaces of the room multiplied by 130 9, which is the emission in Btu per square foot per hour necessary to maintain a body surface temperature of 83 F, gives a total desired emission of 263,S90 Btu per hour It is necessary to supply enough radiant heating surface to increase the total actual mean radiant heat emission by the room from 228,310, as shown in Table 2, to the 263 890 Btu desired. The additional heat needed is the difference between these figures, or 35,580 Btu. Since, from Table 1, the emission per square foot at 200 F is 309 Btu, the required radiant heating surface needed is $\frac{35,580}{309} = 115$ sq ft The effect of this surface suitably placed would be to raise immediately the mean radiant temperature to the required degree and to maintain it at that value as long as the surfaces remained at the values assumed.

In the solution of this particular example, the radiation loss from the human body was selected as 217 Btu per hour, which is that taking place under optimum comfort conditions, with a body surface temperature of 83 F in a uniform environment at 72 F. The mean radiant temperature necessarily was 72 F. If the optimum BET of 72 deg Fahr is desired, an air temperature of 72 F also must be maintained. If it is desired to maintain a lower air temperature than this, a mean radiant temperature greater than 72 F must be selected and the radiation loss from the individual must be recalculated from Equation 1.

The calculation may be simplified by preparing tables showing, at the usual temperatures, the area of hot surface required to bring each square foot of actual wall surface at various temperatures up to a general standard of from 60 F to 70 F. It would then be necessary only to multiply the respective areas by the appropriate factors, and to add the results, to obtain the required total.

REFERENCES

Room Warming by Radiation, by A. H. Barker (A S H.V E. Transactions, Vol. 38, 1932).

Panel Warming, by L. J. Fowler (A.S.H.V.E. Transactions, Vol. 36, 1930).

Calculations for Radiant Heating, by T. Napier Adlam (*Heating and Ventilating*, October, 1931).

Principles of Calculation of Low Temperature Radiant Heating, by A. H. Barker (*Proceedings of The Institution of Heating and Ventilating Engineers*, London, Vol. 30, 1931).

What will be the Future Development of Heating and Air Conditioning, by W. H. Carrier (*Heating, Piping and Air Conditioning*. January, 1933).

Method of Installing the Panel Heating System in the British Embassy Building (*Heating, Piping and Air Conditioning*, July, 1934)

Panel Heating, by C. M. Oates (*Proceedings of Institution of Heating and Ventilating Engineers*, London, Vol. 30, 1931).

Notes on Electric Warming with Special Reference to Low Temperature Panel Systems, by R. Grierson (*Proceedings of Institution of Heating and Ventilating Engineers*, London, Vol. 28, 1929)

Radiant Heat, by A. F Dufton (*Proceedings of Institution of Heating and Ventilating Engineers*, London, Vol. 30, 1931).

Radiant Heat, by A. F. Dufton (*Proceedings of Institution of Heating and Ventilating Engineers*, London, Vol. 31, 1932).

Notes on the Theory of Radiant Heating, by C. G. Heys Hallett (*Proceedings of Institution of Heating and Ventilating Engineers*, London, Vol. 29, 1930).

PROBLEMS IN PRACTICE

1 ● Where did radiant heating derive its name?

The term radiant heaters was introduced about [illegible] years ago to designate flat heating surfaces made to give off practically all their heat by radiant ether waves instead of relying on convected warm air.

2 ● What is actually meant by radiant heating and what are its underlying principles?

The term radiant heating now applies to methods of heating where instead of heating the air in a room to a predetermined temperature flat heating surfaces are placed in a room so that the average effective temperature of walls ceiling, glass and floor surfaces exposed to the body is just sufficient to prevent the body losing too much heat by radiation It takes into consideration that the body generates more heat than it requires, so that it does not require any heat from without The surplus heat however must be given off according to the physiological requirement of the body.

3 ● What kind of heating surfaces are in general use?

The heating units may have flat iron surfaces heated with steam or hot water and placed in side walls or under windows, or they may be supported on the ceiling and suitably decorated and connected as ordinary steam or hot water radiators. Hot water pipes may be embedded in the floor, walls or ceiling, and when in the floors they may be covered with concrete and wood blocks or other suitable material, the finish of the surface being more important than the composition of the material. When in the ceiling or walls, they can be covered with plaster to harmonize with the rest of the room Electrical radiant heaters are made by embedding resistance elements in porcelain, or electric conductors may be woven into thick paper and fastened to the walls and ceilings, electric wires may be woven with tapestry to form portable screens for local heating.

4 ● What surface temperatures are generally used?

Where hot water pipes are embedded in plaster, the surface temperature varies from 90 to 130 F. Where flat iron plates are used these may vary from 140 to 220 F. With electric resistances embedded in porcelain the surface temperature may vary from 200 to 500 F. High surface temperatures are not recommended.

5 ● What kind of heat rays are commonly generated for radiant heating?

All heat rays are generally assumed to be the same as light rays; they travel at the speed of light, but they are invisible and longer. The rays used in heating are 0.00005 to 0 0001 in. long, compared with invisible red rays of about 0.000027 in.

6 ● When and why does the human body feel cold?

The body feels cold not only when it loses heat at a greater rate than it can generate it but also when heat is abstracted from the body disproportionately. Since the human body generates more heat than is necessary, it is only necessary to provide conditions that will regulate the correct ratio of losses; the provision of suitable radiant heating surfaces is one way to establish these conditions.

7 ● Is the heat generated in the body affected by action? If so, does it vary greatly?

Yes. With hard work or energetic exercise, the total heat generated in the body may be 5 to 6 times that generated when it is at rest.

8 ● Why is the heat loss from the body by radiation important?

The heat loss by radiation is proportional to the fourth power of the temperature difference between the surface of the body and the average surface temperature of the surrounding walls, windows, etc ; whereas, for convection losses, it is only proportional to the 1.25 power.

9 ● What is the approximate relation for heat losses?

Heat losses from the body when in a sedentary position are approximately as follows radiation 49 per cent, convection 23 per cent, evaporation 15 per cent, respiration 11 per cent, and miscellaneous 2 per cent Actually, it depends upon age, environment and other conditions.

10 ● Differentiate between radiant and convection heating.

The primary function of radiant heating is to regulate the loss of heat from the body without unduly heating the air, while in convection heating it is generally the function of the heating medium to transfer the heat to the air and thence to the occupant of the room.

11 ● What generally is the air temperature necessary to give equal comfort effect for sedentary conditions?

With radiant heating, 64 to 66 F. With convection heating, 70 to 72 F

12 ● Why is there a saving in fuel consumption with radiant heating?

A saving is effected because the differential between inside and outside temperature is much less for radiant heating Less ventilating air is necessary and this can be supplied at a much lower temperature.

13 ● What is the mean normal surface temperature of the human body as determined for the United States?

83 F.

14 ● Describe how to calculate the required amount of radiant heating surface.

a. Obtain the mean heat emission in Btu per square foot per hour for room surfaces X, using values in Table 1, and surface temperatures as shown in second column of Table 2.

b. Deduct X from 142 (142 being the emission per square foot given off by the human body at 83 F surface temperature) = Y in Btu per square foot per hour

c. From (142-X) deduct 11 1 (11.1 being the average radiation which the human body should lose per square foot for comfort conditions) = (142-X-11 1) = Z

d Multiply total interior surface of room by Z and divide by the emission per square foot from radiant heater, giving the surface S of radiant heater in square feet

15 ● Give a simple formula to calculate radiant heating surface required, and explain.

$$S = \frac{(142 - X - 11.1)\,A}{B}$$

where

S = surface of radiant heater, square feet
142 = Heat emission, Btu per square foot per hour which the human body would give off at 83 F, with surroundings at absolute zero
X = mean heat emission, Btu per square foot per hour from surfaces of room
11 1 = heat emission, Btu per square foot per hour from human body.
A = total surface, square feet of walls, ceilings, windows, etc , in room.
B = heat emission per square foot from radiant heater surface

16 ● What natural evidence have we that air temperature alone is no criterion of comfort and that radiant heat affects the body more quickly?

When standing in the sunshine on a cool spring day, a person feels perfectly comfortable, but when a cloud passes over the sun, he instantly feels much cooler as the shadow reaches him. A shielded thermometer recording the temperature of the air shows no reduction in air temperature in so short a period, so that the person actually feels a sensation of cold which an ordinary thermometer cannot register This shows that light and heat rays are shut off simultaneously and travel at the same speed, it also proves that radiant rays affect the comfort of the body quicker than air temperature does.

Chapter 39

ELECTRICAL HEATING

Resistors, Heating Elements, Electric Heaters, Unit Heaters, Central Fan Heating, Electric Steam Heating, Electric Hot Water Heating, Electric Hot Water Heating for Domestic Supply, Industrial Heating, Cooling and Reverse Cycle Heating by Electric Refrigeration, Auxiliary Electric Heating, Control, Calculating Capacities, Power Problems, Insulation, Electric Heating Data

ELECTRIC heating is steadily assuming a more important place in heating, ventilating and air conditioning installations, accelerated in many territories by the load building efforts of the utilities which usually include reduced rates to encourage such installations. Electrical heating has a logical place in the heating industry because of its features of flexibility, cleanliness, safety, convenience and ease of control. Electrical heating practice has many basic principles in common with fuel heating, but there are also important differences. When heat units are delivered to each room by wire, no combustion process is necessary, either at a central plant or at the individual room units. The maximum output of an electric heater is a fixed constant, unaffected by the temperature of the surrounding air and it follows that the maximum total load on an electrical heating system is the total wattage of connected electric heaters, regardless of weather conditions. The real obstacle to the more general adoption of electric heating for buildings is the cost of the electricity itself. Because the heat units produced electrically are more costly, their conservation is of more relative economic importance than with fuel heating, so that sponsors of electric heating give greater attention to temperature-insulated building construction.

All heat is a form of energy. Fuels hold stored chemical energy which is released into heat by combustion. Electrical power is a form of energy which can be released into heat by passing it through a resisting material. Both fuel and electric heating have two divisions: *first*, the conversion of energy into heat; *second*, the distribution and practical use of the heat after it is produced.

In converting the chemical energy of fuels into heat by combustion, there is necessarily a considerable variation in thermal efficiency. This is not true, however, when converting electric power into heat, because 100 per cent of the energy applied in the resistor is always transformed into heat. In electric heating practice the engineer need not be concerned about efficiencies of heat production, but rather about efficiencies of heat utilization. It is the engineer's problem to distribute the electrically produced heat units in such manner as to obtain conditions of maximum comfort with the minimum consumption of electricity.

DEFINITIONS

Definitions of terms used in fuel heating are given in Chapter 44. The following terms apply particularly to electric heating:

Electric Resistor: A material used to produce heat by passing an electric current through it.

Electric Heating Element: A unit assembly consisting of a resistor, insulated supports, and terminals for connecting the resistor to electric power.

Electric Heater: A complete assembly of heating elements with their enclosure, ready for installation in service.

TYPES OF RESISTORS

Solids, liquids, and gases may be used as resistors, but most commercial electric heating elements have solid resistors, such as metal alloys, and non-metallic compounds containing carbon. In some types of electric boilers, water forms the resistor which is heated by an alternating current of electricity passing through it. One of the more common resistors is nickel-chromium wire or ribbon which, in order to avoid oxidation, contains practically no iron.

HEATING ELEMENTS

Commercial electric heating elements are divided into open type elements, enclosed type elements, and cloth fabrics. *Open type elements* have resistors exposed to view. The resistors may be coils of wire or metal ribbon, supported by refractory insulation, or they may be non-metallic rods, mounted on insulators. Open type elements are used extensively for operation at high temperatures when radiant heat is desired. They are also frequently used at low temperatures for convection and fan circulation heating, especially in large installations.

Enclosed type elements have metallic resistors embedded in a refractory insulating material, and encased in a protective sheath of metal. Fins or extended surfaces may be used to add heat-dissipating area. Enclosed elements are made in many forms, such as strips, rings, plates, and tubes. Strip elements are used for clamping to surfaces requiring heat by conduction, and in convection and fan circulation air heaters. Ring and plate elements are used in electric ranges, waffle irons, and in many small air heaters. Tubular elements may be immersed in liquids, cast into metal, and, when formed into coils, used in electric ranges and air heaters. *Cloth fabrics* woven from flexible resistor wires and asbestos thread, are used for many low temperature purposes.

ELECTRIC HEATERS

Electric heaters may be divided into three groups, conduction, radiant and convection.

Conduction electric heaters, which deliver most of their heat by actual contact with the object to be heated, are used in such applications as aviators' clothing, hot pads, foot warmers, soil heaters, ice melters, and pipe heaters. Conduction heaters are useful in conserving and localizing heat delivery at definite points. They are not suitable for general air heating.

Radiant electric heaters, which deliver most of their heat by radiation, have high temperature incandescent heating elements and reflectors to concentrate the heat rays in the desired directions. The immediate and pleasant sensation of warmth which is caused by radiant heat makes this type desirable for temporary use where the heat rays can fall directly

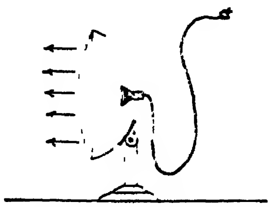


FIG. 1. PORTABLE RADIANT ELECTRIC HEATER

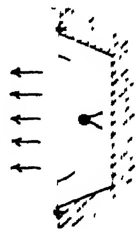


FIG. 2. RADIANT ELECTRIC HEATER RECESSED IN WALL

upon the body. They are not satisfactory for general heating, as radiant heat rays do not warm the air through which they pass. They must first be absorbed by walls, furniture, or other solid objects which then give up the heat to the air. The location of radiant heaters is important. They should never face a window because some rays would pass through the glass and be lost. Figs 1 and 2 show common types of portable and wall-mounted radiant heaters.

Gravity convection electric heaters, designed to induce thermal air circulation, deliver heat largely by convection, and should be located and used

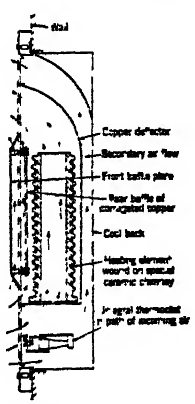


FIG. 3. CONVECTION ELECTRIC HEATER RECESSED IN WALL

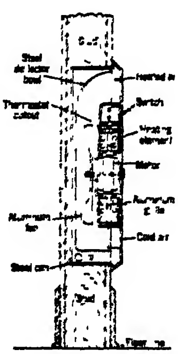


FIG. 4. FAN CONVECTION ELECTRIC HEATER RECESSED IN WALL

in much the same manner as steam and hot water radiators or convectors. They generally have heating elements of large area, with moderate surface temperature, enclosed to give proper stack effect to draw cold air from the floor line (Figs. 3 and 4). The flexibility possible with electric heating elements should discourage the use of secondary mediums for heat transfer. Water and steam add nothing to the efficiency of an electric heater and entail expensive construction.

Fan convection electric heaters, which include a built-in fan unit, circulate room air over the heating elements at 300 to 400 fpm. Heaters of this type are manufactured in three designs, portable, wall hung and wall insert. The fan draws air from the floor zone through a lower grille, forces it over the heater elements and out through an upper grille into the breathing zone. The improved heating effectiveness of this type electric heater has done much to promote electric heating for residences.

UNIT HEATERS

Fan unit electric heaters, similar to steam unit heaters, having a propeller type fan mounted behind the heating element, are made in many styles and can be located and used much the same as steam unit heaters

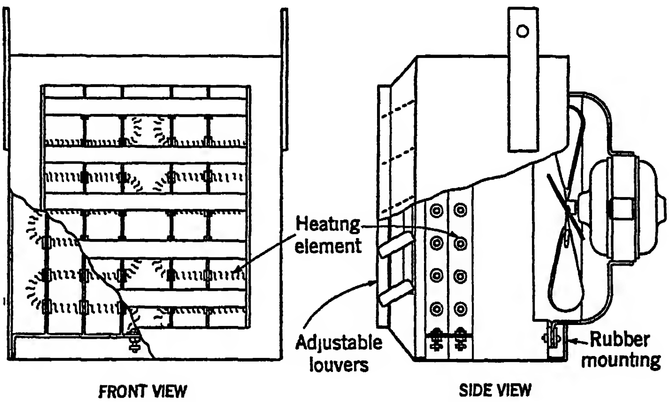


FIG. 5. FAN UNIT ELECTRIC HEATER

(Figs. 5 and 6). These electric unit heaters are used in industrial plants, sub-stations, power houses, pumping stations, etc., where large initial use of electricity for power purposes earns a low rate for electric heating. Portable unit heaters (Fig. 7) of this type are useful for temporary work, such as drying out damp rooms, or for warming rooms during construction.

CENTRAL FAN HEATING

Electric heating elements can be used for the prime source of heat in a central fan electric heating system or in the heating phase of a complete air conditioning system. They can be used in the same manner as steam served heating units for tempering, preheating or reheating the air at the main supply fan location and as booster heaters at the delivery terminals of the duct system. In the humidification phase of air conditioning electric heating elements can be used to provide moisture by the evaporization of water, or for controlling air washer dew-point temperatures when mounted as preheating units on the intake side of the air washer.

In coördinating the input of heat energy and the volume of air circulation, a basic difference between electric heating and steam heating enters into the problem. Steam is approximately a constant-temperature

source of heat for any given pressure as a change in air volume flowing over steam coils does not greatly affect the temperature of the delivered air. The amount of steam condensed (heat input) varies in proportion to the air volume, but the surface temperature of the steam coils remains about the same. Electric heat is quite different, being a constant source of energy. If the volume of air flow over electric heating elements is changed, and no change is made in the electrical power connections, there will be a corresponding change in the temperature of the air delivered because the electrical energy input remains constant and the surface temperature of the heating elements will vary as is necessary to force the air to accept all the heat. With electric heat the total heat is constant

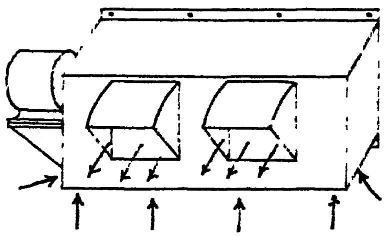


FIG 6 LARGE INDUSTRIAL TYPE FAN UNIT ELECTRIC HEATER

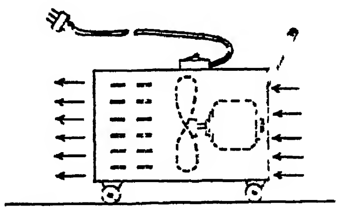


FIG. 7. LARGE INDUSTRIAL TYPE PORTABLE FAN UNIT ELECTRIC HEATER

unless some compensating action is performed by control. Automatic modulation to vary the electrical heat input and synchronize it properly with the air flow has been successfully applied to central fan systems.

ELECTRIC STEAM HEATING

Electric steam heating differs from fuel heating only in the use of *electric boilers* to generate steam. Electric steam boilers are entirely automatic and are well adapted to intermittent operation. Small electric boilers usually have heating elements of the enclosed metal resistor type immersed in the water. Boilers of this construction may be used on either direct or alternating current since the heat is delivered to the water by contact with the hot surfaces. To lessen the likelihood that the heating elements will burn out, they are made removable for cleaning off deposits of scale which restrict the heat flow. Large electric boilers are usually of the type employing water as the resistor. Only alternating current can

be used, as direct current would cause electrolytic deterioration. Large boilers of this kind have electrodes immersed in the water where heat is generated directly. Electric steam boilers are useful in industrial plants which require limited amounts of steam for local processes, and for sterilizers, jacketed vessels and pressing machines which need a ready supply of steam. It sometimes is economical to shut down the main plant fuel burning boilers when the heating season ends, and to supply steam for summer needs with small electric steam boilers located close to the operation. In general, electric steam heating is confined to auxiliary or other limited applications. If the heating system is designed to use electricity exclusively, steam generating or distributing equipment is superfluous.

ELECTRIC WATER HEATING[1]

Electric water heating, using an electric boiler in place of a fuel burning boiler, like electric steam heating, is generally confined to auxiliary or other limited applications. However, the use of insulated water storage tanks in which to store heat generated by electricity during off-peak hours at extremely low rates, is a development which may bring electric water heating systems into general use. This type of system, known as the off-peak hot water storage heating system, offers striking possibilities in the Southern and Pacific Coast States where the degree-day heating load is low and the Utilities have developed or are now developing large blocks of hydro-electric power which can be marketed at 1 cent per kwhr or less.

In this system of heating, the primary storage tank is simply a large, well-insulated, pressure type steel tank, equipped with immersion electric heating elements connected to line with automatic time switches, which also have automatic limit controls for temperature and pressure. The heating system installed in the home or other type building may be of the individual radiator type or fan-served indirect type, with provisions for the heating and humidification phases of an air conditioning system.

The following types of off-peak heating systems are used:

1. Hot water with gravity circulation.
2. Hot water with forced circulation.
3. Warm air utilizing forced circulation.
4. Unit heaters utilizing hot water as the heating medium
5. Warm air utilizing stored heat in connection with waste heat.

The essential parts of all types of off-peak storage systems are:

1. Heavily insulated storage tank.
2. Immersion heating elements.
3. Automatic charging control.
4. Heat distributing system.
5. Automatic temperature control.

The insulated storage tank, the heating elements and the charging control are practically identical for all types of off-peak systems. Only

[1]Performance Test of an Electric Hot Water Heating Boiler, by John James (A S H V E JOURNAL SECTION, *Heating, Piping and Air Conditioning*, September, 1936)

the heat distributing system and the room temperature control differ to any extent.

The size of the storage tank is based upon the heat losses of the building, the heating schedule, the hours of heating from storage, and allowance for heating up. In new buildings and where openings permit, factory fabricated tanks are used. Otherwise, the tanks are welded at the final location.

In *hot water systems*, the storage tank supplies hot water to standard heating units (radiators or convectors installed in the building. The water in the storage tank is heated to approximately 250 F. In the system using gravity circulation, the temperature of the hot water supplied to the heating units is varied by the action of a mixing valve under control of the room thermostat. In larger installations, the circulation is obtained by means of a pump under thermostatic control. In this case, the temperature of the water supplied to the heating units is regulated by mixing the proper amount of return water with the 250 F water from the storage tank through a suitable by-pass.

The *warm air system* makes use of extended heating surface and a plenum chamber assembly (*i.e.*, a central fan system) which is usually installed in the basement alongside the storage tank. The warm air supply ducts are run in the usual manner from the top of the plenum chamber to the registers in the rooms. The return air is brought back through cold air ducts to the inlet side of the fan or blower. Temperature regulation is obtained by thermostatic control of the blower and the valve admitting hot water to the heating units. Air cleaning and humidifying apparatus is readily incorporated in the system.

Unit heaters are used in connection with off-peak storage heating systems for industrial buildings. For sub-stations and certain types of industrial buildings, the electric storage heater is used to supply hot water to unit heaters placed in various parts of the building. Certain sections of the building, such as offices and rooms which are partitioned off from the larger areas, may be heated by ducts leading from indirect heating units supplied with hot water in parallel with the unit heaters.

Warm air supplied from off-peak storage is sometimes used in conjunction with a system designed to utilize waste heat. In this application, heating units are installed in the waste heat ducts, which supply waste heat from the losses of synchronous condensers and other apparatus. When the waste heat does not provide sufficient heating, the air temperature is boosted by supplying the proper amount of hot water from the storage tank to the heating units. Controls are provided, which not only make the entire operation automatic, but insure maximum recovery from the waste heat source.

ELECTRIC WATER HEATING FOR DOMESTIC SUPPLY[3]

Electric water heaters of the automatic storage type for domestic hot water supply are simple and reliable, and in many sections of the country very low electric rates have been established by the electric utilities to secure this load. In some districts, rate schedules divide the current

[3]Test Results of Electric Water Heaters, by C. G. Hillier (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, November, 1936).

used for water heating into two classifications, regular and off-peak. A time switch automatically limits use of the off-peak heating element to the hours of off-peak load, while the regular heating element is a stand-by at all times. Storage of this two-element type of water heater is larger than average to carry over the periods when the off-peak element is timed out, without too frequent demands on the regular heating element which takes the high domestic lighting service rate. Some utilities now offer a schedule which, beyond a stipulated minimum, lowers the rate for all service if an electric water heater is installed.

INDUSTRIAL ELECTRICAL HEATING

Electric heating elements have been successfully developed for innumerable industrial furnaces such as annealing, brazing, carburizing, enameling, forging, ceramic firing, hardening, metal melting, nitriding and process heating. Industrial ovens where precise control of high temperatures is necessary can be very successfully operated with electric resistance elements where temperatures as high as 700 F are required. Electric heaters for heating oil to high temperatures for secondary circulation in process work are used as a substitute for superheated steam. Special oil can be electrically heated as high as 600 F and pumped at a pressure just sufficient to cause flow. When used in heating coils or jacketed vessels, this gives a safe and convenient automatic system for moderate-sized installations.

COOLING AND REVERSED CYCLE HEATING BY ELECTRICAL REFRIGERATION[3]

Reversed refrigeration is frequently referred to as a *heat pump* since the electric motor driving the refrigerating compressor furnishes the motive power to transfer heat from one temperature to a higher temperature level. The compressor acts as a reversible refrigerating unit to extract heat from the outdoor air in winter and deliver it indoors for heating purposes, and, by a reversal, to extract heat from the indoor air in summer and discharge it outdoors.

In normal use a refrigerating machine is arranged to remove heat and the heat removed is dissipated to the condenser cooling water. The driving energy is converted into heat most of which is added to the heat removed and extracted. In so-called reversed refrigeration the heat removed together with the heat converted from the driving energy is utilized to heat the building. This conservation of the heat converted

---

[3]Cooling Homes, A Field for Refrigeration, by A R Stevenson, presented at the symposium of the Refrigeration with Gas Committee of the *American Gas Association*, April 20, 1926

The Heat Pump, An Economical Method of Producing Low-grade Heat from Electricity, by T G N. Haldane (*Electric Review*, Vol. 105, p 1161-1162, December 27, 1929, and *I E E Journal*, Vol 68, p 666-675, June, 1930).

Edison Building Heated and Cooled by Electricity, by H L. Doolittle (*Power*, Vol 74, p 384, September 8, 1931)

House Heating by Pump with 5 to 1 Pick-up Ratio, by Gilbert Wilkes and R E. Marbury (*Electrical World*, Vol 100, p 828, December 17, 1932).

An All Electric Heating, Cooling and Air Conditioning System, by Philip Sporn and D. W. McLenegan (A S H V E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, August, 1935)

Using the Reversed Cycle Refrigerating Principle for a Self-Contained Heating and Cooling Unit, by Henry L Galson (A S H V E JOURNAL SECTION, *Heating, Piping and Air Conditioning*, October, 1935)

from the driving energy enables the reversed refrigeration to show a better performance in heating service than straight refrigeration can show in cooling service.

For a detailed description of this cycle see Chapter 2 on Refrigeration

AUXILIARY ELECTRIC HEATING

In conjunction with heating systems of other types, an auxiliary electrical heating arrangement is a convenient means of caring for mild days in the spring and fall which require little heat to make a house or building comfortable. Likewise, such electrical heating might be used on abnormally cold days to help out the main heating system and by this means reduce the necessary size of the system.

Because of the feeling of comfort that a radiant type heater gives, bathrooms may be heated electrically with this type of heater while the rest of the house is cared for by some other system. Offices and rooms which require heat at periods when the main heating plant is shut down can be conveniently cared for electrically.

CONTROL

Because the efficiency of electric heat production is the same for large or small units, it is possible to reduce heat waste to a minimum by applying local heating, locally controlled. Wherever radiant heaters are used, thermostats can be used which operate through changes in air temperature but such thermostats can not integrate the combination effect of the radiant and convected heat. More accurate control can be had by the eupatheoscope, described in Chapter 38 on Radiant Heating. For all convection and fan circulation heaters thermostatic control is useful. Heaters up to 5000 watts on 230 volt current can be thermostatically controlled by direct-acting wall thermostats. For larger size heating elements the thermostat should operate a relay which in turn interrupts or closes the circuit to the heating element.

CALCULATING CAPACITIES

The methods of calculating heat losses outlined in Chapters 6, 7, and 8 may be used for electric heating exactly as for fuel heating. The total heat requirements in Btu per hour may then be converted into the electrical rating of an equivalent heating system by using the equation:

$$\frac{\text{Total Btu per hour}}{3415} = \text{kw rating of required electric heating} \tag{1}$$

For comparison with steam radiation:

$$\frac{\text{3415 Btu (one kwhr)}}{240} = 14.2 \text{ sq ft of steam radiation}$$

While many empirical rules based on cubic contents, floor areas, etc., are used in steam heating practice, they should never be used to determine size of equipment for an electrical heating installation.

For electrical heating actual heat losses should be figured accurately because the utility company rates and minimum charge items are commonly based on the size of equipment installed, so that cost of operation might be penalized by rule-of-thumb sizing of the heating equipment.

POWER PROBLEMS

The first point to determine is the cost of the power which is available for electric heating. Unlike fuels, there is no uniform cost for electric power because of the unequal cost of distribution to large and small users. The fact that electricity cannot be economically stored, but must be used as fast as it is generated, makes it impossible to operate power plants at uniform loads; hence, even the time of use may affect the cost of power.

Homes are almost universally supplied with lighting current of 115 volts, which cannot be used economically for any but the smallest heaters. Usually the service lines will not permit more than plug-in devices. The underwriters permit heaters of 1250 watts to be used from approved baseboard receptacles. Where homes have 230 volt service for cooking and water heating, and rates are favorable, larger heaters can be installed. For industrial purposes, heaters should be designed to use polyphase power, which is usually supplied at 230, 460 or 575 volts. All polyphase heaters should be balanced between phases.

INSULATION

The value of building construction which incorporates built-in insulation to reduce the outward heat loss in winter and the inward heat gain in summer has been placed in the spotlight by the increasing adoption of complete air conditioning. With electric heating, adequate insulation is very important. Two groups of 16 homes each, in Southern California, are compared in the following actual figures taken from operating records of not less than two years, and in some cases up to eight years.

Item	Insulated Homes	Uninsulated Homes
Average Size of Home: Number of Rooms	8.7	8.7
Number of Bathrooms	2.6	2.3
Average kw Connected Load per Home:		
Range	9.6	9.3
Water Heater	6.3	5.9
Heating	34.0	32.7
Total kw	49.9	47 9
Annual kwhr for Heat per Home	4562	8223
Annual kwhr per kw Heat Connected	134	252

Cost of Heating

If the consumer obtains heating at 2 cents per kwhr his annual cost of heating an insulated home would be $91.24, as compared with $164.46 for heating an uninsulated home. Thus, insulation in the average home creates an annual saving of $73.22, representing 45 per cent of the annual heating cost if the house is not insulated.

Cost of Insulation

When the average cost of insulating a home is from $25.00 to $50.00 per room, or from $200.00 to $400.00 for a home of the size used herein, it is therefore apparent that savings in heating costs will pay for the cost of insulation in from 3 to 6 years. In case the insulation warrants a reduction in connected load, the subsequent reduction in installation cost of heating equipment can be applied against insulating costs which would materially reduce total cost.

ELECTRIC HEATING DATA

Electric heater capacity is rated in kilowatts (kw). Electric energy is measured in kilowatt-hours (kwhr). Cost of operation = kw rating × hours used × cost per kwhr.

One boiler horsepower (bhp) = 33,471.9 Btu per hour

One kilowatt-hour (kwhr) = 3,415 Btu.

One boiler horsepower $= \frac{33,471.9}{3,415} = 9.80$ kwhr

One boiler horsepower will evaporate 34.5 lb water per hour *from and at* 212 F

One kilowatt-hour $= \frac{34.5}{9.80} = 3.52$ lb of water per hour at 212 F

Additional conversion factors are given in Chapter 44.

PROBLEMS IN PRACTICE

1 ● Under what conditions are electric heaters most feasible?

a. In climates such as in the South and in California, where they are used economically for many heating purposes.

b. As auxiliary to central steam or hot water heating plants.
1. Fall and spring.
2. During peak loads.
3. During shut-down periods.

c. In factories, offices, etc., where they have a large minimum load charge for electrical power due to large size of connected motors that they cannot use up but must pay for regardless of non-use.

2 ● On what basis should electric heating costs be compared to heating with fuels?

a. Use.

b. Safety.

c. Rates.

d. Locality.

e. Initial investment:
1. Interest.
2. Depreciation.

f. Ease of serving electric heaters

g. Ease of control.

3 ● Approximately how low must the rates be to permit the use of electricity for heating purposes?

Probably the energy must sell for 2 cents or less per kwhr At 2 cents the cost would be $5.86 per 1000 Mbh (See Chapter 29 for comparison with other fuels) This looks high, but the seasonal energy consumption would not be as large with electricity as with other fuels, for reasons stated in Question 1.

4 ● In fan heating systems, what is an important difference between a steam heated coil and an electrically heated coil?

A coil supplied with steam at constant pressure will remain at constant temperature regardless of the amount of air passing over it The temperature of the electric coil supplied with a constant amount of energy will rise if the air quantity is decreased and fall if the air quantity is increased.

Chapter 40

AIR CONDITIONING FOR INDUSTRIAL PROCESSES

Moisture Content and Regain, Typical Industries Requiring Air Conditioning, Classification of Problems, Control of Regain, Control of Rate of Chemical Reaction, Control of Rate of Biochemical Reaction, Control of Rate of Crystallization, Conditioning and Drying, Atmospheric Conditions Required, General Requirements, Air Conditioning of Libraries, Greenhouse Heating

IN the manufacture or processing of hygroscopic materials such as textiles, paper, wood, leather, tobacco and foodstuffs the temperature and relative humidity of the air have a marked influence upon the rate of production and upon the weight, strength, appearance and general quality of the product. This influence is due to the fact that most materials of vegetable or animal origin, and to a lesser extent minerals in certain forms, take moisture from or give it up to the surrounding air.

In industries where the physical properties of the product affect value, the question of moisture is of special importance. With increase in moisture content, hygroscopic materials ordinarily become softer and more pliable. Standards of regain are firmly fixed in trade with fair penalties for excesses. Deficiencies result in loss of revenue to seller and loss of desirable quality to buyer.

Manufacturing economy therefore requires that the moisture content be maintained at a percentage favorable to rapid and satisfactory manipulation and to a minimum loss of material through breakage. A uniform condition is desirable in order that high speed machinery may be adjusted permanently for the desired production with a minimum loss from delays, wastage of raw material and defective product.

In the processing of hygroscopic materials, it is usually necessary to secure a final moisture content suitable for the goods as shipped. Where the goods are sold by weight, it is proper that they contain a normal or standard moisture content.

MOISTURE CONTENT AND REGAIN

The terms *moisture content* and *regain* refer to the amount of moisture in hygroscopic materials. *Moisture content* is the more general term and refers either to free moisture (as in a sponge) or to hygroscopic moisture (which varies with atmospheric conditions). It is usually expressed as a

Table 1. Regain of Hygroscopic Materials

Moisture Content Expressed in Per Cent of Dry Weight of the Substance at Various Relative Humidities—Temperature, 75 F

Classification	Material	Description	Relative Humidity—Per Cent 10	20	30	40	50	60	70	80	90	Authority
Natural Textile Fibres	Cotton	Sea island—roving	2.5	3.7	4.6	5.5	6.6	7.9	9.5	11.5	14.1	Hartshorne
	Cotton	American—cloth	2.6	3.7	4.4	5.2	5.9	6.8	8.1	10.0	14.3	Schloesing
	Cotton	Absorbent	4.8	9.0	12.5	15.7	18.5	20.8	22.8	24.3	25.8	Fuwa
	Wool	Australian merino—skein	4.7	7.0	8.9	10.8	12.8	14.9	17.2	19.9	23.4	Hartshorne
	Silk	Raw chevennes—skein	3.2	5.5	6.9	8.0	8.9	10.2	11.9	14.3	18.8	Schloesing
	Linen	Table cloth	1.9	2.9	3.6	4.3	5.1	6.1	7.0	8.4	10.2	Atkinson
	Linen	Dry spun—yarn	3.6	5.4	6.5	7.3	8.1	8.9	9.8	11.2	13.8	Sommer
	Jute	Average of several grades	3.1	5.2	6.9	8.5	10.2	12.2	14.4	17.1	20.2	Storch
	Hemp	Manila and sisal—rope	2.7	4.7	6.0	7.2	8.5	9.9	11.6	13.6	15.7	Fuwa
Rayons	Viscose Nitrocellulose Cupramonium	Average skein	4.0	5.7	6.8	7.9	9.2	10.8	12.4	14.2	16.0	Robertson
	Cellulose Acetate	Fibre	0.8	1.1	1.4	1.9	2.4	3.0	3.6	4.3	5.3	Robertson
Paper	M. F. Newsprint	Wood pulp—24% ash	2.1	3.2	4.0	4.7	5.3	6.1	7.2	8.7	10.6	U. S. B. of S.
	H. M. F. Writing	Wood pulp—3% ash	3.0	4.2	5.2	6.2	7.2	8.3	9.9	11.9	14.2	U. S. B. of S.
	White Bond	Rag—1% ash	2.4	3.7	4.7	5.5	6.5	7.5	8.8	10.8	13.2	U. S. B. of S.
	Com. Ledger	75% rag—1% ash	3.2	4.2	5.0	5.6	6.2	6.9	8.1	10.3	13.9	U. S. B. of S.
	Kraft Wrapping	Coniferous	3.2	4.6	5.7	6.6	7.6	8.9	10.5	12.6	14.9	U. S. B. of S.
Misc. Organic Materials	Leather	Sole oak—tanned	5.0	8.5	11.2	13.6	16.0	18.3	20.6	24.0	29.2	Phelps
	Catgut	Racquet strings	4.6	7.2	8.6	10.2	12.0	14.3	17.3	19.8	21.7	Fuwa
	Glue	Hide	3.4	4.8	5.8	6.6	7.6	9.0	10.7	11.8	12.5	Fuwa
	Rubber	Solid tire	0.11	0.21	0.32	0.44	0.54	0.66	0.76	0.88	0.99	Fuwa
	Wood	Timber (average)	3.0	4.4	5.9	7.6	9.3	11.3	14.0	17.5	22.0	Forest P. Lab.
	Soap	White	1.9	3.8	5.7	7.6	10.0	12.9	16.1	19.8	23.8	Fuwa
	Tobacco	Cigarette	5.4	8.6	11.0	13.3	16.0	19.5	25.0	33.5	50.0	Ford
Food-stuffs	White Bread		0.5	1.7	3.1	4.5	6.2	8.5	11.1	14.5	19.0	Atkinson
	Crackers		2.1	2.8	3.3	3.9	5.0	6.5	8.3	10.9	14.9	Atkinson
	Macaroni		5.1	7.4	8.8	10.2	11.7	13.[illegible]	16.2	19.0	22.1	Atkinson
	Flour		2.6	4.1	5.3	6.5	8.0	9.9	12.4	15.4	19.1	Bailey
	Starch		2.2	3.8	5.2	6.4	7.4	8.3	9.2	10.6	12.7	Atkinson
	Gelatin		0.7	1.6	2.8	3.8	4.9	6.1	7.6	9.3	11.4	Atkinson
Misc. Inorganic Materials	Asbestos Fibre	Finely divided	0.16	0.24	0.26	0.32	0.41	0.51	0.62	0.73	0.84	Fuwa
	Silica Gel		5.7	9.8	12.7	15.2	17.2	18.8	20.2	21.5	22.6	Fuwa
	Domestic Coke		0.20	0.40	0.61	0.81	1.03	1.24	1.46	1.67	1.89	Selvig
	Activated Charcoal	Steam activated	7.1	14.3	22.8	26.2	28.3	29.2	30.0	31.1	32.7	Fuwa
	Sulphuric Acid	H_2SO_4	33.0	41.0	47.5	52.5	57.0	61.5	67.0	73.5	82.5	Mason

percentage of the total weight of material. *Regain* is more specific and refers only to hygroscopic moisture. It is expressed as a percentage of the *bone-dry* weight of material. For example, if a sample of cloth weighing 100.0 grains is dried to a constant weight of 93.0 grains, the loss in weight, or 7.0 grains, represents the weight of moisture originally contained. This expressed as a percentage of the total weight (100.0 grains) gives the moisture content or 7 per cent. The regain, which is expressed as a percentage of the bone-dry weight, is $\frac{7.0}{93.0}$ or 7.5 per cent.

The use of the term *regain* does not necessarily imply that the material as a whole has been completely dried out and has re-absorbed moisture. In the case of certain textiles, for instance, complete drying during manufacturing is avoided as it might appreciably reduce the ability of the material to re-absorb moisture. In measuring moisture it is necessary to dry out a sample so that the loss in weight may be used as a basis for calculating the regain of the whole lot.

The moisture content of an hygroscopic material at any time depends upon the nature of the material and upon the temperature and especially the relative humidity of the air to which it has been exposed. Not only do different materials acquire different percentages of moisture after prolonged exposure to a given atmosphere, but the rate of absorption or drying out varies with the nature of the material, its thickness and density.

Table 1 shows the regain or hygroscopic moisture content of several organic and inorganic materials when in equilibrium at a dry-bulb temperature of 75 F and various relative humidities. The effect of relative humidity on regain of hygroscopic substances is clearly indicated. The effect of temperature is comparatively unimportant. In the case of cotton, for instance, an increase in temperature of 10 deg has the same effect on regain as a decrease in relative humidity of one per cent. Changes in temperature do, however, affect the rate of absorption or drying. Sudden changes in temperature cause temporary fluctuations in regain even when the relative humidity remains stationary.

INDUSTRIES REQUIRING AIR CONDITIONING

A few of the industries in which air conditioning plays an important part and the major uses in these industries are as follows:

Automobile. Drying of siccative coatings, manufacture of steel, manufacture of artificial leather, drying of rubber, manufacture of tire fabrics, cementing of inner tubes, conditioning of wooden spokes and other wooden parts, conditioning and manufacturing of all electrical windings in connection with the electrical apparatus, storage battery plates and the rubber containers.

Bakery. Flour storage, yeast and ingredient storage, mixers, fermentation rooms, make-up room, proof boxes, load cooling, wrapping (including paraffin paper), cake mixing and cake icing.

Brewery. Fermentation and starting rooms.

Chemical. Powders (including explosives and baking powder), drying of salts of all kinds, hygroscopic compounds and drugs, glues and gelatins.

Clay Products. Bricks, pottery and ceramics.

Confectionery. Chocolates, bon bons, hard candy, gum drops, marshmallows, caramels, chewing gum and starch and various sugars.

Drugs and Pharmaceuticals. Drugs and pharmaceuticals might also be included under chemicals, but definitely to be added to this group are capsules, hygroscopic colloidal crystals, serums and toxins.

Electrical Goods. Toll cable manufacture. telephone exchanges, winding rooms, lamp manufacturer and filament departments

Films and Film Laboratories Drying cabinets, printing rooms, perforating rooms, projection assembly rooms, moving picture studios, celluloid and color photography.

Foods Bread and cake, cereals, macaroni, meats (cold storage markets), yeast, enzymic products, fruits, including apples and bananas, both for preserving and ripening

Furs. Fur storage.

Incubators. Human babies, chickens and similar hatching

Laboratories All kinds

Leather Drying and processing of hides, skins and manufacture of bags, shoes and findings

Linoleum Drying, printing, oil cloth, and linseed oil buildings

Matches. Storage of raw materials, machine drying and packing.

Minerals. Gold beater rooms, gold and silver leaf manufacturing, metal enameling, and mottled ware, particularly all cutting on iron.

Paper and Paper Products Moisture absorption in manufacture, cutting, folding, binding and furnishing bags, including gluing, parchment paper, cellophane containers, paste board containers, paste board bottles and egg containers

Pearls Artificial pearls

Printing, Lithography and Rotogravure. Playing cards, process work, storage, offset work, binding, rollers and ink.

Soap. Crystallizing under the cold process.

Textiles Cotton: drying, spinning and weaving Rayon: chemical house, spinning, drying, twisting, reeling, winding, inspection and storage Silk storage, twisting and reeling, spinning, weaving, knitting, tin and lead weighing and regain rooms (hosiery and underwear)

Tobacco Cigarettes: storage, mixing, blending, paper and machine manufacture. Cigars storage, curing, cleaning, wrapping and packing.

It is apparent that the subject of air conditioning for industrial processes is extensive and greatly involved, and that a detailed treatment is therefore beyond the scope of this book. A few of the salient points of the general subject are covered in this chapter.

CLASSIFICATION OF PROBLEMS

The problems of industrial air conditioning fall into four general classes:

1. Control of Regain
2. Control of Rate of Chemical Reactions.
3. Control of Rate of Biochemical Reactions
4. Control of Rate of Crystallization.

CONTROL OF REGAIN

In the first class the textile plant offers a good example. The regain or moisture content affects the physical properties of textiles to a marked degree, changing the strength, pliability and elasticity.

The fact that the regain of textiles will come into equilibrium with the conditions of the surrounding air and vary with its temperature and relative humidity is the fundamental basis for the control of physical

qualities during manufacture. During the preparation process in a cotton mill, the fibers should be in a condition easily to be brought parallel to each other.

A relative humidity of 50 to 55 per cent gives the best result for these preliminary processes. As the cotton fiber comes to the spinning operation, more flexibility is needed and the relative humidity is increased in this department. For many years, 65 per cent relative humidity was considered the optimum. As pointed out in a paper presented before the *Cotton Manufacturers Association* in 1926, a higher relative humidity is necessary to offset the extra work performed on the fiber as the spindle speed was increased. Today many cotton mills carry 70 per cent relative humidity in the spinning rooms. Winding, warping and weaving are all processes calling for great flexibility and a consequent need for higher humidity.

Other textile fibers, due to their different natural characteristics, are processed under relative humidities and temperatures applicable to each.

Rayons, on account of great loss of strength with the higher regains, should be processed in a relative humidity of 57 per cent. Acetate silk, another chemical fiber, with approximately 50 per cent of the regain of rayon, may be processed between 60 and 65 per cent relative humidity.

All hygroscopic materials, when absorbing moisture, release sensible heat equivalent to the latent heat of the moisture taken up by the material. This may account for a large percentage of the total load.

CONTROL OF RATE OF CHEMICAL REACTION

Typical examples of the second classification, the control of the rate of chemical reactions, occur in the manufacture of rayon. The pulp sheets are conditioned, cut to size, and passed through a mercerizing process. It is essential that this be under close control of both temperature and relative humidity. Temperature controls the rate of reaction directly, while the relative humidity maintains a constant rate of evaporation from the surface of the solution and gives a solution of known strength throughout the mercerizing period.

Another well known example of this class is the *drying* of varnish which is an oxydizing process dependent upon temperature. High relative humidities have a retarding action on the rate of oxydization at the surface and allow the gases to escape as the chemical oxydizers *cure* the varnish film from the bottom. This produces a surface free from bubbles and a film homogeneous throughout.

Temperatures for *drying* varnish vary with the type. A relative humidity of 65 per cent is beneficial. In the field of biochemical control, industrial air conditioning has been applied to so many different and well known products that it is difficult to select an outstanding example.

CONTROL OF RATE OF BIOCHEMICAL REACTIONS

All problems involving fermentation are classed under this heading. As biochemistry is a subdivision of chemistry, subject to the same laws, the rate of reaction may be controlled by temperature. An example of this is the dough room of the modern bakery. Yeast develops best at a

temperature of 80 F with a relative humidity of 65 per cent to hold the surface of the dough open, so that the CO_2 gases formed by the fermentation may pass out and produce a loaf of bread, when baked, of even, fine texture without large voids.

Another example of a similar nature is found in the curing of macaroni. The flour and water mixture is fermented and dried. As it is necessary to have a definite amount of water present to carry on a fermentation process, the moisture must be removed in a relatively short period to stop fermentation and prevent souring and in such a manner as to avoid setting up internal strains in the mixture. Best results are obtained with the correct cycles of both temperature and humidity.

The curing of fruits, such as bananas and lemons, also come under this classification. Bananas are treated somewhat differently and to accomplish the required results, a cycle of temperatures and relative humidities is used. The starches in the pulp must be changed and the skin cured and colored. Then the fruit is cooled to maintain as slow a rate of metabolism as possible. Ideal conditions range between 55 to 57 F, and in no case should the temperature go below 49 F, as the starches then become indigestible.

The curing of lemons is an entirely different problem. Bananas are cured for a quick market, while lemons are held for a future market. The process, therefore, varies in the temperature used. Temperatures from 54 F to 59 F have been found to be best suited for this process. A high relative humidity, 88 to 90 per cent is necessary to hold shrinkage to a minimum and, at the same time, develop the rind so that it will be sufficiently tough to stand handling.

Tobacco from the field to the finished cigar, cigarette, plug or pipe tobacco, offers another interesting example of what may be done by industrial air conditioning in the control of color, texture and flavor. In the processing of tobacco, the first three classifications of industrial air conditioning are involved, and only through close atmospheric control can the best quality of the leaf be developed.

CONTROL OF RATE OF CRYSTALLIZATION

The rate of cooling of a saturated solution determines the size of the crystals formed. Both temperature and relative humidity are of importance, as the one controls the rate of cooling, while the other, through evaporation, changes the density of the solution.

In the coating kettles for pills, gum and nuts a heavy sugar solution is added to the tumbling mass. As the water evaporates, each separate piece is covered with crystals of sugar. A smooth, opaque coating is only accomplished by blowing into the kettle the proper amount of air at the right temperature and relative humidity. If the cooling and drying is too slow, the coating will be rough and semi-translucent, and the appearance unsightly; if too fast, the coating will chip thru to the interior. Only by balancing temperature, relative humidity, and volume of air to the sugar solution can the proper rate be obtained and a perfect coating assured.

The foregoing is presented as typical of a few of the problems met with in industrial air conditioning. They are far from complete but, with the

help of a few of the natural laws, may act as a stimulus to the imagination and assist in solving others

CONDITIONING AND DRYING

In general, the exposure of materials to desirable conditions for treatment may be coincidental with the manufacture or processing of the materials, or they may be treated separately in special enclosures. This latter treatment may be classified as conditioning or drying. The purpose of conditioning or drying is usually to establish a desired condition of moisture content and to regulate the physical properties of the material.

When the final moisture content is lower than the initial one, the term drying is applied. If the final moisture content is to be higher, the process is termed conditioning. In the case of some textile products and tobacco, for example, drying and conditioning may be combined in one process for the dual purpose of removing undesirable moisture and accurately regulating the final moisture content Either conditioning or drying are frequently made continuous processes in which the material is conveyed through an elongated compartment by suitable means.

ATMOSPHERIC CONDITIONS REQUIRED

The most desirable relative humidity during processing depends upon the product and the nature of the process. As far as the behavior of the material itself and its desired final condition are concerned, each material and process represents a different problem. The best relative humidity may range up to 100 per cent. Similarly the most desirable temperature may range between wide limits for different materials and treatments. Extremes in either relative humidity or temperature require relatively expensive equipment for maintaining these conditions and controlling them automatically. Also, in departments where people are working, their health, comfort, and productive efficiency must be considered. A compromise often is desirable.

It is generally considered that relative humidities below 40 per cent are on the dry side, conducive to low regains, a brittle condition of fibrous materials, prevalence of static electricity, and a tendency toward dryness of the skin and membranes of human beings. At the other end of the scale, humidities above 80 per cent are relatively damp, conducive to high regains, extreme softness, and pliability.

Table 2 lists desirable temperatures and humidities for industrial processing. In using this table, care must be taken in qualifying the process. In preparing many materials, conditions are not maintained constantly, but different temperatures and humidities are held for varying lengths of time.

GENERAL REQUIREMENTS

In general, air conditioning apparatus for industrial purposes must be capable of absorbing heat from various sources such as machinery power, electric lights, people, sunlight and chemical reaction; of warming or cooling to any desired degree, and of giving or permitting ample air supply at all times. Refrigeration may or may not be required, depending upon natural conditions, the required relative humidity and the maximum

Table 2 Desirable Temperatures and Humidities for Industrial Processing

Industry	Process	Temperature Degrees Fahrenheit	Relative Humidity Per Cent
Automobile	Assembly line	65	40
Baking	Cake icing	70	50
	Cake mixing	75	65
	Dough fermentation room	80	76 to 80
	Loaf cooling	70	60 to 70
	Make-up room	75 to 80	55 to 70
	Mixing room	75 to 80	55 to 70
	Paraffin paper wrapping	80	55
	Proof boxes	80 to 90	80 to 95
	Storage of flour	70 to 80	60
	Storage of yeast	28 to 40	60 to 75
Biological Products	Vaccines	below 32	
	Antitoxins	38 to 42	
Brewing	Fermentation in vat room	44 to 50	50
	Storage of grains	60	30 to 45
Ceramic	Drying of auger machine brick	180 to 200	
	Drying of refractory shapes	110 to 150	50 to 60
	Molding room	80	60
	Storage of clay	60	35
Chemical	General storage	60 to 80	35 to 50
Confectionery	Chewing gum rolling	75	50
	Chewing gum wrapping	70	45
	Chocolate covering	62 to 65	50 to 55
	Hard candy making	70 to 80	30 to 50
	Packing	65	50
	Starch room	75 to 85	50
	Storage	60 to 68	50 to 65
Distillery	General manufacture	60	45
	Storage of grains	60	30 to 45
Drug	Storage of powders and tablets	70 to 80	30 to 35
Electrical	Insulation winding	104	5
	Manufacture of cotton covered wire	60 to 80	60 to 70
	Manufacture of electrical windings	60 to 80	35 to 50
	Storage of electrical goods	60 to 80	35 to 50
Food	Butter making	60	60
	Dairy chill room	40	60
	Preparation of cereals	60 to 70	38
	Preparation of macaroni	70 to 80	38
	Ripening of meats	40	80
	Slicing of bacon	60	45
	Storage of apples	31 to 34	75 to 85
	Storage of citrus fruit	32	80
	Storage of eggs in shell	30	80
	Storage of meats	0 to 10	50
	Storage of sugar	80	35
Fur	Drying of furs	110	
	Storage of furs	28 to 40	25 to 40

TABLE 2 DESIRABLE TEMPERATURES AND HUMIDITIES FOR INDUSTRIAL PROCESSING
Continued

INDUSTRY	PROCESS	TEMPERATURE DEGREES FAHRENHEIT	RELATIVE HUMIDITY PER CENT
INCUBATORS	Chicken	99 to 102	55 to 75
LABORATORY	General analytical and physical	[illegible] to 7[illegible]	[illegible] to 70
	Storage of materials	[illegible] to 7[illegible]	35 to 50
LEATHER	Drying of hides	90	
LIBRARY	Book storage see discussion in this chapter	65 to 70	[illegible] to 50
LINOLEUM	Printing	80	40
MATCHES	Manufacturing	72 to 74	50
	Storage of matches	6[illegible]	
MUNITIONS	Fuse loading	70	55
PAINT	Air drying lacquers	70 to 90	25 to 50
	Baking lacquers	180 to 300	----- -----
	Air drying of oil paints	60 to 90	25 to 50
PAPER	Binding, cutting, drying, folding gluing	60 to 80	25 to 50
	Storage of paper	60 to 80	35 to 45
PHOTOGRAPHIC	Development of film	70 to 75	60
	Drying	75 to 80	50
	Printing	70	70
	Cutting	72	65
PRINTING	Binding	70	45
	Folding	77	65
	Press room (general)	75	60 to 78
	Press room (lithographic	60 to 75	20 to 60
	Storage of rollers	60 to 80	35 to 45
RUBBER	Manufacturing	90	
	Dipping of surgical rubber articles	75 to 80	25 to 30
	Standard laboratory tests	80 to 84	42 to 48
SOAP	Drying	110	70
TEXTILE	Cotton— carding	75 to 80	50
	combing	75 to 80	60 to 65
	roving	75 to 80	50 to 60
	spinning	60 to 80	60 to 70
	weaving	68 to 75	70 to 80
	Rayon— spinning	70	85
	twisting	70	65
	Silk— dressing	75 to 80	60 to 65
	spinning	75 to 80	65 to 70
	throwing	75 to 80	65 to 70
	weaving	75 to 80	60 to 70
	Wool— carding	75 to 80	65 to 70
	spinning	75 to 80	55 to 60
	weaving	75 to 80	50 to 55
TOBACCO	Cigar and cigarette making	70 to 75	55 to 65
	Softening	90	85
	Stemming or stripping	75 to 85	70

permissible temperature. Washing, purifying and recirculating of the air may be desirable. Good distribution is essential to the control of air motion and for the prevention of undesirable conditions. Accurate, sensitive and reliable automatic control of humidity or temperature, or both, is vital in most cases.

Ordinarily, outside weather conditions and the ventilation required for workers are of secondary importance in relation to the total work to be done by the air conditioning system. In extreme cases of high concentration of industrial heat from machinery and ovens the error of entirely omitting the heat gain through the building structure would not be serious. At the other extreme, where low temperatures must be produced with refrigeration and where comparatively little power is used for driving the machinery, the heat gain through the building structure will become the major factor in determining the size of equipment and in this case the ventilation requirement assumes a normal degree of importance.

Buildings which are to be air conditioned should therefore be designed with careful consideration of over-all cost and efficiency. Condensation resulting from high humidities must be prevented by suitable materials and construction, or else collected and drained to prevent loss of product or quick deterioration of the structure. Air leakage or filtration may add greatly to operating costs or make the maintenance of low humidities (relative or absolute) wholly impossible. Low temperatures require good insulation.

In the general application of industrial air conditioning, the conditions to be maintained are governed almost entirely by the requirements of the product. If any consideration is to be given to the comfort and, therefore, the efficiency of the occupants, it is secondary. In a great many cases, the requirements of the product must necessarily govern, for the physical properties of the material are more important for maximum production than the efficiency of the worker. There are, however, many cases where the worker can and should be given equal consideration and better overall results may be obtained by a proper compromise.

AIR CONDITIONING OF LIBRARIES[1]

Temperature has little effect on the preservation of books. A temperature over 100 F, combined with low relative humidity, may cause the book materials to become brittle, while a temperature much below freezing may cause permanent deterioration of the glue in the binding. The relative humidity should be maintained between 40 and 70 per cent, although these limits need not hold for short periods of time. If the relative humidity gets much below 40 per cent, first the glue and then the paper will tend to become brittle which will not cause any permanent damage unless the book is used while in this condition, as a subsequent increase in humidity will bring the materials back to their normal condition. If the relative humidity gets above 80 per cent, the growth of mildew may be expected.

One of the principal agents of destruction and deterioration of paper

[1]*U. S Bureau of Standards Bulletin No.* 128 entitled A Survey of Storage Conditions in Libraries, by Kimberly and Hicks

and books in libraries is sulphur dioxide gas in the air. If air containing sulphur dioxide is allowed to come in contact with cellulose, the principal constituent of paper, sulphuric acid is formed on the surface. This acid is not volatile at ordinary temperatures and therefore accumulates throughout the life of the paper. The destructive effect of the acid on the paper is independent of the relative humidity of the surrounding air. Low alkaline concentration spray water may be used in an air washer to neutralize the acid condition. Such an air washer must be especially constructed to resist corrosion.

GREENHOUSES

Table 3 lists customary dry-bulb temperature ranges for different types of plants and flowers raised in greenhouses.

TABLE 3. CUSTOMARY TEMPERATURES FOR DIFFERENT TYPES OF GREENHOUSES[a]

TYPE OF HOUSE	TEMPERATURE RANGE DEG F	TYPE OF HOUSE	TEMPERATURE RANGE DEG F
Carnation	55 to 65	Orchid, cool	50 to 60
Conservatory (general collection)	65 to 70	Palm, warm	65 to 75
Cool	50 to 60	Palm, cool	55 to 65
Cucumber	65 to 70	Propagating	55 to 60
Fern, Common	60 to 65	Propagating, Bottom heat	70 to 75
Fern, Tropical	65 to 75	Rose	65 to 70
Forcing	60 to 70	Sweet pea	50 to 60
General purpose	60 to 70	Tomato	65 to 70
Lettuce	55 to 65	Tropical	65 to 75
Orchid, warm	65 to 75	Violet	45 to 55

[a]Lower values may be considered approximate night temperatures and higher values approximate day temperatures

PROBLEMS IN PRACTICE

1 ● Why is air conditioning required for some industrial processes?

To control the physical properties of the materials being processed.

Example. In the manufacture of chewing gum, it is rolled into slabs and scored. The scored slab must then be broken at the score marks to form the sticks. If the slab is too warm, breaking is impossible, if the slab is too cold or too dry, it becomes brittle and shatters, thereby producing much material to be reworked.

2 ● Why is it necessary to control the physical properties of the material being processed?

To permit permanent adjustment of machinery.

Example. In the manufacture of cigarettes, the amount of tobacco fed upon the paper tape is determined by pressure against springs When the tobacco is over-moist and, therefore, over-soft, a great excess will go into the finished cigarette; when the tobacco is too dry and, therefore, harsh, too little goes into the finished cigarette.

3 ● A condition of 75 F dry-bulb temperature and 55 per cent relative humidity is being maintained in a cigarette manufacturing department. What will be the regain and moisture content of the tobacco?

The regain, from Table 1 = 17.75 per cent.

$$\text{The moisture content} = \frac{17.75 \times 100}{100 + 17.75} = 15.1 \text{ per cent.}$$

4 ● A 1-lb sample taken from a 100-lb batch of material is found to have a bone dry weight of 0.89 lb. This material is to be processed under atmospheric conditions which should produce a regain of 15 per cent. Compute the finished weight for each original 100-lb batch.

Let W equal the number of pounds of moisture in a finished batch

$$\frac{W}{89} = \text{regain} = 15 \text{ per cent} = \frac{15}{100}$$

$$W = 13.35$$

$$89 + 13.35 = 102\ 35 \text{ lb finished weight}$$

5 ● A bundle of sea island cotton is found to have a bone dry weight of 9.26 lb. What is the proper relative humidity at 75 F to produce a weight of 10 lb at equilibrium?

Desired conditioned weight	=	10 00 lb
Bone dry weight	=	9 26 lb
Weight of moisture required	=	0 74 lb

$$\text{Regain} = \frac{0.74}{9\ 26} \times 100 = 7\ 9 \text{ per cent}$$

From Table 1, the proper relative humidity required is 60 per cent.

6 ● Compute the bone dry weight of 1000 lb of manila rope which has been stored for a considerable period of time in a conditioned room at 75 F dry-bulb temperature and 50 per cent relative humidity.

Assuming that this material has come to equilibrium under the atmospheric conditions given, Table 1 shows a regain of 8 5 per cent

Let W equal the total weight of moisture in pounds.

$1000 - W$ = bone dry weight in pounds

$$\frac{W}{1000 - W} = \text{regain} = 8.5 \text{ per cent} = \frac{8\ 5}{100}$$

$$W = 78.3 \text{ lb moisture}$$

$$1000 - 78\ 3 = 921\ 7 \text{ lb bone dry weight}$$

7 ● An egg evaporating plant wishes to dry 2000 lb of egg whites (85 per cent water) to crystalline form each 24 hours. The maximum permissible air delivery temperature in the dryer is 140 F. What air volume will be required, assuming that outside air is at 95 F dry-bulb and 78 F wet-bulb and that air leaves the dryer 70 per cent saturated?

Moisture to be removed = 2000 × 0 85 = 1700 lb. Using psychrometric chart and starting at the intersection of the vertical 95 F dry-bulb temperature line and the 46 per cent humidity line, move horizontally to the right to the intersection with the 140 F vertical temperature line at 13 per cent relative humidity; then move along the constant heat (or wet-bulb line) to its intersection with the 70 per cent relative humidity curve and read 97 5 F dry-bulb, which will be the temperature of the air leaving the dryer.

Moisture per cubic foot at 97 5 F and 70 per cent relative humidity	=	13 2 grains
Moisture per cubic foot at 95 F and 78 F wet-bulb	=	8 3 grains
Moisture added per cubic foot of air handled	=	4 9 grains

$$\frac{1700 \times 7000}{24 \times 60 \times 4.9} = 1685 \text{ cfm}$$

No allowance is made for heat lost in the transmission to and from the dryer or for the heat required to raise the product from its entering temperature to that maintained in the dryer. This would necessitate a trial and error solution common to all drying problems

Chapter 41

DRYING SYSTEMS

Drying Methods, Driers, Mechanism of Drying, Moisture, General Rules for Drying, Equipment, Humidity Chart, Combustion, Design, Estimating Methods

DRYING, in its broader sense, refers to the removal of water, or other volatile liquid from either a gaseous, liquid, or solid material. In practice, the process of direct drying gaseous material is referred to generally as dehumidifying, or condensing, and in some cases chemicals are used in the adsorption or absorption of moisture. Drying a liquid is called evaporation or distillation. The common usage of the word *drying* refers to the removal of water or other liquid, such as a solvent, by evaporation from a solid material.

When the solid to be dried contains large amounts of free water, the actual drying process is frequently preceded by the removal of part of the water by some mechanical means, such as filtration, settling, pressing or centrifuging. Removal of as much water as possible by such methods is usually advisable, as the cost of these operations, per pound of water removed, is generally much less than by evaporation.

DRYING METHODS

Drying may be accomplished in any one or combination of the following methods:

1. Radiation
2. Conduction, or direct contact
3. Convection

Radiation

The source of heat for radiation may be either the sun, or heated surfaces. Sun drying is practiced where danger from rain is slight, and where sufficient time can be allowed. Where a strict adherence to a schedule is necessary, or where dusty atmosphere is present, this method is not in favor. Fruits are often dried in the sun.

Radiation from hot surfaces (heated by steam, electricity, or other means) furnishes generally, from one-third to one-half the total heat required for evaporation. Convection currents set up by these hot surfaces and the cooler materials carry the balance of the heat.

TABLE 1. DRIERS FOR EVAPORATION OF WATER

Type	Kind	Materials Handled	Means of Handling	Temp Range Deg F	Heat Supply	Uses and Remarks
Batch or Intermittent	Compartment	Paper, Leather, Yarns, Lumber, Foodstuffs	Suspended, Truck, Tray	80 to 180	Steam Coils, Air, Electricity	When production does not warrant continuous drier
	Agitated	Chemicals too sticky for Rotary Drier	Shoveled into Drum or Pan	100 to 330	Water, Steam Jacketed, may have Vacuum on top	Where dust must be saved
	Vacuum	Chemicals, Explosives, Pharmaceuticals, Food Products	Tray, Basket, Tumbling Drum	80 to 300	Water, Steam	Cost of operation high, for expensive materials
Continuous	Tunnel	Ceramics, Chemicals, Lumber, Food Products	Truck, Tray, Belt	100 to 350	Steam Coils, Air, Electricity, Products of Combustion	For High Production
	Rotary	Bulk	Cascades through	80 to 500	Air, Steam, Products of Combustion	Where material will stand rough handling and is not subject to balling up
	Drum	Liquids, Slurries	Flowed on Drum, Dry Material Scraped off	to 310	Steam, may have Vacuum on Top	Hygroscopic materials dried with vacuum, and packed immediately
	Cylinder	Paper, Textiles, Chemicals	Continuous Sheets, Endless Chain Belt	to 350	Steam inside of Drum	Where material comes in sheets or rolls, and will stand direct contact with heating surface
	Festoon	Paper, Chemicals	Continuous Sheets, Suspended on Metal Screens	to 200	Air, Steam Coils	Where one side cannot come in contact with supports until dry
	Tower or Column	Grains, Sand	Falls through by Gravity	125 to 250	Air, Steam Coils	Where headroom is available
	Spray	Solutions over 30% Solids	Sprayed into Chamber	120 to 350	Air, Products of Combustion	Drying is almost instantaneous
	Induction	Metals, for removal of traces of Water	Placed in High Frequency Field	to 400	Electricity	Where heating of metal from inside out is important

Conduction or Direct Contact Drying

This method of drying is advantageous where the material can be flowed on to the drying surface and the dried material scraped off, or where the material to be dried can be handled in a sheet, and where there is no danger of subjecting the product to the full temperature of the heating medium. The source of heat for this method may be steam, electricity, hot oil or hot water.

Convection

The circulation of heated air or other gases about the material to be dried is generally termed convection drying. The convection may be either natural or forced. With forced circulation, the temperature of the dryer is more uniform and the rate of drying is much higher than with natural circulation. Where humidity is used, the control is much easier, and more accurate.

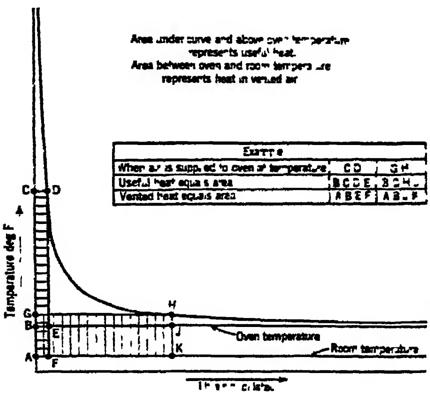


FIG. 1. RELATION BETWEEN USEFUL AND TOTAL HEAT SUPPLIED

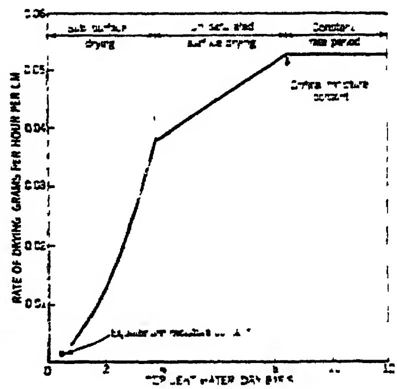


FIG. 2. RATE OF DRYING OF WHITING SLAB

The source of heat for a convection dryer may be steam, electricity, hot water, oil fired heater, gas fired heater, or coal furnace. Where either oil, gas or coal is used, the type of heater may be direct or indirect; *i.e.*, the products of combustion may be used (direct), or the circulated air may be heated through an interchanger (indirect).

Where the direct type is used, there is naturally a higher thermal efficiency, but it can only be used where the odor, soot, or the chemical elements of the products of combustion do not affect the material being dried. When heat economy is an important consideration this method (Fig. 1) may be used, permitting a small amount of air to be circulated, if a sacrifice of accurate control of temperature and humidity can be justified.

DRIERS

The term *adiabatic drier* is applied to a drier in which all the heat is supplied by air externally heated. The temperature of the air in the

drier decreases as a transfer of heat to the material being dried takes place. Where part or all of the heat is supplied by steam coils or other means, within the drier itself, the drier is known as a *constant temperature drier.* Driers using little air for heating medium with a high temperature drop, are difficult to hold at uniform temperatures; the more air used, the easier it is to secure accurate control of temperature and humidity. Driers may be classified as shown in Table 1.

MECHANISM OF DRYING

The modern theory of drying may be summed up as follows: Assuming uniform velocity and distribution of air at a constant temperature and humidity over the surface to be dried, the drying cycle will be divided into two distinct stages:

1. Constant rate period
2. Falling rate period.

The *constant rate period* occurs while the material being dried is still very wet, and continues as long as the water in the material comes to the surface so rapidly that the surface remains thoroughly wet, and evaporation proceeds at a constant rate, precisely as from a free water surface. The material assumes a temperature corresponding to the wet-bulb temperature of the surrounding air, or slightly higher, due to radiation and conduction from dry surfaces adjoining the material. The constant rate period continues until a time when the moisture no longer comes to the surface as fast as it is evaporated. This point is called the *critical moisture content.*

As the drying proceeds, a period of *uniform falling rate* is entered. During this period, the surface of the material is gradually drying out, and the rate of drying falls as the remaining wet surface decreases in area. This period is also known as unsaturated surface drying.

As drying continues, the surface is completely dry and the water from the interior evaporates and comes through the surface as vapor. As the plane of water recedes, the diffusion of the vapor becomes more difficult and hence the period is known as *varying falling rate period*, or subsurface drying.

As drying progresses another point called *equilibrium moisture content* is reached, where the vapor pressure of the moisture in the air and the vapor pressure of the moisture in the material are equal, and drying ceases. The drying of a slab of whiting is shown in Fig. 2 and illustrates the principles pointed out above. The factors affecting the variations of drying rates during the above periods are pointed out in Table 2.

Omissions in the Cycle

Many solids, such as lumber, are so dry at the beginning of the drying operation that the constant rate period of free surface evaporation does not occur. Frequently the surface of the material is dry enough so that no surface drying can take place, in which case only the final stage of subsurface drying is involved. In other instances, the critical moisture content of a wet solid is sufficiently low that sub-surface drying starts almost immediately after the conclusion of the constant rate period. Thus the

intermediate state of unsaturated surface drying does not occur and the drying is of the sub-surface type during practically the whole of the falling rate period. With other kinds of material, particularly thin sheets, such as newsprint paper, sub-surface drying may occur at such a low moisture content that it is not encountered in commercial work, the

TABLE 2 FACTORS INFLUENCING DRYING

FACTOR	DRYING PERIOD	
	Constant Rate Unsaturated Surface	Sub-surface
Temperature	Increase in temperature increases drying rate	Increase in temperature increases drying rate because with decreased viscosity diffusion increases
Humidity	Drying rate increases as humidity is decreased	No effect until equilibrium content is reached; drying then ceases
Air Velocity	Drying rate varies approximately as the 0 6 power of the velocity	No effect
Air direction	Drying rate increases the more nearly the air blows perpendicular to surface, for dead air film becomes thinner	No effect
Thickness of Material	Drying rate is not affected by the thickness	Drying rate varies inversely as the square of the thickness

falling rate period being confined solely in practice, to unsaturated surface drying.

MOISTURE

Moisture in the solid may be in either of two forms:

1. Capillary or free
2. Hygroscopic or chemically combined

Free moisture is contained in the capillary spaces between the particles or fibers of the materials. The loss of this moisture changes only the weight of the material. *Chemically combined* or hygroscopic moisture is intimately associated with the physical nature of the material and its removal changes both the physical characteristics as well as the chemical properties. The amount of hygroscopic moisture a material can contain is limited. This limit is called the *fiber saturation point*. When material is dried below this point, care must be exercised to avoid physical changes in the material, such as shrinkage, hardening, etc. All hygroscopic materials have definite equilibrium moisture contents dependent on temperature and humidity. Materials are frequently dried to a lower moisture content than those of equilibrium conditions in use, and allowed to regain the necessary moisture after leaving the drier to equalize the moisture in the material. Fig. 3[1] shows the equilibrium moisture content of wood.

[1]U S Department of Agriculture Bulletin No. 1136

GENERAL RULES FOR DRYING

Temperature

The highest temperature possible should be used because of faster drying and smaller requirements for ventilation. The amount of moisture that can be carried by a pound of air increases rapidly with rise in temperature as shown in the humidity chart of Fig. 4. Too high a temperature may cause spoilage of materials; many materials calcine or change their chemical properties if heated too hot; gypsum and glauber salts lose some of the chemically combined water, fall apart, and change their chemical properties. Too high or rapid rise in temperatures in drying lumber or ceramics may create a liquid vapor tension within the material so high that the cells explode, causing permanent injury to the fiber. If too high a temperature is used on some chemicals, they begin to react

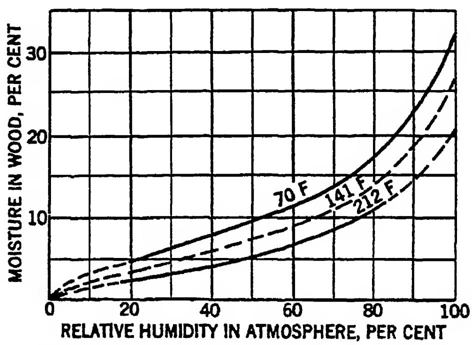


FIG. 3. RELATION OF EQUILIBRIUM MOISTURE CONTENT IN WOOD TO THE RELATIVE HUMIDITY OF SURROUNDING AIR

exothermally; a temperature rise and chemical action from within will burn the materials, *e.g.*, bakelite products, gunpowder, etc. During the constant rate period of drying, the material heats only to the wet-bulb temperature of the surrounding air, consequently high temperatures will not injure the material in this stage.

Humidity

Moisture in the drying air may be very important. Many materials tend to case-harden, dry on the outside, forming a skin which retards the moisture flow from the inside to the surface, or stops it completely, and so increases the drying time very much or causes a change of the physical properties of the material. It is often necessary to add humidity to the air in the initial stage of drying. Lumber case-hardens, cracks, and warps if the outside is dried too fast. Ceramics crack if not heated through before drying commences. Elastic materials warp while others crack if not evenly dried. Many paints case-harden if not dried under high humidity.

On the other hand, in the case of those materials whose physical or chemical properties require that they be dried at relatively low temperatures high humidity tends to retard drying in the first stage and may even stop it altogether in the final stage. Where drying temperatures

below 120 to 140 F are used the drying rate may be highly dependent on atmospheric humidity conditions. In such instances it is often desirable to dehumidify the air entering the drier during periods of high atmospheric humidity; where a high degree of uniformity is required it is often possible to secure complete independence of atmospheric conditions by recirculating the air in a closed system which includes a suitable dehumidifier. For this purpose absorptive dehumidifying systems have the advantage of accomplishing the desired reduction of humidity without appreciably elevating or lowering the dry-bulb temperature of the air; for this reason after-cooling is not required, and reheating is reduced to a minimum. Complete descriptions of such dehumidifying systems are given in Chapter 11 on Cooling Methods.

Air Circulation

As noted under Mechanism of Drying, air velocity is more important in the first two stages of drying than in the last, and for this reason zone drying in continuous driers is frequently considered. It permits accurate regulation of temperature, humidity, and velocity in the different zones. High velocity results in more rapid drying, more even distribution of temperature and consequently more even drying in the first period. Too high a velocity may be detrimental because of excessive power needed for creating it, or because the material may blow away if it is light and fluffy. In the drying of paints, varnishes, and enamels, high velocity or improper distribution of the air even with the use of filters, may cause dust already in the drier, to be blown against the material, ruining the finish. Table 3 presents data on drying of various materials.

EQUIPMENT FOR DRYING

Equipment for drying may be divided into the following classes:

1. Heat and humidity supply
2. Methods of handling.
3. Ovens.

The heat and humidity supply for low temperature work up to 250 F is often steam; steam coils either in the oven or outside, heat the air used for drying. Circulation of heated oil is used to a limited extent, but the danger of leaks is serious, for if the oil is hotter than the flash point, a fire may start if the oil is released to the atmosphere. In many cases where steam is not available, direct or indirect fired heaters are used with gas or oil as fuel. Indirect heaters should be carefully selected from a standpoint of long life and efficiency. The heat exchange surface should be adequate in area and easily accessible for cleaning and removal. For extremely high temperatures, alloy surface may be used. With direct fired equipment care must be used in the selection of burners and sufficient combustion space allowed to insure complete combustion of fuel. Humidity can be obtained in driers by the use of steam spray, humidifiers, or recirculation.

Methods of *handling of material* have been indicated in Table 1.

For low temperature work up to 200 F *ovens* and driers are commonly built of two thicknesses of insulating board (fireproof preferred), with air space between. As the temperature increases materials better able to

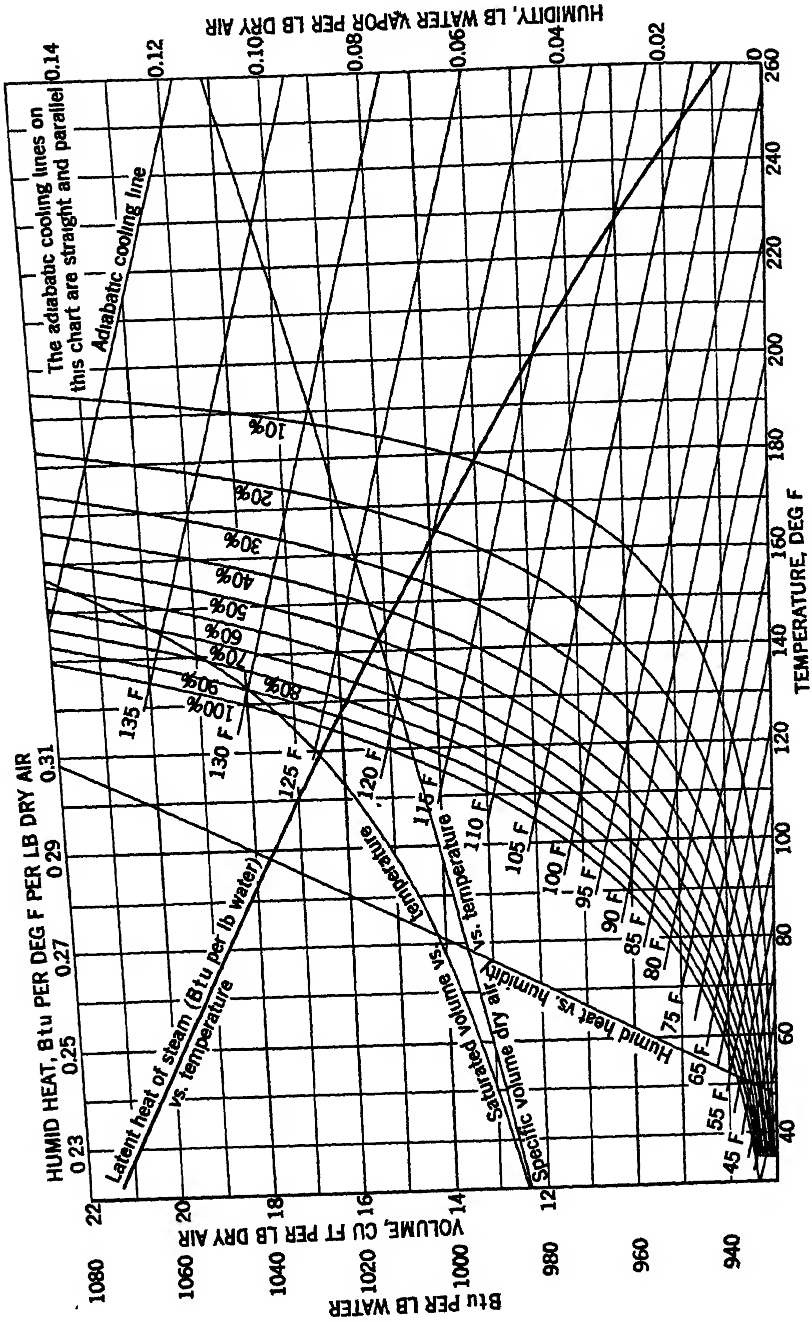


Fig. 4. Humidity Chart

withstand the heat must be used. Metal lined ovens are easy to keep clean, and many high temperature driers up to 1000 F are made of metal panels with insulation between. Care should be taken to avoid *through metal* (metal extending through the oven from inside to out). Batch type ovens are entirely closed while in use and control of air leakage is easily taken care of. In the continuous drier where the ends are open, heat and air leakage become important. Warm air leaking out of the ends of ovens means a heat loss, and often the temperature and humidity outside the oven becomes unbearable. For this reason, inclined or bottom entry ovens are used, as the warm air leakage can be more easily controlled. See Figs. 5 and 6.

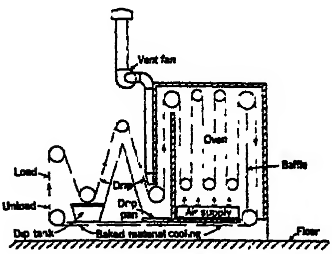


FIG. 5. SMALL PART MULTIPLE PASS OVEN

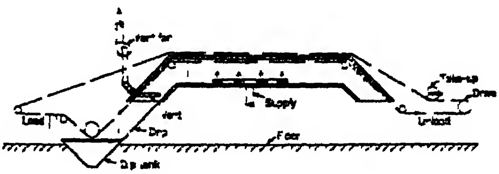


FIG. 6. INCLINED END ENAMELING OVEN

HUMIDITY CHART FOR DRYING WORK

In drying problems the chemical engineer uses different psychrometric values than those used by the heating, ventilating and air conditioning engineer. The humidity chart illustrated in Fig. 4 is based upon values determined from the following explanations:

Humidity (*H*) is the number of pounds of water vapor carried by one pound of dry air.

Percentage Humidity (*%H*) is the number of pounds of water vapor carried by one pound of dry air at a definite temperature, divided by the number of pounds of vapor that one pound of dry air would carry if it were completely saturated at the same temperature.

Percent Relative Humidity (Φ) is the ratio of weight of water vapor contained in any given volume of air, to the weight of water vapor present in the same volume of saturated air, all values referring to the same temperature.

To convert from one relation to the other,

$$\% H = \frac{29.92 - p_s}{29.92 - p} \times \Phi \qquad (1)$$

where

p_s = vapor pressure of water, inches mercury; at dry-bulb temperature, degrees Fahrenheit.

p = $\Phi\, p_s$.

COMBUSTION

Where products of combustion are used directly in the oven, a knowledge of their formation and heat values is important. The properties of

Table 3. Drying Time and Conditions for Representative Materials[a]

Material	Temperature Deg F	Per Cent Relative Humidity	Drying Time
Apples	140–180		6 Hrs
Armatures Varnish	200		2.5 Hrs
Banana Food ¼ in Thick	140		4–6 Hrs
Barrels	300		15 Min
Beans	140		18 Hrs
Bedding	150–190		
Blankets	120		40 Min
Brake Lining	325		12 Min
Brick continuous	350 to 90		24 Hrs
Briquets	1100		108 Min
Cabbage Raw	150		4 5 Hrs
Candied Peel	165		2 Hrs
Casein	180		5 Hrs
Cereals	110–150		
Ceramics before firing	150	70 to 20	24 Hrs
Chicle	95–100		
Coco-fiber mats	170–210		10 Hrs
Cocoanut	150–200		4–6 Hrs
Coffee	160–180		24 Hrs
Conduit (Enamel)	400 Max		2 Hrs
Cores, Oil sand for molding ½—1 in thick	300		30 Min
Black sand with goulic binder about 0.6 of time 3 in thick	480		2.5 Hrs
8 in thick	480		4 5 Hrs
16 in thick	700		10 Hrs
Cores, Crank case (in continuous ovens)	525–600		2–3 Hrs
Cores, Radiator (in continuous ovens)	275–450		1 5 Hrs
Cornstalk Board	150		2 Hrs
Cotton Linters	180		
Enamels synthetic			
Finish coat on autos	225		2 Hrs + Air Dry
Ice boxes all metal (white)	290–425		1 Hr
Ice boxes wood inside (white)	225		3 Hrs
Enamel not synthetic			
Fence posts green	200		1 Hr
Golf balls (white)	90–95	40–50	18–36 Hrs
Small parts (auto) black	450		1 Hr
Steel furniture	225–300		30–350 Min
License plates	250		1 5 Hrs
Feathers	150–180		
Films, Photographic	85–110		20–30 Min
Fruits and Vegetables	140		2–6
Furs	110		
Gelatin	110		
Glue bone, thin sheets on wire trays	70–90		6–9 Days
Glue skin	70–90		2 Days
Glue size on furniture	130		4 Hrs
Gut	150		
Gypsum board ⅜ in thick Start Wet	350		60 Min
Gypsum board ⅜ in thick Finish	275		
Gypsum block	350 to 190		8–16 Hrs
Hair felt	180–200		
Hair goods	150–190		1 Hr
Hanks on poles	120		2 Hrs
Hats felt	140–180		
Hides thin leather	90		2–4 Hrs
Hides heavy	70–90		4–6 Days

[a]See references at end of chapter.

Table 3 Drying Time and Conditions for Representative Materials—Con.

Material	[illegible]	[illegible]	[illegible]
Hops	[illegible]		
Ink printing	[illegible]		
Japan beds	[illegible]		1 5–2 Hrs
Japan cash register	[illegible]–450		1 5 Hrs
Japan metal shelving	[illegible]		30 Min
Knitted fabrics	140–[illegible]		
Leather mulling	75–[illegible]	85	
Leather thick sole	[illegible]	70	4 Days
Leather uppers	80		2–3 Days
Linoleum varnish	1[illegible]0–145	10–30	6–10 Hrs
Lithographing on tin color work	250–270		18–25 Min
Lithographing on tin Japan	350		
Lumber green hardwood	100–180		3–1[illegible]0 Days
Lumber green soft wood	160–220		2–14 Days
Macaroni	90–110		7 5–8 Hrs
Matches	140–180		
Matrix	350		15 Min
Milk and other liquid foods spray dried	135–300		Instantaneous
Millboard sheets	95		10 Hrs
Moulds green sand C I flasks (one 8 in thick	600		6 Hrs
surface only exposed) 13 in thick	700		13 Hrs
Motors, field coils	180		6 Hrs
Motors, stators	250		6.5 Hrs
Noodles	90–95		
Nuts	75–140		24 Hrs
Oil cloth	150		
Paint, wood wheels	150	35	8–24 Hrs
Paint, on sheet metal	350–140	22–30	2.5 Hrs
Paper, machine dried	180		
Paper, air dried	90–200		
Paper wall, ground coat	140		3 Min
Paper wall, varnished	140–160	45	15 Min
Paper cardboard, spirit varnish	150		1–2 Min
Peaches	135		26 Hrs
Pears	140		24 Hrs
Peas	150		6 Hrs
Potatoes sliced	85		4 Hrs
Potatoes steamed	170		6.5 Hrs
Prunes	140		
Rags	180		
Ramie fiber	140		10 Hrs
Rice	150		
Rock wool insulation	300		8 Hrs
Rubber	85–90		6–12 Hrs
Rubber reclaimed	140–200		1–2 Hrs
Rugs	190		4–8 Hrs
Salt	350		Rotary Drier
Sand loose 1 in deep	300		10–15 Min
Sausage casings	110		5 Hrs
Shade cloth	240		1–2 Hrs
Shirts	120		20 Min
Soap	100–125		12–72 Hrs
Starch	180–200		1–4 Hrs
Stock feed mixed	180–220		20–30 Min
Storage battery plates	100–110 250	90 for Low for	24 Hrs 6 Hrs
Sugar	150–200		20–30 Min

[a]See references at end of chapter.

Table 3 Drying Time and Conditions for Representative Materials[a]—Con.

Material	Temperature Deg F	Per Cent Relative Humidity	Drying Time
Tanin and other chemicals spray dried	250–300		Instantaneous
Terra Cotta (air drying in conditioned room)	150–200		12–96 Hrs
Tobacco leaves	85–130		12 Hrs
Tobacco stems	180–200		12 Hrs
Varnish refrigerator boxes	110	35	5–7 Hrs
Varnish steering wheels	110–140	25–35	Overnight
Veneer 1/4 in 3-ply	120–130	35–40	6–8 Hrs + 2 Hrs acclimation
13/16 in. 5-ply	120–130	35–40	16–18 Hrs + 4 Hrs acclimation
1 1/4 in. 5-ply	120–130	35–40	20–24 Hrs + 4 Hrs acclimation
Vitreous Enamel sheets before firing	170		
Wallboard pasted plywood	300		15–20 Min
Wallboard fiber insulating, roller type drier	300–385		2½–3 Hrs
Wallboard fiber insulating, truck type drier	300–385		24–48 Hrs
Walnuts	100		24 Hrs
Wheat, corn, oats, rice, barley	180		
Wire cloth Japan	200		20 Min
Wool	105		

[a]See references at end of chapter.

the common constituents of fuel are shown in Table 4. The heating values of oils are shown in Fig. 7. The sensible heat in Btu contained in the products of combustion of an average fuel oil and various gases is given in Fig. 8. The problem of securing complete combustion in a heater is important, in order to secure efficiency and the absence of soot formation, but unlike the ordinary power or heating boiler, excess air need not be maintained at a minimum in most cases. Excess air is generally admitted either in the heater or before the products go into the drier.

DESIGN

In all drying problems, data regarding temperatures, time, and humidity must be obtained by experiment or previous experience. Experiments are best performed at the temperatures, humidities, and velocities to be actually used in the full sized drier, and with full size samples.

The following nomenclature and explanation of terms will be used in the discussion of drying calculations:

H = humidity of air, pounds of water vapor per pound of dry air.
G = pounds of dry air supplied to the drier per unit of time.
S = pounds of stock dried per unit of time in a continuous drier.
S^1 = pounds of stock charged per batch to a discontinuous drier
Θ = time.
Q = total heat supplied to the drier.

t = air temperature.
t' = stock temperature
t'' = average stock temperature over short time interval in a batch drier
t_w = wet-bulb temperature
s' = specific heat of the stock
B = total radiation and conduction losses per unit time.
w = pounds of water per pound of dry stock
r = heat of evaporation of water
s = humid heat of air, i.e., heat necessary to raise 1 lb of dry air + H lb of steam 1 F.

Subscript (1) designates conditions at the point where the material in question (air or stock) enters and (2) where it leaves the drier.

Air driers may be divided into two classes, those in which *all moisture* evaporated from the stock *leaves the drier as vapor* in the effluent air, and those in which *part or all* of the moisture *is condensed* from the air *in the drying equipment itself*. In any continuously operating drier of the first type the relation between moisture content of the stock and quantity of air required for the drying operation is given by the equation:

$$G(H_2 - H_1) = S'(w_1 - w_2) \qquad (2)$$

TABLE 4. GAS COMBUSTION CONSTANTS[a]

Gas	Chemical Formula	Molecular Weight	Cu Ft per Lb	Heat of Combustion, Btu per Lb, Gross	Heat of Combustion, Btu per Lb, Net	Lbs per Lb of Combustible, Required for Combustion, O_2	Required for Combustion, N_2	Required for Combustion, Air	Flue Products, CO_2	Flue Products, H_2O	Flue Products, N_2
Carbon	C	12 000	—	14,140	14,140	2 667	8 873	11 540	3 667	—	8 873
Hydrogen	H_2	2 015	187 723	61,100	51,643	7 939	26 414	34 353	—	8 939	26 414
Oxygen	O_2	32 000	11 819	—	—	—	—	—	—	—	—
Nitrogen	N_2	28 016	13 443	—	—	—	—	—	—	—	—
Carbon Monoxide	CO	28 000	13 506	4,369	4,369	0 571	1 900	2 471	1 571	—	1 900
Carbon Dioxide	CO_2	44 000	8 548		—	—	—	—	—	—	—
Methane	CH_4	16 031	23 565	23,912	21 533	3 992	13 282	17 274	2 745	2 248	13 282
Ethane	C_2H_6	30 016	12 455	22,215	20,312	3 728	12 404	16 132	2 929	1 799	12 404
Propane	C_3H_8	44 062	8 365	21,564	19,834	3 631	12 081	15 712	2 996	1 635	12 081
Sulphur Dioxide	SO_2	64 060	5.770	—	—	—	—	—	—	—	—
Water Vapor	H_2O	18 015	21 017	—	—	—	—	—	—	—	—
Air	—	28 900	13 063	—	—	—	—	—	—	—	—

[a]All gas volumes corrected to 60 F and 30 in. mercury barometric pressure dry.

In discontinuous driers, *e.g.*, compartment driers, the drying operation is given by the equation:

$$G(H_2 - H_1) = S^1 \frac{dx}{d\Theta} \tag{2a}$$

In the continuous drier, the heat consumption per unit time is:

$$\frac{Q}{\Theta} = Gs_1(t_2 - t_1) + G(r_2 + t_2 - t'_2)(H_2 - H_1) + S(t'_2 - t'_1)(s^1 + w_1) + B \tag{3}$$

Equation (3) assumes continuity of operation. For charge or batch operations, the total time of the drying cycle may be broken up into a number of periods, sufficiently short so that over each period average

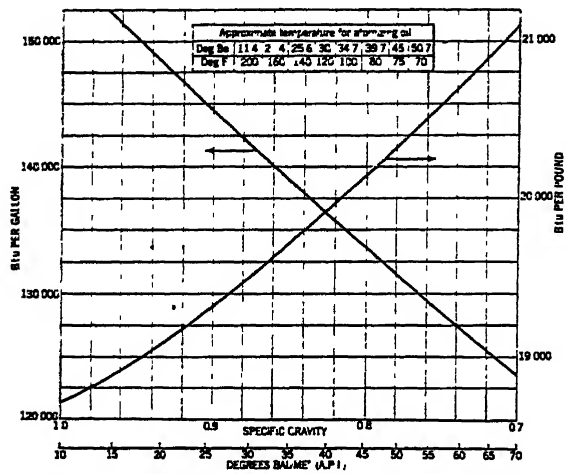


Fig. 7. Heating Values of Fuel Oil, Btu Gross

values of t, t^1 and H may be employed provided the third term of the right hand member of the equation is modified to read:

$$S^1 (t^{11}_1 - t^{11}_1)(s^1 - w_1)$$

and in the second term t^1_2 be replaced by

$$\frac{t^1_1 + t^{11}_2}{2}$$

Theoretically these periods should be very short and the equation integrated. Practically the error introduced by using a small number of long periods and employing average values of the variables over each, rarely introduces serious error. The evaluation of equation (2a) may be approximated in a similar manner.

The first term of the right hand member of equation (3) represents heat lost as sensible heat in the effluent air. In many drying operations this becomes excessive. Each pound of air supplied should remove the maximum amount of moisture. This is best accomplished by bringing the air

into contact with the stock with sufficient intimacy so that the air leaving the drier is saturated, or nearly so. Counter-current as against parallel flow of air and stock gives rise to optimum operating conditions, resulting in a minimum quantity of air required (G), and a corresponding minimum loss, as sensible heat, in the exit air. Similarly, continuous operation is superior to intermittent operation.

Despite the fact that the sensible heat loss increases with the rise in temperature of the air, the percentage of heat lost from this source decreases, provided the increase in moisture carrying capacity of the air,

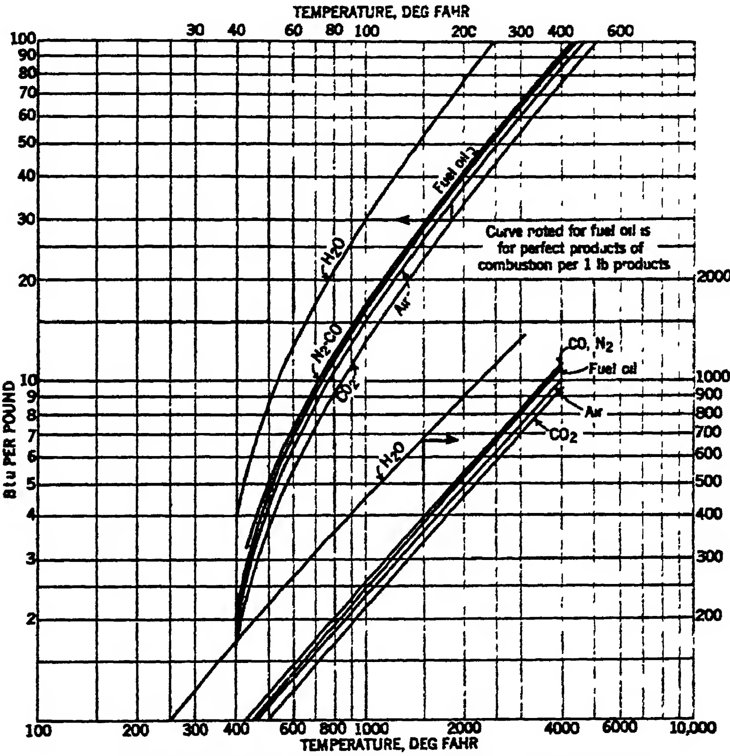


FIG 8. HEAT CONTENT OF GASES ABOVE 32 F IN BTU PER POUND

due to high temperature, is actually utilized. To secure maximum thermal efficiency in drying, a high drying temperature and high saturation of the outlet air is imperative.

Ventilation Phase

The technique of attack of the *ventilation phase* of a drying problem is best made clear by an illustration. Assume that a material containing 40 per cent moisture is to be dried until this quantity of moisture is reduced to 5 per cent by weight. The material will stand an air temperature of 150 F and it is possible to provide sufficiently good contact between the material and the drying air so that the effluent air can be

brought up to 50 per cent humidity at 150 F. The drier is to use room air, the temperature and humidity of which may be assumed to average 70 F and 50 per cent. A counter-current drier will be employed and the air in this drier will be kept at a substantially constant temperature of 150 F by heaters thermostatically controlled. The stock enters at 70 F, rises quickly to the wet-bulb temperature of the air, with which it is in

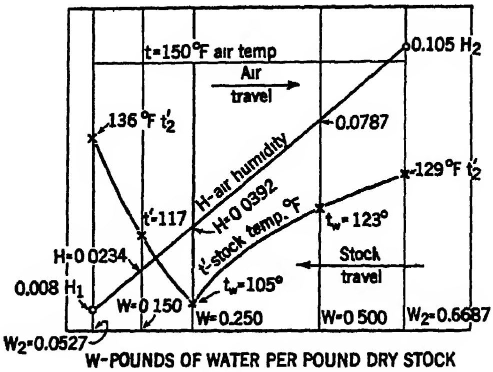


Fig. 9. Temperature-Humidity Relations in a Drier

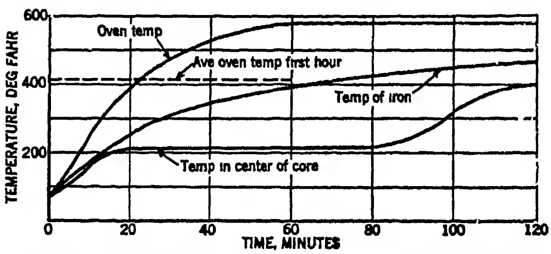


Fig. 10. Core Drying Time Temperature Relations

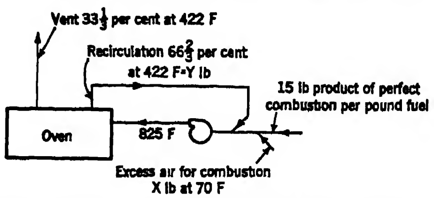


Fig. 11. Core Drying Diagram of Combustion Products and Air

contact, and is found experimentally to maintain wet-bulb temperature until the moisture content has fallen to 20 per cent. From this point its temperature rises progressively as it dries. In this range the difference in temperature between stock and air, divided by the wet-bulb depression, may be assumed proportional to the moisture content.

The moisture content of the entering stock, in the units here employed, is:

$$w_1 = \frac{40 \text{ per cent water}}{60 \text{ per cent dry stock}} = 0.6667. \quad w_2 = \frac{5 \text{ per cent water}}{95 \text{ per cent dry stock}} = 0.0527$$

$w_1 - w_2 = \Delta w = 0.614$ lb water evaporated per pound of dry stock. Since the air leaving the drier is 50 per cent saturated at 150 F from Fig. 4, $H_2 = 0\ 105$ Similarly, $H_1 = 0\ 008$, corresponding to 50 per cent humidity at 70 F. Consequently $H_2 - H_1 = \Delta H = 0\ 097$ lb water evaporated per pound dry air.

Inspection of equation (2) shows that (H) is linear in w. Hence, one can construct on Fig. 9, the line marked (H) being drawn connecting the initial and final points just computed.

Since the air leaving the drier has a temperature of 150 F and a humidity of 0.105, Fig. 4 shows that its wet-bulb temperature is 129 F. This is plotted at the right hand side of Fig. 9. Since the stock maintains a wet-bulb temperature down to 20 per cent moisture, where $w = 0.25$, the corresponding humidity can be computed by the use of equation (2) or by reading directly from the diagram, the value being 0.0392. Fig. 4 shows that the corresponding wet-bulb temperature is 105 F. Any intermediate point on the wet-bulb temperature curve can be calculated similarly. The points for $w = 0.5$ are shown in Fig. 9.

Below the point, $w = 0.25$, the temperature of the stock begins to rise appreciably above the wet-bulb temperature. Its temperature at any given point in this range, for example at $w = 0.15$, may be computed as follows: At this point, $H = 0.0234$ (from equation (2)) and from Fig. 4, $t_w = 95$ F. Hence the wet-bulb depression, $t - t_w = 150 - 95 = 55$ F. The assumption made regarding the relation between stock temperature and moisture content in this range may be formulated:

$$\frac{\Delta t'}{t - t_w} = \frac{w}{0\ 25}$$

At the point $w = 0.15$, $\Delta t' = 33$ F, $t' = 117$ F. The temperature of the stock leaving the drier, similarly computed, is 136 F.

Fig. 9 thus computed gives in graphical form the information as to the temperature humidity relationships in the drier. The air requirements can be computed by equation (2). Thus, per 100 lb of dry stock, it is necessary to supply 633 lb of dry air. Furthermore, since from Fig. 4 it is seen that the volume of 50 per cent saturated air at 70 F, is 13.55 cu ft per pound; 8580 cu ft of room air must be supplied per 100 lb dry stock. Similarly, since the volume of 50 per cent saturated air at 150 F is 18.0 cu ft per pound, the volume of hot wet air discharged from the drier is 11,400 cu ft per 100 lb of dry stock. Finally, the heat necessary to supply to the drier, as a whole, or to any section of it, may be computed from equation (3).

High Temperature Drier

In the design of a high temperature drier unit a method of approach to the necessary calculations involved are outlined as follows:

Example 1 Cores 4 and 5 in. thick are to be dried by heating to a temperature at 400 F. An intermittent type box oven is to be used, size 12 x 14 x 10 ft with 856 sq ft surface having an average heat transfer of 0 3 Btu per square foot per degree per hour. Drying time as determined by test is 2 hr (Fig. 10) Cores weighing 6 tons, and 15-ton steel plates, trucks etc. are delivered to the drier at 70 F. The oven is heated by an external heater; the products of combustion and 66⅔ per cent recirculated air will be delivered to the oven at 825 F. Fuel oil of 19,980 Btu gross and 18,830 Btu per pound net heating value, weighing 6 75 lb per gallon and having 15 lb product per pound fuel

for perfect combustion. Cores consist of 91 per cent sand, 3 per cent oil binder, and 6 per cent water.

Solution Heat required per ton of cores

	Lb Material × Temp. Rise × Sp Ht. =	Btu
Sand	0.91 × 2,000 × (400 − 70) × 0 2 =	120,120
Binder	0 03 × 2,000 × (400 − 70) × 0 4 =	7,920
Water heating	0 06 × 2,000 × (212 − 70) × 1 0 =	17,040
Water evaporation	0.06 × 2,000 × 970 (Fig. 4) =	116,520
Water superheating (approx 50 per cent reaches 575 F)	= 0.5 × 0 06 × 2,000 × (575 − 212) × 0.45 =	9,800
Total Heat		271,400 Btu

HEATING LOAD FIRST HOUR

	HEATED TO		BTU
Sand	212 F	$\frac{142}{330}$ × 120,120 =	51,688
Binder[a]	212 F	$\frac{142}{330}$ × 7,920 =	3,408
Water	212 F	=	17,040
Evaporation	66 7%	0 667 × 116,520 =	77,680
Superheat	66 7%	0 667 × 9,800 =	6,530
Total Per Ton			156,346
For 6 ton		6 × 156,346 =	938,076
Steel plates	390 F	320 × 30,000 × 0 12 =	1,152,000
Radiation[b]	422 F Avg.	352 × 856 × 0.30 =	90,394
Total			2,180,470

HEATING LOAD SECOND HOUR

Sand	400 F	$\frac{188}{330}$ × 120,120 =	68,432
Binder[a]	400 F	$\frac{188}{330}$ × 7,920 =	4,512
Water			
Evaporation	33.3%	0 333 × 116,520 =	38,840
Superheat	33 3%	0 333 × 9,800 =	3,270
Total Per Ton			115,054
For 6 ton		6 × 115,054 =	690,324
Steel plates	460	70 × 30,000 × 0 12 =	252,000
Radiation[b]	575	505 × 856 × 0.30 =	129,684
Total			1,072,008

[a]Binder oxidizes and liberates heat, which is neglected in this calculation.

[b]Average value of coefficient is less than 0 3 because oven is not up to 575 F This is neglected 422 F is arrived at by taking area under curve as compared to area under 575 F ordinate.

Heat in 1 lb fuel oil	=		18,830 Btu
Heater Loss (10 per cent)	=	1883	
Duct Loss (5 per cent)	=	942	2,825 Btu
			16,005 Btu available to heat oven

Heat content of gases in 1 lb fuel oil at 825 F is 205 Btu (Fig. 8)

15 lb × 205	=	3,075 Btu sensible heat in products of perfect combustion
		12,930 Btu To heat air X and Y (Fig 11).

$$Y\,(S_{825} - S_{422}) + X\,(S_{825} - S_{70}) = 12930 \tag{4}$$

$Y = 2\,(X + 15)$ for 66 7 per cent recirculation.

where

S = heat content of air at temperature noted taken from Fig 8

(Recirculation and exhaust contains water vapor, products of combustion, and a greater portion of air. Heat capacities of all vary so little that they have all been assumed to be air)

$S_{825} - S_{422} = 190 - 91 = 99$

$S_{825} - S_{70} = 190 - 8.6 = 181\ 4$

Substituting values of Y, H, etc. in Equation 4,

$(2\,X + 30)\,99 + 181\ 4\,X = 12{,}930$

$X = 26\ 3$ lb excess air

$Y = 82\ 6$ lb recirculating air

Total = 26.3 + 82 6 + 15 = 123 9 lb air and products of combustion circulated per pound fuel burned.

Heat in air exhausted from oven at 422 F per pound fuel burned = 0 333 × 123 9 × $(S_{422} - S_{70})$ = 41 3 (91 − 8 6) = 3,400 Btu

Btu available for heating material = 16,005 − 3,400 = 12,605 Btu per pound fuel

Fuel used in first hour = 2,180,470 − 12,605 = 173 lb = 25.6 gal.

During the second hour the heater capacity will be much greater than required If an automatic oven temperature control operates on the oil supply, the delivery temperature of the air entering the oven and the quantity of oil burned will decrease, the air supply being constant.

Heat in air exhausted = 41 3 $(S_{675} - S_{70})$ = 41 3 (127 − 8 6) = 4880 Btu per pound fuel

Heat available for heating material = 16,005 − 4880 = 11,125 Btu.

Fuel used in second hour = 1,072,008 − 11,125 = 96 5 lb oil = 14 3 gal.

Total oil used per load = 25 6 + 14.3 = 39 9 gal

ESTIMATING METHODS

Values based on practical experience are available for rough estimating of drying problems. The temperature will drop approximately 8½ F per grain of water evaporated per cubic foot of air (measured at 70 F) or approximately 0.62 F per pound of air at any temperature. Air will drop 55 F per cubic foot for each Btu extracted. Generally air will absorb from 2 grains to 5 grains per cubic foot of air in one passage through an air drier, depending on the temperature and the degree of contact with the material. The amount of steam required to evaporate a pound of water will vary from 1½ lb to a more usual figure of from 2½ to 3 lb of steam per pound of water evaporated.

REFERENCES

Commercial Drying Apparatus, by L P Dwyer (A S.H.V E. TRANSACTIONS, Vol 22, 1916, p 479)

Artificial Drying with Special Reference to the Use of Gas, by G. C Shadwell (A S H. V E TRANSACTIONS, Vol. 23, 1917, p 231).

Drying by Evaporation, by F R Still (A S H.V E TRANSACTIONS, Vol. 23, 1917, p 255).

High Temperature Drying, by Burt S Harrison (A.S H V.E TRANSACTIONS, Vol 24, 1918, p 7).

The Temperature of Evaporation, by W H Carrier (A S.H V E TRANSACTIONS, Vol 24, 1918, p. 25)

Commercial Dehydration, by J E Whitley (A.S H V E TRANSACTIONS, Vol 26, 1920, p. 551).

Drying as an Air Conditioning Problem, by A W. Lissauer (A S H V.E TRANSACTIONS, Vol 27, 1921, p. 251).

A Chronological Survey of Drying and Driers, by J E Bolling (A S H.V E JOURNAL, October, 1921, p 715)

Calculations for Drying Design, by Grosvenor (*Transactions, A. I. Chem Eng*, 1908, p 184).

The Rate of Drying Solid Materials, by J Lewis (*Ind. Eng. Chem*, 1921, p. 427).

Adiabatic Drying of Hygroscopic Solids, by A M McCready, and W L McCalc (*Trans Am Inst Chem. Engr*, 1933).

Lignite Drier, by Lavine & Sutherland (*Chem and Met Eng*, July, 1929).

Enameling Oven Economy, by B S Harrison (*Fuels & Furnaces*, February, 1931)

Powdered Yeast Prepared by Spray Drying, by A W Farrell (*Food Ind*, December, 1931).

Factors that Influence Drier Performance, by A Weisselberg (*Chem. and Met Eng*, August, 1932).

Principles of Drying Lumber and Humidity Diagram, by H. D. Tiemann (*Forest Service Bul.* 104, 1912)

Symposium on Drying Articles by W. K. Lewis, W H Carrier, A E. Stacey and Fleming, R G. Metz, G. B Ridley, C O Lavett, D. J. Van Marle (*Jour Ind. Eng. Chem.*, May, 1921).

The Drying of Solids, by T. K. Sherwood (*Bul Mass. Inst Tech*, Nos. 237, 247 and 258).

Air Conditioning and Engineering, American Blower Co.

Combustion (*American Gas Association*, 3rd edition, 1932)

Die Trockentechnik, by M. Hirsch (Julius Springer, Berlin, 1932).

Drying (*Kent's Mechanical Engineers Handbook*, 10th edition, 1923, 11th edition, 1936)

Drying, by W H. Carrier (*Marks' Mechanical Engineers Handbook*, 3rd edition, 1930).

Drying, by Perry (*Chemical Engr. Handbook*, 1934).

Drying by Means of Air and Steam, by E Hausbrand (*D. Van Nostrand & Co.*, 1901)

Drying in Industrial Plants, by J O Ross.

Elements of Chemical Engineering, by Badger and McCabe (*McGraw Hill Co*, 1931).

Fan Engineering, Buffalo Forge Co.

Fuels and Their Combustion, by Haslam and Russell (*McGraw Hill Co.*, 1926).

Heat Transmission, by W H McAdams (*McGraw Hill Co.*, 1933).

Modern Drying Machinery, by H. B. Grenshaw, London, 1926.

Principles of Chemical Engineering, by Walker, Lewis, McAdams (Chapters on Evaporation, Humidity and Drying, 2nd edition, *McGraw Hill Co*).

The Kiln Drying of Lumber, by A. Koehler and R. Thelen, New York, 1926

The Kiln-Drying of Lumber, by H. D. Tiemann (*Lippincott*, 1920).

PROBLEMS IN PRACTICE

1 ● What makes a commercial adiabatic drier differ from a theoretical one?

The word *adiabatic* means no heat lost to the outside and that the sensible heat lost by the air is equal to the latent heat of the water evaporated In an actual drier, the solid containing the water, and the water itself must be heated to the temperature of evaporation, before evaporation can begin Radiation losses from the drier enclosure is the other factor causing deviation from the theoretical adiabatic process

2 ● What is a zone drier?

This term refers to a continuous drier where the drying medium is divided into two or more sections, in order to have better control of the temperature and humidity gradients through the drier, and often different velocities.

3 ● If a material enters a drier containing 70 per cent water and 30 per cent solids, and leaves the drier with 10 per cent water and 90 per cent solids, (a) what is the evaporation per pound of dried product? (b) What is the evaporation per pound of bone dry material?

a $\frac{90}{30} - 1 = 2$ lb water per pound dried product.

b. Water entering $= \frac{70}{30} = 233$ per cent on bone dry basis

Water leaving $= \frac{10}{90} = 11$ per cent on bone dry basis.

Water evaporated 222 per cent on bone dry basis.

Evaporation $=$ 2.22 lb water per pound bone dry material.

4 ● What items must be included in a calculation of the drier heat requirements?

a Heating water to be evaporated from the entering temperature to the temperature of evaporation.

b. Evaporating water to be removed.

c. Superheating evaporated water from the temperature of evaporation to the exit temperature of the air

d Heating material from entering to leaving temperatures.

e. Heating residual water from the entering to the leaving temperatures.

f. Heating conveyor or other supporting materials

g Radiation losses through the enclosure.

h. Sensible heat in the exit air.

5 ● The following conditions prevail in a drier; 250 lb water evaporated per hour. Air enters heater at 80 F dry-bulb and 65 F wet-bulb. Air exhausted from drier at 130 F dry-bulb and 100 F wet-bulb. Stock enters drier at 70 F. Heat required for warming stock and radiation losses are not considered. Fan is located ahead of heater. Find conditions of air entering and leaving drier, volume handled by fan, and temperature of air entering drier to supply the necessary heat, using Humidity Chart in Fig. 4.

Entering Air: Humidity, H = 0 01 lb water vapor per pound dry air.
Dew-point = 57 F
Per Cent Humidity, $\%H$ = 46

Leaving Air Humidity, H = 0 0355 lb water vapor per pound dry air.
Per Cent Humidity, $\% H$ = 32

Water pick up = 0 0355 − 0 01 = 0 0255 lb per pound bone dry air

Bone dry air circulated per hour = 250 ÷ 0 0255 = 9800 lb

Volume of air circulated at 80 F dry-bulb, and 46 per cent humidity
14 1 − 13 6 = 0.5 cu ft vapor (Fig 4).

Volume = 13 6 + (0 46 × 0 5) = 13 87 cu ft = 1 lb dry air + vapor

$$\text{Volume handled by fan at 80 F} = \frac{9800 \times 13\,87}{60} = 2260 \text{ cfm.}$$

Btu received by water = (130 − 70) × 1 0 = 60

Latent heat of steam at 130 F (Fig. 4) = 1019

Total = 1079 Btu per pound.

Heat used for evaporation per pound dry air = 1079 × 0 0255 = 27.43 Btu

For entering air: Humidity, H = 0 01, Humid Heat = 0 2425 Btu per pound (Fig. 4)

$$(t_1 - t_2) \times S = \text{Btu for evaporation}$$
$$(t_1 - 130) \times 0.2425 = 27\,43$$
$$t_1 = 247 \text{ F}$$

6 ● Given the following conditions, air 160 F dry-bulb, 49.6 per cent relative humidity (Φ), 29.92 in. Hg, barometric pressure, find the per cent H, absolute humidity.

For 160 F, p_s = 9.65 in. Hg (From Table 6, Chapter 1)

$$p = \Phi p_s = 0.496 \times 9\,65 = 4\,78$$

$$\frac{29\,92 - 9\,65}{29\,92 - 4\,78} \times 0.496 = 0\,40 \text{ or 40 per cent absolute humidity.}$$

Chapter 42

MOTORS AND CONTROLS

Direct Current Motors, Alternating Current Motors for Single Phase and Polyphase, Special Applications, Classification of Motors, Manual Control, Automatic Control, Pilot Controls, Direct Current Motor Control, Squirrel Cage Motor Control, Multispeed Motor Control, Slip Ring Motor Control, Single Phase Motor Control

THE electric motor, available in many different types suitable for various services, is now the most widely used form of prime mover. The equipment for starting, controlling and protecting these motors varies with the type and with the functions it is desired to attain. Motors used for heating, ventilating and air conditioning applications may be divided into two general classifications as follows:

1. For use with direct current.
2. For use with alternating current

DIRECT CURRENT MOTORS

There are three types of direct current motors available:

1. Shunt Wound.
2. Compound Wound
3. Series Wound

Shunt Wound motors being suitable for application to fans, centrifugal pumps, or similar equipment where the amount of starting torque required is relatively small, are used for the majority of applications in the field of heating, ventilating and air conditioning. They may be used on reciprocating pumps and compressors, if started under unloaded conditions.

Compound Wound motors are required for application to compressors, stokers, reciprocating pumps when started under loaded conditions, and also when applied to similar equipment where high starting torque is required. Whenever frequent starting makes high starting and accelerating torque desirable, or where sudden changes of load are encountered, compound wound motors are used

Series Wound motors find only limited application in a few special cases and are available in only a limited range of sizes.

Speed Characteristics

Direct current motors are available with speed characteristics of four types:

1. Constant speed.
2. Adjustable speed
3. Adjustable varying speed
4. Varying speed

Constant Speed motors may be shunt wound or compound wound. Shunt wound motors have a nearly flat speed-load characteristic, with a regulation of 15 per cent for up to ¾ hp, 12 per cent for one to 5 hp and 10 per cent for 7½ hp and larger, based on full load speed.

Compound wound motors have a speed regulation over the range from full load to no load of not more than 25 per cent, based on full load speed.

Adjustable Speed motors are usually shunt wound since it is impractical to maintain the proper relation between the shunt and series fields of compound wound motors when wide variations of the field strength are required to obtain the speed adjustment.

Adjustment of the speed of shunt wound motors is obtained by field control on motors rated at ¾ hp and larger, with the minimum or base speed at full field strength and higher speeds at reduced field strength (obtained by adding resistance in the field circuit). The speed regulation from no load to full load will not exceed 22 per cent for 2 to 5 hp; nor 15 per cent for 7½ hp and larger. Below 2 hp, the regulation may exceed 22 per cent. If closer speed regulation is required, specifically wound motors must be obtained.

Practically constant horsepower output is obtained at all speeds up to a ratio of 2 to 1. For higher speed ratios, the horsepower rating at the minimum speed is less than at the maximum speed, this difference varying with the speed ratio. High efficiency is maintained over the entire speed range. Most listed constant speed motors are suitable for operation up to a speed ratio of 2 to 1 by the use of proper control equipment.

Adjustable Varying Speed motors may be either shunt or compound wound and speed adjustment is obtained by adding resistance in series with the armature. The speed thus obtained is always below the rated full-field speed. Any standard shunt or compound wound constant speed motor may be used in conjunction with the proper armature resistor. The usual range of speed reduction is 50 per cent. The speed obtained for any setting of the resistor depends on the load of the motor and will vary with this load.

The speed regulation at high speed is comparable to a constant speed motor, but becomes poorer as the speed is decreased.

When operating at reduced speed, an increased torque requirement which the motor could easily handle at rated speed is easily sufficient to stall the motor; for example, a motor operating at two-thirds speed would be stalled by a torque about 50 per cent in excess of the normal requirement.

The efficiency of the motor is reduced as the speed is reduced, since the

loss in the resistor is greater at lower speeds. Speed reduction by armature control is usually selected where:

1 A wide speed range is not required
2 Close speed regulation is not necessary.
3 Operating time at reduced speed is short
4 Operating load at reduced speed is small so that the reduced efficiency can be ignored
5 The rating is less than 1 hp.

Varying Speed motors are series wound and the speed varies with the load on the motor. They should be used where:

1. The load is practically constant or increases with speed
2 The motor can easily be controlled by hand

They should not be used where there is a possibility of operation without load or at a reduced load, as the speed of the motor may become dangerously high.

For shunt wound motors with full field strength, the starting torque varies almost directly with the starting current, which is dependent on the resistance in the armature circuit. With varying positions of the starting rheostat, it is possible to obtain a wide range of starting torque, within the limits of starting current permitted by the power company.

A compound wound motor requires somewhat less current for the same starting torque. The maximum torque of shunt, series, and compound wound motors is limited by commutation.

ALTERNATING CURRENT MOTORS

Alternating current motors may be divided into two main groups, namely, (1) those operating on single phase current, and (2) those operating on polyphase current.

1. Single phase motors are available in four common types·
   a Capacitor motors
      1. Full capacitor
      2. Capacitor start-induction run
   b. Repulsion induction motors.
   c. Repulsion start, induction run motors
   d Split phase motors.

2. Polyphase (2 or 3 phase) motors are available in four common types.
   a. Squirrel cage induction motor
   b. Automatic start induction motor
   c. Slip ring, wound rotor induction motor.
   d Synchronous motor

Where the public utility supplying the current determines that a particular installation should be served with polyphase current, it is generally understood that the major portion of the motors will be for polyphase current, although it is commonly acceptable for the smaller motors to be single phase. This will limit the use of single phase current to the smaller motor ratings and the polyphase to the larger motors. Domestic and semi-commercial installations will invariably be single phase.

Single Phase Motors

Capacitor type motors are available in ratings up to 10 or 15 hp for general purposes. These motors are recommended for pumps, compressors and fan duty including housed centrifugal fans and propeller fans The general purpose motor is commonly known as a high torque capacitor motor having approximately 300 per cent starting torque with normal current and having a different value of capacitance for starting and running which is automatically changed over by a mechanical or electrical means.

Capacitor motors for *fan duty* are usually divided into the open high torque type for belted fans and the totally inclosed non-ventilated low torque type for propeller fans mounted directly on the motor shaft. The open low torque capacitor motor may be used with small centrifugal fans mounted on the motor shaft.

Although the motors for *belted fans* are called high torque, the available starting torque is somewhat less than the torque of the general purpose motor and the slip at full load is approximately 8 per cent. With this larger amount of slip, adjustable speed down to 60 or 70 per cent of rated speed may be obtained by line voltage variation. Motors for *propeller fan* drive may be supplied with sleeve bearings to obtain greater quietness in the smaller sizes where the fan thrust does not exceed approximately 25 lb. For larger fans, thrust ball bearing motors should be used. Low torque capacitor motors have approximately 50 per cent starting torque and do not change the value of capacitance from start to run.

Capacitor motors with *high slip* may have taps brought out from the main winding which when connected to the line, give a second speed of from 65 to 70 per cent of the normal speed. This type of motor must be specially designed for the individual fan, otherwise the correct low speed will not be obtained. Care should be exercised in applying it to centrifugal fans where restriction to the air flow through the use of adjustable dampers changes the motor load and consequently the speed. This same effect is also found in transformer speed controllers, however, a series of transformer taps allow for a selection which partially overcomes the effect of change in motor load.

Capacitor start-induction run motors are usually confined to the smaller horsepower ratings and differ from the capacitor motors by having no running capacitor. The value of starting capacitance used may vary with the different types of applications involved. These motors may be used for practically any of the applications met in air conditioning. However, consideration should be given to the fact that they are not as quiet as a capacitor motor.

Repulsion induction motors start as repulsion motors and operate under full speed as combined repulsion and induction motors through the inherent characteristics of the motor which has, in addition to the wire winding with commutator, a buried squirrel cage winding. No additional switching devices are required to change over from start to run. This and the repulsion motor described below may be used for constant speed drives where high starting torque is required and where commutator and brush noise is not a factor.

The *repulsion start-induction* run motor starts as a repulsion motor, has a switching means for transferring from start to run which short circuits the commutator and permits operation under full speed as a wound induction motor. This motor is suitable for applications similar to those for which the repulsion induction motor is used.

The *split phase* motor has a high resistance auxiliary winding in the circuit during starting which is disconnected through the action of a centrifugal switch as the motor comes up to speed. Under running conditions, it operates as a single phase induction motor with one winding in the circuit. These units are available for the lower horsepower ratings and when equipped with a high slip rotor may be used for adjustable varying speed through line voltage control.

Polyphase Motors

Squirrel cage induction motors are available in three types and a full range of sizes:

1 The normal torque, normal starting current squirrel cage motor has close speed regulation, high efficiency, high power factor, medium starting torque, high pull-out torque, and is suitable for general purpose applications This motor has a large current inrush and a low starting current power factor It operates with these characteristics only when started directly across the line on full voltage. When central stations require current limiting starting equipment on such motors, the starting torque is less. Current limiting hand operated starters are standard equipment.

2. The normal torque, low starting current squirrel cage motor has approximately the same torque as the normal current motor, but the starting current is about 20 per cent less than the normal torque motor on full voltage and ordinarily within the *National Electric Light Association* locked rotor current limits on sizes up to 30 hp.

This motor lends itself to automatic or remote control because no current limiting starting equipment is necessary up to and including 30 hp A magnetic starter with low voltage and thermal relay overload protection gives the most satisfactory service

3 The high torque, low current squirrel cage motor has a starting torque approximately 25 to 50 per cent greater than the normal torque motor on full voltage with starting current approximately 10 per cent less than the normal torque motor started on full voltage, but within the required limits on 30 hp sizes and smaller. These motors are also started directly across the line on full voltage through a magnetic starter or other approved starting device

These three types of motors are also available in two, three, or four speed designs with variable torque, or constant torque characteristics. Two speed motors may be either single, or two winding; three speed motors are single, two, or three winding; and four speed motors are two, three, or four winding. When a motor is wound with a winding for each speed, better operating characteristics may be obtained because no sacrifice is made for the other speed and operating characteristics approaching single winding motors may be expected.

Frequently, multispeed motors lend flexibility to an installation that cannot be obtained in any other way.

Multispeed motors are started directly across the line through magnetic starting equipment with overload and low voltage protection and compelling relays to insure starting on low speed regardless of the ultimate running speed. Starting on low speed limits the starting current to the starting current of the low speed winding and consequently lowers the maximum demand.

TABLE 1 CLASSIFICATION OF MOTORS

CURRENT	TYPE	SPEED CHARAC-TERISTICS	FULL VOLTAGE STARTING TORQUE	FULL VOLTAGE STARTING CURRENT	HP RANGE	TYPE OF APPLICATION SEE FOOTNOTE*
		Constant Speed Drives				
DIRECT	1. Shunt	Constant	Medium	Medium	All	(*a*) Fans and (*c*) Centrifugal Pumps
	2. Compound	Constant or Variable	High	Medium	All	(*b*) (*c*) (*e*) Reciprocating Pumps and frequent or hand starting
	3. Series	Variable	High	Medium	Small	(*d*) Fans direct connected
POLY-PHASE	4. Squirrel Cage General Purpose	Constant	Normal	High 6-8 Times	All	(*a*) Fans and (*c*) Centrifugal Pumps
	5. Squirrel Cage Medium Torque	Constant	Normal	Medium 5-6 Times	Medium Small	(*a*) Fans and Centrifugal Pumps
	6. Squirrel Cage High Torque	Constant	High	Medium 5-6 Times	Medium Small	(*b*) Reciprocating Pumps (*e*) and Compressors started loaded
	7 Automatic Start High Torque	Constant	High	Low 3 Times	Medium	(*b*) Reciprocating Pumps (*e*) and Compressors started loaded
	8. Slip Ring Wound Rotor	Constant	High	Low 1-3 Times with secondary control	All	(*a*) and Hoists (*b*) Reciprocating Pumps (*c*) and Frequent (*e*) or Hand Start
	9 Synchronous High Speed	Constant	Medium	Medium 5-7 Times	Medium Large	(*a*) Fans and Centrifugal Pumps
	10. Synchronous Low Speed	Constant	Low	Low 3-4 Times	Medium Large	(*a*) Reciprocating Compressors Starting Unloaded
SINGLE PHASE	11. Capacitor	Constant	High	Normal	Medium Small	(*b*) Pumps and Compressors

*Applications

a Drives having medium or low starting torque and inertia (WR^2) such as fans and centrifugal pumps or reciprocating pumps and compressors started unloaded

b Drives having high starting torques, such as reciprocating pumps and compressors started loaded.

c Similar to (*a*) except where frequent or hand starting (large WR^2) requires a higher starting and accelerating torque

d Fans direct connected

e Stoker drives

Table 1 Classification of Motors—(Continued)

Current	Type	Speed Characteristics	Full Voltage Starting Torque	Full Voltage Starting Current	Hp Range	Type of Application See Footnote*
Single phase	12 Capacitor Fan	Constant	High	Medium	Medium Small	(a) Fans—belted
	13. Capacitor Fan	Constant	Low	Medium	Medium Small	(d) Fans—direct
	14. Capacitor Start Induction Run	Constant	Any	Medium	Medium Small	(a) Fans (b) Pumps and Compressors
	15 Repulsion Induction	Constant	High	Medium	Medium Small	(a) Fans (b) Pumps and Compressors
	16. Repulsion Start Induction Run	Constant	High	Medium	Medium Small	(a) Fans (b) Pumps and Compressors
	17 Split Phase	Constant and Adjustable	Medium	Medium	Fractional	(a) Fans (b) Pumps and Compressors
		Adjustable Speed Drives				
Direct	18. Shunt Field Adjustment	Constant	Medium	Medium	All	(a) Fans and (e) Centrifugal Pumps
	19. Shunt Armature Resistor	Variable	Medium	Medium	All	(a) Fans and (e) Centrifugal Pumps
Polyphase	20. Squirrel Cage High Slip, Tapped Winding	Variable	Medium	Medium	Medium Small	(a) Fans
	21. Squirrel Cage High Slip, Transformer Adjustment	Variable	Medium	Medium	Medium Small	(a) Fans
	22. Squirrel Cage Separate Winding or Regrouped Poles	Constant Multi-Speed	Medium or High	Low	All	(a) Fans (b) Pumps and (c) Compressors

Table 1 Classification of Motors—(Continued)

Current	Type	Speed Characteristics	Full Voltage Starting Torque	Full Voltage Starting Current	Hp Range	Type of Application See Footnote*
Polyphase	23 Wound Rotor, Slip, Ring, External Secondary Resistance	Variable	High	Low	All	(a) Fans and (b) Centrifugal Pumps
Single phase	24 Capacitor High Torque Tapped Winding	Variable	High	Normal	Medium Low	(a) Fans, belt
	25 Capacitor Low Torque Tapped Winding	Variable	Low	Medium	Medium Low	(d) Fans, direct
	26 Capacitor High Torque Transformer Adjustment	Variable	Low	Low	Fractional	(d) Fans
	27 Capacitor Low Torque Transformer Adjustment	Variable	Low	Low	Fractional	(d) Fans
	28. Split Phase Regrouped Poles	Constant	Normal	Normal	Fractional	(d) Fans

Often where the central station requires current limiting starting equipment for the normal torque, normal starting current motor, it is advisable to use the normal torque low starting current multispeed motor.

High slip polyphase motors may be used for adjustable varying speed drives in a manner similar to that described for capacitor motors, with either a transformer speed regulator or tapped motor windings.

It is apparent from these motor characteristics that a squirrel cage motor may be selected for operating any air conditioning and allied equipment.

Automatic start induction motors are constructed with two windings on the rotor, one of which is a high resistance, squirrel cage winding used in starting and gives a high starting torque approximately the same as the high torque, squirrel cage A centrifugal mechanism within the motor switches to the second low resistance winding when the motor comes up to speed, thus obtaining running characteristics equal to the normal torque, normal current squirrel cage motor. The power factor of the starting current is high.

Slip ring wound rotor motors are built for two classes of service, constant speed and adjustable variable speed. The motors are identical in each case and use the same primary control, the only difference being in the secondary control.

Slip ring motors for constant speed service are used where high starting torque with low starting current is required for bringing heavy loads up to speed. The resistance is in the secondary or rotor circuit, only when starting, and is short circuited when the motor is up to speed.

For adjustable varying speed service, part or all of the secondary controller resistance is in the circuit whenever the motor is operating below full speed. The speed obtained with a given resistance in the secondary circuit is dependent on, and changes with the load on the motor. The horsepower developed by the motor is approximately proportional to the speed, whereas the power required by the motor is practically the same at reduced speed as at full speed, hence the efficiency at reduced speeds is much lower than at full speed.

Synchronous motors are ordinarily used only where there is a need for, or advantage in, obtaining power factor correction. It is necessary to consider each application as a special case which must be individually engineered, since for satisfactory operation, the combined moment of inertia of the compressor fly wheel and motor rotor must be correctly established.

The general classification of motors used for heating, ventilation and air conditioning is shown in Table 1.

SPECIAL APPLICATIONS

A few applications of motors may require special constructions such as splash proof, explosion proof, fully enclosed, and self-ventilated to meet hazardous or special duty conditions. These requirements are frequently encountered in certain industrial applications, in which cases it is necessary to select the motors from the viewpoint of service conditions, as well as the required operating characteristics to meet the demands of the machines being driven.

CONTROL EQUIPMENT FOR MOTORS

In selecting control for alternating and direct current motors it is necessary to determine whether the installation is to be operated by manual or automatic control. The available controls and the function of each group of apparatus may be outlined as follows

1 Manual Control:
- *a.* To establish current.
  - (1) Snap switch
  - (2) Knife switch
  - (3) Manually operated contactor
  - (4) Drum switch.
- *b* Establish current and add overload protective device
  - (1) Snap switch with overload element.
  - (2) Knife switch with fuse or thermal cutout.
  - (3) Manual contactor with overload protective device; also reduced voltage starting compensator.
  - (4) Drum switch with overload protection
- *c* Establish current and add overload and low voltage protective devices
  - (1) Not used.
  - (2) Not used.

(3) Manual contactor or reduced voltage compensator with overload and low voltage release.
(4) Drum switch equipped with latch coil to give low voltage release

2 Automatic Control

a. To start on full voltage
(1) Without overload device.
(2) With overload device.
(3) With combination overload device and knife switch

b. Reduced voltage starting
(1) Primary resistance type starter.
(2) Auto compensator type.
(3) Reactance type.

PILOT CONTROLS

In selecting pilot control devices to operate in conjunction with either manual or automatic motor control, it is necessary that they be classified as follows:

1. *Two Wire Control.* Most thermostats, float switches, and pressure regulators, provide two wire control which gives low voltage release. A three position pilot switch can be used in connection with this method and thus provide manual control. With a low voltage (12 or 20 volt) control circuit it is desirable to use a low voltage thermostat When this type of thermostat is used it will be found that a saving in the wiring cost results. When using the low voltage thermostat on a control circuit a relay and transformer panel should be used instead of the low voltage coil on the starter

2 *Three Wire Control* Momentary contact start and stop push button stations are usually furnished as standard accessories with automatic starters, which gives low voltage protection. This control cannot be used in combination with two wire pilot devices.

In selecting manual control for an alternating or a direct current motor, the common practice is to locate the control near the motor. When the control is installed at the motor, an operator must be present to start and stop or change the speed of the motor by operating the control mechanism. Frequently manual control is employed only as a device to give overload protection and another device is employed to start and stop the motor. Manual control is used particularly on small motors which operate unit heaters, small blowers, and room coolers in an air conditioning system In other cases manual control in the form of drums, when used with multispeed motors, is only used as a speed setting device with the starting and stopping functions operated automatically through thermostats, and pressure switches.

Because of the increasing complexity of air conditioning systems, heating, ventilating and air conditioning equipment is being operated on automatic control with less dependence on manual operation and regulation.

Automatic control of motor starters may be accomplished by the use of remote push button stations, by a thermostat, float switch, pressure regulator or other similar pilot devices. An added advantage of automatic control is that the main wiring for the starter may be installed near the motor, while the starter may be operated by a control device located elsewhere. In the majority of air conditioning installations, requiring motors 1 hp and larger, two or three phase alternating current is usually supplied.

DIRECT CURRENT MOTOR CONTROLS

Air conditioning installations using direct current power are now only used where alternating current is not available. Direct current motors are always started through starters, which are devices using a resistance to be put in series with the armature circuit during starting only, the resistance being gradually cut out as the motor comes up to speed. The starting current is held within safe limits by the use of the resistance.

The speed of a direct current motor may be regulated by the following methods:

1. Speed regulation by field control—by using a device with resistance to be put in series with the field winding. After the motor has been started to be used to increase the speed of the motor above full field speed.

2 Speed regulation by armature control—by using devices with resistance to be put in series with the armature circuit to be used to reduce the speed of the motor below full field or normal speed

3 Combinations of field and armature control, so that the starting, field control, or armature control may be combined in a single unit

Field control is usually preferred, depending on the size of the installation. For example, if a direct current motor were required with speed regulation between 1200 and 600 rpm, a choice of supplying a 1200 rpm motor with armature control or a 600 rpm motor with field control, both giving the same speed variation would be possible. While the 1200 rpm motor with armature control is lower in first cost than the 600 rpm motor with field control, the cost of operating the 600 rpm motor with field control is less and will save the difference in first cost over a period of time depending on the size of installation. A wide speed variation can be easily obtained in a direct current motor by using a combination of field and armature control.

SQUIRREL CAGE MOTOR CONTROL

To meet the requirements of various drives of an air conditioning system, three types of squirrel cage, two or three phase motors may be used:

1. Normal torque, normal starting current.
2. Normal torque, low starting current.
3. High torque, low starting current.

Because of the large current inrush of the normal torque, normal starting current motor, central stations usually require current limiting starting equipment on such motors above 5 hp. To meet the starting current requirements, manual or automatic current limiting starting compensators are used. These compensators are equipped with 50, 65 and 80 per cent voltage taps, the 65 per cent tap being regularly furnished when the compensator leaves the factory. Motors 5 hp and smaller have starting currents within the requirements of central stations and manual or magnetic, full voltage control may be used.

The normal torque, low starting current motor has a starting current which is approximately 20 per cent less than the normal current motor on

full voltage and well within the required current limits on 30 hp sizes and smaller. This motor, therefore, lends itself to across-the-line control because no current limiting equipment is necessary. In selecting motors for fans, pumps, or blowers, it should be noted that while the cost of the normal starting torque, low starting current motor is higher, the cost of full voltage control is lower, so that the total cost of low starting current motors with across-the-line control is lower.

A magnetic starter with low voltage and thermal overload protection gives the most satisfactory service. These switches may be controlled by remote push button stations, thermostats, or pressure switches to meet the requirements of any particular installation.

The high torque, low starting current motor has a starting current approximately 10 per cent less than the normal torque, low starting current motor when started on full voltage. These motors, most commonly used on compressor drive, can be started directly across-the-line with manual or magnetic starters.

Adjustable varying speed motor control by terminal voltage regulation requires a tap-changing switch manually or magnetically operated. Such a control switch operates to alter the voltage applied to the motor by contacting different auto-transformer voltage-ratio taps or by changing the amount of resistance inserted in the primary or line circuit.

MULTISPEED MOTOR CONTROL

To make an installation more flexible, multispeed motors are available with two, three or four speed designs, with variable torque, constant torque or constant horsepower characteristics. Multispeed may be started by means of manual or magnetic starting equipment.

When using automatic magnetic control with two, three, and four speed separate winding or consequent pole motors, control is obtained from a remote point by means of a push button master switch. The various speeds of the motor are obtained from the master switch by simply depressing the correct push button, which is known as selective speed control. It is commonly used in the smaller theatre installations where the fan and motor is located backstage and the speed control is located in the lobby.

Magnetic multispeed motor controllers may also be provided with a compelling relay which makes it necessary that the operator press the first speed button before regulating the motor to the desired speed. This assures the operator that the motor is always started at low speed before the motor is adjusted to one of the higher speeds. Starting on low speed limits the starting current to the starting current of the low speed winding, and therefore, permits the use of motors in sizes larger than ordinarily permitted by central stations for full voltage starting.

Timing relays, which provide for automatic acceleration, may be used for control. With the automatic acceleration feature, it is only necessary to press the button for the desired speed. The motor will always start in low speed and automatically step up to the desired speed.

Where the change of speeds does not occur at regular intervals, and where it is only necessary to change from one speed to another to take

care of seasonal requirements, a manual drum speed selector may be used. This drum is used to select the proper motor speed while an automatic starter is used to start and stop the motor.

The smaller size speed selector drums rated 10 hp at 220 volts and smaller may also be used as a motor starter to make and break the current, as well as, serving as a speed selector device. Reversible or non-reversible drums may be supplied depending on the requirements of the installation.

In the large size drums, a separate contactor must be provided to make and break the current. The contactor may be any approved starter. Overload and low voltage protection may be accomplished by using a magnetic starter. No push button station is required, the handle switch on the drum having the same characteristics as a three wire push button station.

In selecting two speed motors for fan, pump, blower, or compressor drive it will be found that the two winding motors are more expensive than the single winding. The control for two speed, two winding motors is more economical and the combined price of the motor and contactor is only slightly higher. Because of the better performance of the two speed motor and the factor of safety in having two independent motor windings, the increased cost is considered worth the difference.

SLIP RING MOTOR CONTROL

When close speed regulation and low starting current is required slip ring or wound rotor motors are used. Slip ring motors are built for two classes of service, constant speed and adjustable varying speed. The motors for the two classes of service are identical, the only difference being in the secondary control used with the motors. Control for both primary and secondary of a slip ring motor is required.

The primary control for a constant or adjustable speed is the same type as used with squirrel cage motors. Manual or magnetic starters, across-the-line type, may be used depending on the installation.

The starting current and starting torque of a slip ring motor are almost entirely dependent on the amount of resistance in the secondary control and in the manner in which the secondary control is operated. The *National Electric Manufacturers Association* has adopted service classifications which allow a selection of resistors permitting a starting current on the first contact of resistance varying from approximately 25 per cent of full load current to approximately 200 per cent of full load current or more, and permitting the resistor to remain in the secondary circuit of the motor for a period varying from not more than 15 seconds during an interval of operation from 4 minutes to continuous.

Speed regulation of a slip ring motor is obtained by inserting resistance in the secondary circuit and usually provides for a 50 per cent speed reduction when the motor takes its full rated current at normal speed. As resistors are supplied for both fan duty and constant torque duty, care should be taken in selecting the proper resistors.

Slip ring motors when used with centrifugal pumps and fans should have fan duty resistors. Because of the low current inrush of the fan and pump

load a starting resistor *NEMA* classification No. 15 may be used. For speed regulation resistor, classification No. 93 should be selected. On a compressor drive using an unloader, a constant torque resistor classification No. 15 should be used. If the compressor is started under load, *NEMA* classification No. 56 or 76 are used. For constant torque speed regulation, resistor No. 95 is used.

SINGLE PHASE MOTOR CONTROL

Where three phase current is not available or where single phase operation is preferred, then single phase repulsion induction, capacitor type or multispeed single phase motors may be used. Since the starting currents of all single phase motors are required to be within the starting-current limits established by the local power-supply company, a suitable type of starter may be chosen from the following selection:

1. Enclosed two pole manually operated motor starters with thermal overload protection.
2. Enclosed two pole automatic motor starter operated by a push button, thermostat or similar device, with thermal overload relay and low voltage protection.
3. A manual or magnetic resistance type starter with low voltage protection
4. A manual or magnetic control for pole changing motors and for adjustable varying speed motors using an auto-transformer or resistance in the primary circuit to obtain line (or terminal) voltage drop.

In selecting across-the-line control for single phase capacitor type motors it is usually very desirable to use three pole across-the-line starters. Control for multispeed, single phase capacitor motors may be selected from tables on three phase rating when consideration is given to the increased current and the necessary switching of connections.

PROBLEMS IN PRACTICE

1 ● When motors are being considered as prime movers, what are some of the basic considerations that determine the final selection of the correct unit?

a. The kind of current available for driving the necessary motors is a primary consideration. There are two groups of motors available for driving the equipment on any job, which are the direct current type or alternating current type. The proper group selection depends entirely on the current available.

b It is also necessary to decide whether constant speed or variable speed operation is desired.

c. Consideration must also be given to the type of service required.
1. Whether variable torque or constant torque motors will be required.
2. Whether a high starting torque is required or whether a relatively small starting torque is required.

d It is important to take into consideration the atmospheric conditions surrounding the motor location

2 ● When using direct current motors: *a.* What three types are available as regards their windings; *b.* What four types are available with reference to their speed characteristics?

a. Shunt wound, compound wound, and series wound.

b. Constant speed, adjustable speed, adjustable varying speed, and varying speed.

3 ● With direct current motors as prime movers what type would you use: *a*. For driving a fan; *b*. For driving a compressor?

a A fan requires a relatively small starting torque, therefore, a shunt wound motor would be ideal for this type service

b A compressor has a constant torque, therefore, a compound wound motor would be the proper selection for this duty

4 ● What is one of the important factors that should be taken into account when a series wound direct current motor is being considered?

With a series wound motor the speed varies with the load, therefore this type should never be used where there is a possibility of the motor operating without being loaded. The resultant high speed may prove to be dangerous

5 ● With the use of alternating current motors what two groups are generally considered?

Motors using single and polyphase power supply.

6 ● Under the alternating current group of motors what common types are available: *a*. For single phase duty; *b*. For polyphase duty?

a Capacitor high torque, capacitor fan—(1) high torque, (2) low torque, capacitor start-induction run, repulsion induction, and split phase

b Squirrel cage—(1) general purpose, (2) medium torque, (3) high torque, automatic start high torque, normal torque normal current, normal torque low current, high torque low current, slip ring wound rotor, synchronous high speed, synchronous low speed

7 ● What is the most commonly used of the polyphase motors?

The squirrel cage induction motor is the type most generally used for ordinary application.

8 ● With the use of squirrel cage motors what speed characteristic is available and what construction is used to make these more flexible?

The squirrel cage motor is basically a constant speed motor. However both single phase and polyphase high slip motors are used for adjustable varying speed drive through the use of line voltage control. When using an adjustable varying speed motor, particularly with a centrifugal fan and to a somewhat lesser extent, with a propeller fan, special care should be taken to assure that the fan is closely motored (*i e*, adequately loads the motor) in order to obtain the desired speeds under reduced speed operation To make the squirrel cage motor more flexible, multispeed units are used quite frequently. These units may be single winding for the two speed unit or for different number of windings depending upon the number and combination of speeds required For two speed single winding units the second speed is always one-half of top speed

9 ● Differentiate between synchronous speed and full load speed of a motor.

Synchronous speed is the theoretical or no load speed With the induction motor there is a certain amount of slip depending upon the load. As a rule, at full load, the speed is approximately 96 per cent of synchronous speed, however, motor manufacturers generally list full load speeds on their motor name plates

The synchronous type of motor has a full load speed which is the same as the synchronous.

10 ● What are the general requirements usually recommended by the power company with reference to connecting polyphase motors to the power line?

For motors up to and including 5 hp, normal torque, normal starting current type of units can be connected directly to the line

For motors from 5 to 30 hp, both high torque and normal torque, low starting current types of units can be used with across-the-line type of control.

Above these sizes, it is necessary to furnish current limiting starting equipment.

It is always advisable to check with local power companies as there are no standards for

connecting of loads on the power line and they are likely to vary with different power companies

11 ● In controlling direct current motors what two methods are used, what speed ranges are obtained, and what is the relative efficiency of each method?

In controlling direct current motors, resistance is placed in either the armature circuit or the field circuit For armature control, the speed is reduced with the increase of resistance With the field control, the speed is increased with the addition of resistance in the field circuit

For most listed direct current motors, it is possible to obtain operation up to a speed ratio of two to one with field control equipment. This type of control is used in connection with shunt wound motors for best results

For speed adjustment by resistance in series with the armature circuit, a reduction of 50 per cent in speed can generally be obtained This control can be used with either shunt or compound wound motors

The field control method of changing speeds on direct current motors is the most efficient. Due to the large current in the armature circuit, this method results in a high loss when the speed is reduced any appreciable amount It is well to remember that with field control only constant horsepower output is obtained, therefore, care should be taken that the motor at normal speed is large enough to care for any increase in load as a result of speeding up the unit.

12 ● What reduction in speed is possible and how is it obtained when alternating current slip ring motors are used?

Speed variation in slip ring motors is obtained by inserting resistance in the secondary circuit. This generally allows for a 50 per cent speed reduction when it is fully loaded at normal speed.

From 20 to 30 per cent speed reduction can be obtained through the use of line voltage control of an adjustable varying speed motor with a fan closely motored (*i e.*, the fan approximately fully loads the motor)

Chapter 43

TEST METHODS AND INSTRUMENTS

Pressure Measurement, Temperature Measurement, Air Movement, Humidity Measurement, Carbon Dioxide Determination, Dust Determination, Flue Gas Analysis, Measurement of Smoke Density, Heat Transmission, Eupatheoscope Problems in Practice

SEVERAL types of measuring apparatus are available for accurately determining the thermal capacity and air movement of gaseous vapors and homogeneous materials. This chapter gives a brief description of the principal instruments used in connection with the proper control and testing of heating and air conditioning installations.

PRESSURE MEASUREMENT

Atmospheric pressure is usually measured by a *mercurial barometer* which, in its simplest form, consists of a glass tube about 3 ft long, closed at the upper end, filled with mercury and inverted in a shallow bath of mercury. The pressure of the atmosphere on the exposed top of the mercury in the cistern supports a column of mercury in the tube to a height of about 30 in. Readings are taken of the height of the column between the levels of mercury in the tube and in the cistern. Atmospheric pressure is the same as the pressure exerted by this supported column of mercury, and, in pounds per square inch, is equal to its height in inches times 0.491, which is the weight in pounds of 1 cu in. of mercury. At latitude 45 deg and sea level, and at a temperature of 32 F, the atmosphere will support a column of mercury 29.921 in. in height. The pressure of 14.7 lb per square inch, derived by multiplying 29.921 by 0.491, is called *standard* or *normal barometric pressure*. Since the height of the barometer depends on the density of the mercury as well as on the pressure of the atmosphere, and since the density is dependent on the temperature, mercurial barometer readings should always be corrected for temperature. An *aneroid barometer* contains no liquid; it is portable but less accurate than the mercurial barometer. Atmospheric pressure in bending the thin corrugated top of a partially exhausted metallic box, or in distorting a thin-walled bent tube of metal, is made to move a pointer.

Pressures above or below atmospheric are usually measured by means of gages which indicate the difference between the pressure being measured and atmospheric pressure at the same time and place. A gage which indicates pressures higher than atmospheric is known as a *pressure gage*, and a gage which indicates pressures lower than atmospheric is known as a *vacuum gage*. The most common type of these gages contains a flexible hollow metal tube of oval cross section, known as a *Bourdon tube*. When subjected to unequal inside and outside pressures, this tube tends to

straighten out, and a pointer motivated by this straightening indicates the pressure difference on a suitably graduated scale.

High vacuum readings such as are encountered in condenser and steam jet refrigeration practice are commonly obtained by the use of mercury column vacuum gages. When the readings obtained with the mercurial barometer and those with the mercury vacuum gage have both been corrected to 32 F, the difference in the two readings will give the absolute vacuum in inches of mercury. The following equation may be used to make corrections for temperature:

$$h = h_1 [1 - 0.000101 (t_1 - t)] \qquad (1)$$

where

h = height of mercury column corrected to temperature t

h_1 = actual height of mercury column.

t_1 = actual temperature of mercury column.

t = temperature to which column is to be corrected

A gage which indicates pressures slightly above or below atmospheric is known as a *draft gage*. It is essentially a *U* tube containing either water, kerosene, alcohol, or mercury, with one leg exposed to the air and the other connected to a point where the pressure is to be determined. When the pressure being read is equal to atmospheric, the level of the liquid in the legs will be the same, indicating a zero gage pressure. When a pressure is applied to one leg, one side will fall and the other will rise an equal amount. The difference in height between the two liquid levels indicates the pressure expressed in inches of liquid used in the gage.

Various forms of high sensitivity draft gages[1] frequently called micromanometers are available for the measurement of small pressure differentials and may be sensitive to pressures as small as 0.001 in. of water. These gages are often useful where measurements are to be made on pressure differentials less than 0.1 in. of water, although their total range may extend as high as 5 to 10 in. of water.

TEMPERATURE MEASUREMENT

In engineering work, *mercurial thermometers* are largely employed to measure the intensity of heat. These depend on the uniform expansion of mercury to indicate changes in temperature. An amount of mercury held in a sealed tube with a bulb at one end will rise to one definite level when immersed in melting ice, and to another definite level when immersed in boiling water. These two points are marked, and the space between them is divided into a number of equal portions, each of which is called a degree. In the Fahrenheit scale, there are 180 deg thus obtained, while the centigrade scale has 100 and the Réaumur has 80. Like divisions are marked off on the column above and below these two determined points in order that a greater range of temperature may be read.

Thermocouples[2] may be used to measure any range of temperatures up to 2,900 F. When two dissimilar metals are joined at two points and a

llinois Micromanometer, University of Illinois *Engineering Experiment Station Bulletin No* 120, p. 91

tudy of the Application of Thermocouples to the Measurement of Wall Surface Temperatures, by Kratz and E L Broderick (A S H V E TRANSACTIONS, Vol 38, 1932)

temperature difference exists between these junctions, an electromotive force will be developed. Its magnitude depends on the composition of the wires and the difference in temperature between the junctions. A potentiometer or sensitive galvanometer of high resistance connected to the thermocouple will give a deflection which is a function of the temperature difference between the hot and cold junctions. Thermocouples connected in series are called *thermopiles.* Thermocouples for the measurement of high temperatures are calibrated with the aid of the known melting points of pure metals.

Resistance thermometers are suitable for temperature measurements up to 1800 F. These thermometers depend for their operation on the change of resistance with temperature of a platinum, nickel, or copper wire coil, and they are calibrated in the same way as thermocouples.

Pyrometers of various types may be used for temperatures above 500 F. The *mercurial pyrometer* is a thermometer with an inert gas, such as nitrogen or carbon dioxide, above the mercury column to prevent the mercury from boiling. The *radiation pyrometer* consists of a thermopile upon which the radiation from a hot source is focused by a concave mirror or lens. A sensitive galvanometer or potentiometer with a calibrated temperature scale indicates the thermo-electromotive force created by the heat on the thermopile. The *optical pyrometer* measures radiant energy by comparing the intensity of a narrow spectral band, usually red light emitted by the object, with that emitted by a standard light source (electric lamp). *Thermo-electric pyrometers* operate on the same principle as thermocouples. When measuring high temperatures, it is customary to hold the cold junction at room temperature and this may cause some error if the room temperature is above or below the calibration point. For extremely precise temperature measurements, the cold junction is usually immersed in melting ice to fix the cold junction temperature. Various forms of hand-operated and automatic cold junction temperature compensators are also available.

In the measuring of room temperatures care must be exercised to prevent the results from being affected by the body heat of the observer, by drafts from doors, windows and other openings, or by radiant heat from some local source such as a radiator or wall. All glass thermometers should be mercury thermometers with engraved stems. The total graduations of the thermometers should be from 20 to 120 F, in one degree graduations. No ten degrees should occupy a space of less than one-half inch. The accuracy throughout the whole scale must be within one-half degree. The operator should take hold of the top and no part of the body, including the hand, should be nearer than 10 in. to the bulb. The thermometer should not be closer than 5 ft to any door, window, or other opening; should not be closer than 12 in. to any wall; and should be between 3 and 5 ft from the floor. A sling instrument should be used for extreme accuracy. Thermocouples or resistance thermometers may also be used for room temperature measurements, an advantage being that the operator can read temperatures from outside the room if desired, and thus eliminate the errors which might be caused by his presence close to the temperature measuring device.

For measuring duct temperatures a duct thermometer should be used, with the bulb extending into the duct at least 6 in. When the thermo-

meter is to be permanently located in the duct, a pipe flange or nipple should be used to receive the threaded portion of the thermometer stem. When the thermometer is not to be permanently located, a cork or rubber stopper may be placed around the stem to prevent errors from air leakage. Readings should be taken at various locations in a duct so due consideration may be given to temperature stratification. Other forms of temperature measuring devices may be used, but the active part must be at least 6 in. from the duct wall.

Recording instruments may be used for testing and for making continuous records of operation. Potentiometer and Wheatstone bridge recorders for thermocouples and resistance thermometers respectively may have accuracies of $\pm$ 1/8 per cent of their range, or, for example, to $\pm$ 1 F in a range of 0 to 300 F. This accuracy compares favorably with that of other forms of temperature measuring devices.

AIR MOVEMENT MEASUREMENT

The quantity, velocity and pressure of air moved by a fan or flowing through a duct or grille may be determined by various methods. The instruments in common use are the Pitot tube, anemometer, direct reading velocity meter, and Kata-thermometer, the latter being suitable for low air velocities and being commonly used for measurements at points where the air is not confined in a duct. Electrical anemometers are also available, operating on the principle of measurement of the variation of resistance of a hot wire cooled to various degrees by air velocities past the wire. The use of calibrated nozzles, orifice plates, and Venturi meters are recognized methods, which, however, have little application in connection with ventilation practice.

Pitot Tube

This usually consists of two tubes, one within the other, which when properly held in the air stream will register the total or impact pressure and the static pressure, respectively. If these tubes are connected to opposite sides of a water column, or other type of manometer, the recorded pressure will be the differential or velocity pressure. Volume measurements may thus be made in a duct of known area. Pitot tube measurements are preferably used for air velocities exceeding 20 fps. Volumetric determinations from Pitot tube readings should take into account the barometric pressure and the temperature and humidity of the air measured.

Air flow in ventilation practice is generally in the turbulent range. When stratification of velocity, vortex motion, or violent eddy currents of air in ducts exist, accurate velocity pressure measurements are difficult. To insure accuracy a straight section of duct from 5 to 10 times its own diameter is desirable in order to straighten out the air currents. If it is necessary to take Pitot tube readings in shorter sections of straight duct, the results must be considered subject to some doubt and checked accordingly. For accurate work it is necessary to make a traverse of the duct, dividing its cross section into a number of imaginary equal areas and taking a reading in the center of each, the average of the velocities corresponding to these pressures giving the true velocity in the duct.

Anemometer

This instrument is delicate, and requires frequent calibration when accuracy is desired. The vanes of the instrument should never be touched and it should never be held in air having a velocity greater than that for which it is calibrated. Readings taken directly in a fan inlet or discharge are likely to harm the instrument because of excessive velocities. In duct measurements the same procedure is followed as for the Pitot tube. The anemometer usually reads directly in linear feet. To obtain the velocity in feet per minute, the reading must be divided by the elapsed time in minutes.

The following procedure for obtaining anemometer readings is based on research conducted at *Armour Institute of Technology* in cooperation with the A.S.H.V.E. Research Laboratory[3].

Supply Grilles. The surface of the grille should be marked off into a number of equal areas approximately 6 in. square. A 4-in. anemometer should be used and should be held at the center of each section in contact with the grille (or as close as possible) for a period of time sufficient to insure an average reading. In the case of supply grilles, the instrument should always be held with the dial facing the operator. The average of the corrected readings should then be used in the following formula to obtain the flow in cubic feet per minute:

$$cfm = CV\frac{A + a}{2} \text{ or } \frac{CVA\,(1 + p)}{2} \quad (2)$$

where

V = average of corrected anemometer readings, feet per minute
A = gross area of grille, square feet.
a = net free area of grille, square feet.
p = percentage of free area of grille expressed as a decimal.
C = a coefficient that varies with the velocity from grille and may vary slightly with type of grille For average use, with supply grilles, C can be taken as 0.97 at velocities from 150 to 600 fpm, and as 1 00 at higher velocities.

Particular care should be exercised in the case of long, narrow grilles. The nature of the approach sometimes results in there being a narrow strip along the top or bottom of the grille through which no air will be flowing. This may be detected by holding the anemometer completely out of the air stream and then moving it slowly inward over the grille until the vanes just start to move. The distance which the vanes extend over the grille opening at this moment will indicate the width of the dead strip. Only the remaining portion of the grille should be considered in making the calculations for gross and free area.

Exhaust Grilles. The surface of the grille should be marked off and readings taken in the same manner as with supply grilles, except that the instrument should be held with the dial facing the grille, and in contact with it. The traverse should be taken at a uniform rate, allowing sufficient time in each space to minimize the percentage of error. In the case of exhaust grilles it is found that the formula

$$cfm = KVA \quad (3)$$

[3]Measurement of Flow of Air through Registers and Grilles, by L E Davies (A S H V E TRANSACTIONS, Vol. 36, 1930, Vol 37, 1931, and Vol 39, 1933)

in which

V = average indicated velocity obtained by the anemometer traverse.
A = gross area of grille, square feet.
K = coefficient determined by experiment. For average use, with exhaust grilles, K may be taken as 0.8 for all usual velocities.

This formula is of advantage, especially with ornamental grilles, in that the free area need not be measured.

The flow of air through registers and grilles is of considerable importance, being frequently the only convenient method of measuring the volume of supply air to a room. While duct measurements, if available, are more dependable, grille measurements provide a fairly accurate method, if care is taken in the technique of using the anemometer.

Kata-Thermometer

The Kata-thermometer can be used to determine air velocities provided the walls and surrounding objects are at or near the room temperature. Especially at low velocities it constitutes a useful instrument for readily detecting drafts.

The instrument is essentially an alcohol thermometer with a bulb approximately ⅝ in. in diameter and ½ in. long with a stem 8 in. long reading from 100 F to 95 F, graduated to tenths of a degree. To take readings the bulb is heated in water until the alcohol expands and rises into a top reservoir. The time in seconds required for the liquid to fall from 100 F to 95 F is recorded with a stop watch and this time is a measure of the rate of cooling.

The dry Kata loses its heat by radiation and by convection so for constant velocities the time of cooling is a function of the dry-bulb temperature of the surrounding air. The wet Kata, which has a cloth covering fitted snugly around its bulb, loses heat by radiation, convection, and evaporation, and for constant velocities its rate of cooling is a function of the wet-bulb temperature of the air irrespective of the dry-bulb temperature or relative humidity. It does not follow, however, that the difference in rate of cooling of the dry and the wet Kata is caused by evaporation. A change in the wet-bulb temperature produces a change in the surface temperature of the wet Kata which in turn affects the heat lost by radiation and by convection.

Several precautions should be taken to obtain the best results with this instrument:

1. To obtain velocity readings use the dry Kata since the error in timing is reduced.
2. The instrument should be heated and allowed to cool two or three times before recording the final time of cooling The first reading is not reliable
3. All traces of moisture must be removed from the dry Kata before timing to eliminate error introduced by evaporation.
4. Use only the formula applying to a particular instrument. Each Kata receives an individual calibration.

HUMIDITY MEASUREMENT

The sling psychrometer is the recognized standard instrument for determining humidities. In order to obtain accurate readings considerable

skill is required on the part of the operator. The wicking and water must be clean and the temperature of the water should be slightly above the wet-bulb temperature of the surrounding air. The psychrometer should be swung rapidly and several and frequent observations should be made to see that the wet-bulb temperature has become stationary before the final reading is noted. Care should be taken that the wet-bulb has reached a minimum temperature, but the wick must still be moist. Standard psychrometric tables should be used[4].

In making wet-bulb measurements below 32 F the same procedure is followed as above 32 F. The water is liquid at the start, but as the sling is operated it will freeze rapidly enough so that in quickly giving up the latent heat of fusion, the indicated wet-bulb temperature may drop below the actual wet-bulb temperature. After the liquid on the bulb has become thoroughly frozen the wet-bulb temperature will rise to normal. A very thin film of ice is more desirable than a thick film. Care must be taken to read the temperatures in the region below 32 F accurately because the spread between the wet- and dry-bulb is small.

In taking humidity readings in ducts it is usually impracticable to use a sling psychrometer. For this work the stationary hygrodeik arranged for bolting on to the side of the duct, with two bulbs extending into the duct, will be found very convenient. Owing to the velocity of the air passing over the bulbs within the duct an accurate reading will be secured, corresponding to that given by the sling psychrometer.

Various forms of humidity recorders are available, some merely recording wet- and dry-bulb temperatures, and others recording relative humidity directly. Any form of wet- and dry-bulb device must have sufficient air velocity over the thermometer bulbs to insure accurate readings; this velocity should be secured by a fan if the air is not itself in motion, as in a duct. For extremely low humidities, or for humidity measurements above 212 F, a thermal conductivity method is available[5].

CARBON DIOXIDE DETERMINATION[6]

At ordinary concentrations carbon dioxide is not harmful. The amount of carbon dioxide in the air is a convenient index of the rate of air supply, and of the distribution of the air within rooms. Unequal carbon dioxide concentrations in parts of a room indicate improper air distribution.

The Petterson-Palmquist apparatus has been generally accepted as the standard device for the determination of carbon dioxide in air investigations. The principle involved is the measurement of a given volume of air, the absorption of the contained carbon dioxide in a caustic potash solution, and the remeasurement of the volume of air at the original pressure in a finely graduated capillary tube, the difference in volume representing the absorbed carbon dioxide. (See Report of Committee on Standard Methods for Examination of Air, *American Public Health Association*, Vol. 7, No. 1; *American Journal of Public Health*, Jan., 1917.)

---

[4]Psychrometric Tables for Vapor Pressure, Relative Humidity and Temperatures of the Dew Point; U. S Department of Agriculture, Weather Bureau, Washington, D C

[5]Gas Analysis by Measurement of Thermal Conductivity, H A Daynes, Cambridge Press, 1933

[6]Indices of Air Changes and Air Distribution, by F C Houghten and J L Blackshaw (A S H V E TRANSACTIONS, Vol 39, 1933)

A thermal conductivity method may also be used to measure carbon dioxide in air over a range of 0 to 1.5 per cent[7].

Where field conditions are such that this apparatus may not be conveniently used, as in street cars, air samples may be collected in clean bottles having mercury-sealed rubber stoppers, and these may be subjected to laboratory analysis.

DUST DETERMINATION

Many laboratory methods have been developed to measure the dust in the air. These involve the collection of dust on sticky plates, on filter paper, in water, on porous crucibles, or by electric precipitation, and the subsequent determination of the amount of dust by microscopic counting, weighing, or titration. While there is no standard method, the Hill dust-counter, using a microscope, the impinger[8], using chemical changes in water, and the Lewis sampling tube[9], involving the analytical weighing of a porous crucible, are accepted. All test results should be accompanied by the name of the instrument used as great variation in counts with the different instruments will be obtained. The American Society of Heating and Ventilating Engineers has developed a code[10] for the testing and rating of air cleaning devices used in general ventilation work.

FLUE GAS ANALYSIS

The analysis of flue gases by chemical means is made with the *Orsat apparatus*. A solution of KOH is used to absorb the CO_2. Free oxygen is absorbed by a mixture of pyrogallic acid and KOH. The solution for absorbing the CO is cuprous chloride. The apparatus consists of a burette surrounded by a water jacket, to receive and measure the volume of gas. The burette is connected by a manifold of glass to *pipettes* containing liquids for absorbing CO_2, O_2 and CO.

Various forms of automatic indicating and recording gas analysis devices are available, operating on either chemical or physical principles. Such devices are convenient for plant operation.

MEASUREMENT OF SMOKE DENSITY

Relative smoke density is usually measured by comparison with the Ringelmann Chart (Fig. 1). In making observations of the smoke issuing from a chimney, four cards ruled like those in Fig. 1, together with a card printed in solid black and another left entirely white, are placed in a horizontal row and hung at a point 50 ft from the observer and conveniently in line with the chimney. At this distance, the lines become invisible, and the cards appear to be of different shades of gray, ranging from very light gray to almost black. The observer glances from the smoke coming from the chimney to the cards, which are numbered from 0 to 5, determines which card most nearly corresponds with the color of the smoke, and makes a record accordingly, noting the time. Observa-

[7]Loc Cit Note 5

[8]*Public Health Bulletin*, No 144, 1925, U. S. Public Health Service.

[9]Testing and Rating of Air Cleaning Devices Used for General Ventilation Work, by Samuel R Lewis (A S H V E. Transactions, Vol 39, 1933)

[10]A S.H V.E Standard Code for Testing and Rating Air Cleaning Devices Used in General Ventilation Work (A S H.V E. Transactions, Vol 39, 1933)

tions are made continuously during one minute, and the estimated average density during that minute recorded. The average of all the records made during a boiler test is taken as the average figure for the smoke density during the test, and the entire record is plotted on cross-section paper in order to show how the smoke varied in density from time to time.

Smoke Recorders

Smoke recorders are available which give a much more accurate indication of the amount of smoke being produced than does the Ringelmann Chart. They all depend upon projecting a beam of light through the smoke flue or through a separate compartment from which a sample of the flue gas is drawn continuously. The light of the beam which passes

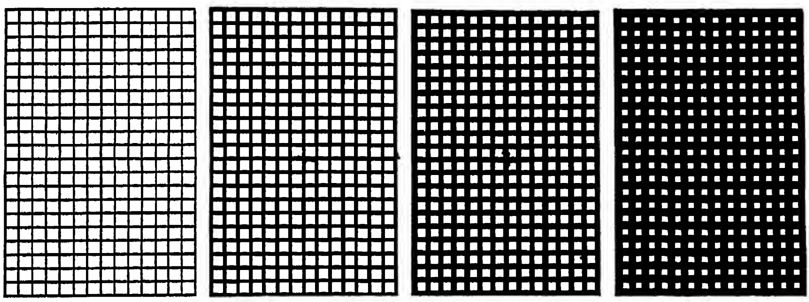


FIG. 1. RINGELMANN SMOKE CHART

through without being absorbed by the smoke is measured to determine the smoke density. Most of these instruments make use of a photoelectric cell or a thermopile to measure the relative amount of light which has not been absorbed. Standard electrical instruments serve for indicating or recording.

MEASUREMENT OF RATE OF HEAT TRANSMISSION

The standard methods of testing built-up wall sections are by means of the *guarded hot-box*[11] and the *guarded hot-plate*[12]. The *Nicholls heat-flow meter*[12] may be used for testing actual walls of buildings.

It would be obviously impossible to determine the air-to-air heat transmission coefficients of every type of wall construction in use with the heat-flow meter, the guarded hot-box or the guarded hot-plate on account of the great amount of time involved. Hence, the method of computing the coefficients from the fundamental constants must be resorted to in most cases. The guarded hot-plate is used to determine the fundamental constants. The heat-flow meter, guarded hot-box and guarded hot-plate tests can be used to good advantage in checking the accuracy of the computed values.

[11]Standard Code for Heat Transmission through Walls (A S H V.E TRANSACTIONS, Vol 34, 1928) and Report of the Committee on Heat Transmission, *National Research Council*

[12]Measuring Heat Transmission in Building Structures and a Heat Transmission Meter, by P. Nicholls (A.S H V E TRANSACTIONS, Vol 30, 1924)

If the hot-box or hot-plate methods are used, tests are usually run under still air conditions, which means there is no wind movement over the surfaces of the wall during the test. In the hot-plate method of test the inside surface coefficient is eliminated by the plate's being in direct contact with the wall. In practice, some wind movement over the exterior surface of the wall should always be allowed for; hence, still-air coefficients cannot be used over the outside of the building during the heating season. Moreover, still-air transmission coefficients cannot be corrected to provide for moving-air conditions by applying a single constant factor. Computed coefficients of transmission for various types of construction are given in Chapter 5.

EUPATHEOSCOPE

The eupatheoscope affords a means of evaluating the combined effect of radiation and convection in a given environment in terms of a standard environment and in some terms related to human comfort. See Chapter 38.

PROBLEMS IN PRACTICE

1 ● What is the corrected barometric pressure of the atmosphere at 32 F when a mercurial barometer reading of 29.51 in. Hg, is determined in a room having a temperature of 91 F?

Substitute in Equation 1 $h = 29.51\,[1 - 0.000101\,(91 - 32)]$.

$h = 29.33$ in Hg

2 ● What advantages other than its sensitiveness, has the U-tube draft gage or manometer for measurement of low pressures?

Inherent accuracy without calibration and low cost of the essential parts, which are glass tubing and an ordinary scale

3 ● Are thermocouples as accurate as mercury thermometers?

Within the range which can be measured with both instruments (below 1000 F) either one may be made as sensitive as the service requires The accuracy of a thermocouple temperature measurement depends chiefly on (1) an accurate calibration of the wire, (2) the sensitiveness of the electrical instrument, (3) accurate cold-junction control, and (4) proper placement of the sensitive junction

4 ● When an anemometer is used for measuring the air discharged from a grille or register, does it read the velocity through the gross face area or the velocity through the net free area?

Neither. If either of these velocities is required, it should be calculated by means of Equation 2.

5 ● Do common errors made in humidity determination produce a result that is too high or too low?

A higher relative humidity than the true value is likely to be found, either because there is insufficient velocity over the wet-bulb or because the reading is not taken at the right time.

6 ● What is the purpose of the carbon dioxide determination?

It is an index of the adequacy of fresh air supply and also an indicator of air distribution

Chapter 44

TERMINOLOGY

Glossary of Physical and Heating, Ventilating and Air Conditioning Terms Used in the Text, Standard Abbreviations, Conversion Equations, Drafting Symbols, A.S.H.V.E. Codes

Absolute Humidity: See *Humidity.*

Absolute Pressure: The sum, at any particular time, of the gage pressure and the atmospheric pressure.

Absolute Temperature: The temperature of a substance measured above *absolute zero.*

Absolute Zero: The temperature (−459.6 F) at which the molecular motion of a substance theoretically ceases. This is the temperature at which the substance theoretically contains no heat energy.

Acceleration: The rate of change of velocity. In the fps system this is expressed in units of one foot per second per second.

$$a = \frac{V}{t}$$

Acceleration Due to Gravity: The rate of gain in velocity of a freely falling body. In the fps system this is 32.174 ft per second per second.

Adiabatic: An adjective pertaining to or designating variations in volume or pressure not accompanied by gain or loss of heat. When a substance undergoes adiabatic expansion, since it does not receive heat from without, the work which it does is at the expense of its internal energy, and therefore its temperature falls; similarly, when it is adiabatically compressed its temperature rises.

Adsorption: The adhesion of the molecules of gases or dissolved substances to the surfaces of solid bodies, resulting in a concentration of the gas or solution at the place of contact.

Air Cleaner: A device designed for the purpose of removing air-borne impurities such as dusts, fumes and smokes. (Air cleaners include air washers and air filters.)

Air Conditioning: The simultaneous control of all or at least the first three of those factors affecting both the physical and chemical conditions of the atmosphere within any structure. These factors include temperature, humidity, motion, distribution, dust, bacteria, odors, toxic gases, and ionization, most of which affect in greater or lesser degree human health or comfort.

Air Infiltration: The inleakage of air through cracks and crevices, and through doors, windows and other openings, caused by wind pressure or temperature difference.

Air Inlet and Air Outlet: Used to designate that point or location in an air handling device or system where air enters or leaves. These terms are misleading and meaningless standing by themselves; must be accompanied by other words to designate location, space, or device which the air is entering or leaving. Thus, *air inlet to a room* may be opening in end of a duct leading from the *air outlet of a fan.*

Air Washer: An enclosure in which air is forced through a spray of water in order to cleanse, humidify, or dehumidify the air.

Anemometer: An instrument for measuring the velocity of moving air.

Atmospheric Pressure: The pressure exerted by the atmosphere in all directions, as indicated by a barometer. *Standard atmospheric pressure* is considered to be 14.7 lb per square inch, which is equivalent to 29.92 in. of mercury.

Baffle: A plate or wall for deflecting gases or fluids.

Blast: This word was formerly used to denote forced air circulation, particularly in connection with central fan systems using steam or hot water as the heating medium. As applied in this sense, the word *blast* is now obsolete.

Boiler: A closed vessel in which steam is generated or in which water is heated.

Boiler Heating Surface: That portion of the surface of the heat-transfer apparatus in contact with the fluid being heated on one side and the gas or refractory being cooled on the other, in which the fluid being heated forms part of the circulating system; this surface shall be measured on the side receiving heat. This includes the boiler, water walls, water screens, and water floor. (*A.S.M.E.* Power Test Codes, Series 1929.)

Boiler Horsepower: The equivalent evaporation of 34.5 lb of water per hour from and at 212 F. This is equal to a heat output of $970.2 \times 34.5 = 33{,}471.9$ Btu per hour.

British Thermal Unit: The *mean* British thermal unit is $\frac{1}{180}$ of the heat required to raise the temperature of 1 lb of water from 32 F to 212 F. It is substantially equal to the quantity of heat required to raise 1 lb of water from 63 F to 64 F. One Btu $= \frac{1}{3415}$ kwhr.

By-pass: A pipe or duct, usually controlled by valve or damper, for short-circuiting fluid flow.

Calorie: The *mean* calorie is $\frac{1}{100}$ of the heat required to raise the temperature of 1 gram of water from *Zero* C to 100 C. It is substantially equal to the quantity of heat required to raise one gram of water from 14.5 C to 15.5 C.

Central Fan System: A mechanical indirect system of heating, ventilating, or air conditioning, in which the air is treated or handled by

equipment located outside the rooms served, usually at a central location, and is conveyed to and from the rooms by means of a fan and a system of distribution ducts. See Chapters 9 and 10.

Chimney Effect: The tendency in a duct or other vertical air passage for air to rise when heated, owing to its decrease in density.

Coefficient of Transmission: The amount of heat (Btu) transmitted *from air to air* in one hour per square foot of the wall, floor, roof or ceiling for a difference in temperature of 1 F *between the air on the inside and that on the outside of the wall, floor, roof or ceiling.*

Column Radiator: A type of direct radiator. This radiator has not been listed by manufacturers since 1926.

Comfort Line: The effective temperature at which the largest percentage of adults feel comfortable.

Comfort Zone *(Average):* The range of effective temperatures over which the majority (50 per cent or more) of adults feel comfortable. *Comfort Zone (Extreme):* The range of effective temperatures over which one or more adults feel comfortable. (See Chapter 3.)

Concealed Radiator: A heating device located within, adjacent to, or exterior to the room being heated but so covered or enclosed or concealed that the heat transfer surface of the device, which may be either a radiator or a convector, does not *see* the room. Such a device transfers its heat to the room largely by convection air currents.

Conductance: The amount of heat (Btu) transmitted from surface to surface in one hour through one square foot of a material or construction, whatever its thickness, when the temperature difference is 1 F between the two surfaces.

Conduction: The transmission of heat through and by means of matter unaccompanied by any obvious motion of the matter.

Conductivity: The amount of heat (Btu) transmitted in one hour through one square foot of a homogeneous material *1 in. thick* for a difference in temperature of 1 F between the two surfaces of the material.

Conductor *(heat):* A material capable of readily conducting heat The opposite of an insulator or insulation.

Constant Relative Humidity Line: Any line on the psychrometric chart representing a series of conditions which may be evaluated by one percentage of relative humidity, there are also *constant* dry-bulb lines, wet-bulb lines, effective temperature lines, vapor pressure lines, and lines showing other physical properties of air mixed with water vapor.

Control: Any manual or automatic device for the regulation of a machine to keep it at normal operation. If automatic, it is considered that the device is motivated by variations in temperature, pressure, time, light, or other influences.

Convection: The transmission of heat by the circulation of a liquid or a gas such as air. Convection may be *natural* or *forced.*

Convector: A heat transfer surface designed to transfer its heat to surrounding air largely or wholly by convection currents. Such a surface may or may not be enclosed or concealed. When concealed and enclosed

the resulting device is sometimes referred to as a concealed radiator. (See also definition of *Radiator*. See also Chapter 30.)

Corrosive: Having the power to wear away or gradually change the texture or substance of a material.

Decibel: The standard unit for noise or sound intensity. One decibel is equal to ten times the logarithm to the base e of the ratio of the sound intensities.

Degree-Day: A unit, based upon temperature difference and time, used in specifying the nominal heating load in winter. For any one day there exist as many degree-days as there are degrees Fahrenheit difference in temperature between the average outside air temperature, taken over a 24-hour period, and a temperature of 65 F.

Dehumidify: To remove water vapor from the atmosphere; to remove water vapor or moisture from any material.

Density: The weight of a unit volume, expressed in pounds per cubic foot. $d = \frac{W}{V}$.

Dew-Point Temperature: The temperature corresponding to saturation (100 per cent relative humidity) for a given moisture content.

Diffuser: A vaned device placed at an air supply opening to direct the air flow.

Direct-Indirect Heating Unit: A heating unit located in the room or space to be heated and partially enclosed, the enclosed portion being used to heat air which enters from outside the room.

Direct Radiator: Same as *Radiator*.

Direct-Return System (*Hot water*)*:* A hot water system in which the water, after it has passed through a heating unit, is returned to the boiler along a direct path so that the total distance traveled by the water is the shortest feasible, and so that there are considerable differences in the lengths of the several circuits composing the system.

Down-Feed One-Pipe Riser (*Steam*)*:* A pipe which carries steam downward to the heating units and into which the condensation from the heating units drains.

Down-Feed System (*Steam*)*:* A steam heating system in which the supply mains are above the level of the heating units which they serve.

Draft Head (*Side Outlet Enclosure*)*:* The height of a gravity convector between the bottom of the heating unit and the bottom of the air outlet opening.

Draft Head (*Top Outlet Enclosure*)*:* The height of a gravity convector between the bottom of the heating unit and the top of the enclosure.

Drip: A pipe, or a steam trap and a pipe, considered as a unit, which conducts condensation from the steam side of a piping system to the water or return side of the system.

Dry Air: Air with which no water vapor is mixed. This term is used comparatively, since in nature there is always some water vapor included in air, and such water vapor, being a gas, is dry.

Dry-Bulb Temperature: The temperature of the air indicated by any type of thermometer not affected by the water vapor content or relative humidity of the air.

Dry Return: A return pipe in a steam heating system which carries both water of condensation and air. The dry return is above the level of the water line in the boiler in a gravity system. See *Wet Return.*

Dust: Solid material in a finely divided state, the particles of which are large and heavy enough to fall with increasing velocity, due to gravity in still air. For instance, particles of fine sand or grit, the average diameter of which is approximately 0.01 centimeter, such as are blown on a windy day, may be called dust.

Dynamic Head or Pressure: The total or impact pressure. This is the sum of the radial pressure and the velocity pressure at the point of measurement.

Effective Temperature: An arbitrary index of the degree of warmth or cold felt by the human body in response to temperature, humidity, and movement of the air. Effective temperature is a composite index which combines the readings of temperature, humidity, and air motion into a single value. The numerical value of the effective temperature scale has been fixed by the temperature of saturated air which induces an identical sensation of warmth.

Enthalpy: Total heat or thermal potential.

Entropy: A ratio, evaluated for practical purposes by dividing the heat content of a unit weight of a substance by its absolute temperature. Useful in examining changes during a heat cycle. Entropy is constant during a reversible adiabatic change of state.

Equivalent Evaporation: The amount of water a boiler would evaporate, in pounds per hour, if it received feed water at 212 F and vaporized it at the same temperature and atmospheric pressure.

Estimated Design Load: The load, stated in Btu per hour or equivalent direct radiation, as estimated by the purchaser for the conditions of inside and outside temperature for which the amount of installed radiation was determined. It is the sum of the heat emission of the radiation to be actually installed plus the allowance for the heat loss of the connecting piping plus the heat requirement for any apparatus requiring heat connected with the system. (A.S.H.V.E. Standard Code for Rating Steam Heating Solid Fuel Hand-Fired Boilers—edition of April 1932.)

Estimated Maximum Load: Construed to mean the load stated in Btu per hour or equivalent direct radiation that has been estimated by the purchaser to be the greatest or maximum load that the boiler will be called upon to carry. (A.S.H.V.E. Standard Code for Rating Steam Heating Solid Fuel Hand-Fired Boilers—edition of April 1932.)

Extended Heating Surface: See *Heating Surface.*

Extended Surface Heating Unit: A heating unit having a relatively large amount of extended surface which may be integral with the core containing the heating medium or assembled over such a core, making good thermal contact by pressure or by being soldered to the core or by both pressure and soldering. An extended surface heating unit is usually placed within an enclosure and therefore functions as a convector.

Fan Furnace System: See *Warm Air Heating System.*

Force: The action on a body which tends to change its relative condition as to rest or motion. $F = \frac{WV}{gt}$.

Fumes: Particles of solid matter resulting from such chemical processes as combustion, explosion, and distillation, ranging from 0 1 to 1.0 micron in size.

Furnace: That part of a boiler or warm air heating plant in which combustion takes place. Also, a firepot.

Furnace Volume (*total*): The total furnace volume for horizontal-return tubular boilers and water-tube boilers is the cubical contents of the furnace between the grate and the first plane of entry into or between tubes. It therefore includes the volume behind the bridge wall as in ordinary horizontal-return tubular boiler settings, unless manifestly ineffective (*i.e.*, no gas flow taking place through it), as in the case of waste-heat boilers with auxiliary coal furnaces, where one part of the furnace is out of action when the other is being used. For Scotch or other internally fired boilers it is the cubical contents of the furnace, flues and combustion chamber, up to the plane of first entry into the tubes. (*A.S.M.E.* Power Test Codes, Series 1929.)

Gage Pressure: Pressure measured from atmospheric pressure as a base. Gage pressure may be indicated by a manometer which has one leg connected to the pressure source and the other exposed to atmospheric pressure.

Grate Area: The area of the grate surface, measured in square feet, to be used in estimating the rate of burning fuel. This area is construed to mean the area measured in the plane of the top surface of the grate, except that with special furnaces, such as those having magazine feed, or special shapes, the grate area shall be the mean area of the active part of the fuel bed taken perpendicular to the path of the gases through it. For furnaces having a secondary grate, such as those in double-grate down-draft boilers, the effective area shall be taken as the area of the upper grate plus one-eighth of the area of the lower grate, both areas being estimated as defined above. (A.S.H.V.E. Standard and Short Form Heat Balance Codes for Testing Low-Pressure Steam Heating Solid Fuel Boilers.)

Gravity Warm Air Heating System: See *Warm Air Heating System.*

Grille: A perforated covering for an air inlet or outlet usually made of wire screen, pressed steel, cast-iron or plaster. Grilles may be plain or ornamental.

Heat: A form of energy generated by the transformation of some other form of energy, as by combustion, chemical action, or friction. According to the molecular theory, heat consists of the kinetic and potential energy of the molecules of a substance. The addition of heat energy to a body increases the temperature or the kinetic energy of motion of its molecules (*sensible heat*) or increases their potential energy of position but does not increase the temperature, as when melting or boiling occurs (*latent heat*).

Heat Capacity: The amount of heat (Btu or calories) required to raise the temperature of a body of any mass and variety of parts one degree (Fahrenheit or centigrade). This will depend on the masses and specific heats of the various parts of the body.

Therefore

$$S = m_1 s_1 + m_2 s_2 + m_3 s_3 \ldots \text{etc.}$$

where

S is the heat capacity and m_1, m_2, m_3, and s_1, s_2, s_3 stand for the masses and corresponding specific heats of the parts, respectively

Heating Medium: A substance such as water, steam, air, electricity or furnace gas used to convey heat from the boiler, furnace or other source of heat or energy to the heating unit from which the heat is dissipated.

Heating Surface: The exterior surface of a heating unit. *Extended heating surface (or extended surface):* Heating surface having air on both sides and heated by conduction from the prime surface. *Prime Surface:* Heating surface having the heating medium on one side and air (or extended surface) on the other. (See also *Boiler Heating Surface.*)

Heat of the Liquid: The sensible heat of a mass of liquid above an arbitrary zero.

Horsepower: A unit to indicate the time rate of doing work equal to 550 ft-lb per second or 33,000 ft-lb per minute. (One horsepower = 745.8 watts. In practice this is considered 746 watts.)

Hot Water Heating System: A heating system in which water is used as the medium by which heat is carried through pipes from the boiler to the heating units.

Humidify: To add water vapor to the atmosphere; to add water vapor or moisture to any material.

Humidity: The water vapor mixed with dry air in the atmosphere. *Absolute humidity* refers to the weight of water vapor per unit volume of space occupied, expressed in grains or pounds per cubic foot. *Specific humidity* refers to the weight of water vapor in pounds carried by one pound of dry air. *Relative humidity* is a ratio, usually expressed in per cent, used to indicate the degree of saturation existing in any given space resulting from the water vapor present in that space. Relative humidity is either the ratio of the actual partial pressure of the water vapor in the air to the saturation pressure at the dry-bulb temperature, or the ratio of the actual density of the vapor to the density of saturated vapor at the dry-bulb temperature. The presence of air or other gases in the same space at the same time has nothing to do with the relative humidity of the space.

Humidistat: A regulatory device, actuated by changes in humidity, used for the control of humidity.

Hygrostat: Same as *Humidistat.*

Inch of Water: A measure of pressure which refers to the difference in the heights of the legs of a water filled manometer.

Insulation *(heat):* A material having a relatively high heat-resistance per unit of thickness.

Isobaric: An adjective used to indicate a change taking place at constant pressure.

Isothermal: An adjective used to indicate a change taking place at constant temperature.

Latent Heat: See *Heat.*

Laws of Thermodynamics: The *first law* states that the total energy of an isolated system remains constant and cannot be increased or diminished by any physical process whatever. The *second law* states that no change in a system of bodies that takes place of itself can increase the available energy of a system.

Manometer: An instrument for measuring pressures, essentially a U-tube partially filled with a liquid, usually water, mercury, or a light oil, so the amount of displacement of the liquid indicates the pressure being exerted on the instrument.

Mass: The quantity of matter, in pounds, to which the unit of force (one pound) will give an acceleration of one foot per second per second.

$$m = \frac{W}{g}.$$

Mb, Mbh[1]**:** Symbols which represent, respectively, 1000 Btu and 1000 Btu per hour

Mechanical Equivalent of Heat: The mechanical energy necessary to produce 1 Btu of heat energy. J = 777.5 ft-lb.

Micron: A unit of length, the thousandth part of one millimeter or the millionth of a meter.

Mol: The unit of weight for gases. It is defined as m lb where m denotes the molecular weight of a gas. For any gas the volume of 1 *mol* at 32 F and standard atmospheric pressure is 358.65 cu ft and the weight of a cubic foot is 0.002788 m lb.

Neutral Zone: The level within a room or building at which the pressure is exactly equal to the outside barometric pressure.

One-Pipe Supply Riser (*steam*)*:* A pipe which carries steam upward to a heating unit and which also carries the condensation from the heating unit in a direction opposite to the steam flow.

One-Pipe System (*hot water*)*:* A hot water system in which the water flows through more than one heating unit before it returns to the boiler; consequently, the heating units farthest from the boiler are supplied with cooler water than those near the boiler in the same circuit.

One-Pipe System (*steam*)*:* A steam heating system consisting of a main circuit in which the steam and condensate flow in the same pipe, usually in opposite directions. Ordinarily to each heating unit there is but one connection which must serve as both the supply and the return, although separate supply and return connections may be used.

Overhead System: Any steam or hot water system in which the supply main is above the heating units. With a steam system the return must be below the heating units; with a water system, the return *may* be above the heating units.

Panel Radiator: A heating unit placed on or flush with a flat wall surface and intended to function essentially as a radiator.

---

[1]*These symbols were approved by the* A S H V E, June, 1933.

Panel Warming: A method of heating involving the installation of the heating units (pipe coils) within the wall, floor or ceiling of the room, so that the heating process takes place mainly by radiation from the wall, floor or ceiling surfaces to the objects in the room.

Plenum Chamber: An air compartment maintained under pressure and connected to one or more distributing ducts.

Potentiometer: An instrument for measuring or comparing small electromotive forces.

Power: The rate of performing work, expressed in units of horsepower, one of which is equal to 550 ft-lb of work per second, or 33,000 ft-lb per minute.

Prime Surface: See *Heating Surface.*

Psychrometer: An instrument for ascertaining the humidity or hygrometric state of the atmosphere. *Psychrometric:* Pertaining to psychrometry or the state of the atmosphere as to moisture. *Psychrometry:* The branch of physics that treats of the measurement of degree of moisture, especially the moisture mixed with the air.

Pyrometer: An instrument for measuring high temperatures.

Radiation: The transmission of heat through space by wave motion.

Radiator: A heating unit exposed to view within the room or space to be heated. A radiator transfers heat by radiation to objects "it can see" and by conduction to the surrounding air which in turn is circulated by natural convection; a so-called *radiator* is also a *convector* but the single term *radiator* has been established by long usage.

Recessed Radiator: A heating unit set back into a wall recess but not enclosed.

Refrigerant: A substance which produces a refrigerating effect by its absorption of heat while expanding or vaporizing.

Register: A grille with a built-in multiblade damper or shutter.

Relative Humidity: See *Humidity:* see also discussion of relative humidity in Chapter 1.

Return Mains: The pipes which return the heating medium from the heating units to the source of heat supply.

Reversed-Return System (*hot water*)*:* A hot water heating system in which the water from several heating units is returned along paths arranged so that all circuits composing the system or composing a major subdivision of the system are practically of equal length.

Roof Ventilator: A device placed on the roof of a building to permit egress of air.

Saturated Air: Air containing as much water vapor as it can hold without any condensing out; in saturated air, the partial pressure of the water vapor is equal to the vapor pressure of water at the existing temperature.

Sensible Heat: See *Heat.*

Smoke: Carbon or soot particles less than 0.1 micron in size which result from the incomplete combustion of carbonaceous materials such as coal, oil, tar, and tobacco.

Smokeless Arch: An inverted baffle placed in an up-draft furnace toward the rear to aid in mixing the gases of combustion and thereby to reduce the smoke produced.

Specific Gravity: The ratio of the weight of a body to the weight of an equal volume of water at some standard temperature, usually 39.2 F.

Specific Heat: The quantity of heat, expressed in Btu, required to raise the temperature of 1 lb of a substance 1 F.

Specific Volume: The volume, expressed in cubic feet, of one pound of a substance. $v = \frac{1}{d} = \frac{V}{W}$.

Split System: A system in which the heating and ventilating are accomplished by means of radiators or convectors supplemented by mechanical circulation of air (heated or unheated) from a central point.

Square Foot of Heating Surface (*equivalent*): Equivalent direct *radiation* (EDR). By definition, that amount of heating surface which will give off 240 Btu per hour. The *equivalent* square feet of heating surface may have no direct relation to the actual surface area.

Stack Height: The height of a gravity convector between the bottom of the heating unit and the top of the outlet opening.

Standard Air: As defined by A.S.H.V.E. codes, *standard air* is air weighing 0.07488 lb per cubic foot, which is air at 68 F dry-bulb and 50 per cent relative humidity with a barometric pressure of 29.92 in. of mercury. (Most engineering tables and formulae involving the weight of air are based on air weighing 0.07492 lb per cubic foot, which is dry air at 70 F dry-bulb with a barometric pressure of 29.921 in. of mercury. The error involved in disregarding the difference between the above two weights is very slight and in most instances may be neglected.)

Static Pressure: The compressive pressure existing in a fluid. It is a measure of the potential energy of the fluid.

Steam: Steam is water vapor which exists in the vaporous condition because sufficient heat has been added to the water to supply the latent heat of evaporation and change the liquid into vapor Steam in contact with the water from which it has been generated may be *dry saturated* steam or *wet saturated* steam. The latter contains more or less actual water in the form of mist. If steam is heated, and the pressure maintained the same as when it was vaporized, its temperature will increase and it will become *superheated*.

Steam Heating System: A heating system in which heat is transferred from the boiler or other source of steam to the heating units by means of steam at, above, or below atmospheric pressure.

Steam Trap: A device for allowing the passage of condensate and preventing the passage of steam, or for allowing the passage of air as well as condensate.

Superheated Steam: See *Steam*.

Supply Mains (*steam*): The pipes through which the steam flows from the boiler or source of supply to the run-outs and risers leading to the heating units.

Surface Conductance: The amount of heat (Btu) transmitted by radiation, conduction, and convection *from a surface to the air or liquid surrounding it*, or vice versa, in one hour per square foot of the surface for a difference in temperature of 1 deg between the surface and the surrounding air or liquid

Synthetic Air Chart: A chart for evaluating the air conditions maintained in a room.

Therm: Symbol used in the gas industry representing 100,000 Btu.

Thermal Resistance: The reciprocal of *conductance*.

Thermal Resistivity: The reciprocal of *conductivity*.

Thermodynamics: The science which treats of the mechanical actions or relations of heat.

Thermostat: An instrument which responds to changes in temperature and which directly or indirectly controls the source of heat supply.

Ton of Refrigeration: The extraction of 12,000 Btu per hour.

Ton Day of Refrigeration: The heat removed by a ton of refrigeration operating for one day; 288,000 Btu.

Total Heat: A thermodynamic quantity, variously called heat content, thermal potential, enthalpy. It is the heat required per unit mass (Btu per pound) to raise a given substance to a given point from an arbitrary datum point. It is the sum of the heat of the liquid, the latent heat, and any miscellaneous heat which may be present.

Total Pressure: The sum of the static and velocity pressures in a fluid. It is a measure of the total energy of the fluid.

Tube (or Tubular) Radiator: A cast-iron heating unit used as a radiator and having small vertical tubes.

Two-Pipe System (*steam or water*)*:* A heating system in which one pipe is used for the supply of the heating medium to the heating unit and another for the return of the heating medium to the source of heat supply. The essential feature of a two-pipe system is that each heating unit receives a direct supply of the heating medium which medium cannot have served a preceding heating unit.

Underfeed Distribution System (*hot water*)*:* A hot water heating system in which the main flow pipe is below the heating units.

Underfeed Stoker: A stoker which feeds the coal underneath the fuel bed.

Unit: As applied to heating, ventilating and air conditioning equipment this word means a factory-built and assembled equipment with apparatus for accomplishing some specified function or combination of functions.

It is loosely applied to a great variety of equipment. Usually the function is included in the name, and hence come terms like Unit Heater, Unit Ventilator, Humidifying Unit, and Air Conditioning Unit.

Units are said to be *direct*, or *room*, when intended for location, or located in, the treated space; *indirect*, or *remote*, when outside or adjacent to the treated space. They are *ceiling* units when suspended from above,

and *floor* when supported from below. Other descriptive words include *free delivery* when the unit is not intended to be attached to ducts or similar resistance-producing devices, and *pressure* when for use with such ducts. Complete description requires the use of several of these qualifying words or phrases. See Chapter 13.

Up-Feed System (*steam*): A steam heating system in which the supply mains are below the level of the heating units which they serve.

Vacuum Heating System: A two-pipe steam heating system equipped with the necessary accessory apparatus which will permit operating the system below atmospheric pressure when desired.

Vapor: Any substance in the gaseous state.

Vapor Heating System: A steam heating system which operates under pressures at or near atmospheric and which returns the condensation to the boiler or receiver by gravity. Vapor systems have thermostatic traps or other means of resistance on the return ends of the heating units for preventing steam from entering the return mains; they also have a pressure-equalizing and air-eliminating device at the end of the dry return. *Direct Vent Vapor System:* A vapor heating system with air valves which do not permit re-entry of air.

Vapor Pressure: The equilibrium pressure exerted by a vapor in contact with its liquid.

Velocity: The time rate of motion of a body in a fixed direction. In the fps system it is expressed in units of one foot per second. $V = \frac{s}{t}$.

Velocity Pressure: The pressure corresponding to the velocity of flow. It is a measure of the kinetic energy of the fluid.

Ventilation: The process of supplying or removing air by natural or mechanical means, to or from any space. Such air may or may not have been conditioned. (See *Air Conditioning*.)

Warm Air Heating System: A warm air heating plant consists of a heating unit (fuel-burning furnace) enclosed in a casing, from which the heated air is distributed to the various rooms of the building through ducts. If the motive head producing flow depends on the difference in weight between the heated air leaving the casing and the cooler air entering the bottom of the casing, it is termed a *gravity* system. A booster fan may, however, be used in conjunction with a gravity-designed system. If a fan is used to produce circulation and the system is designed especially for fan circulation, it is termed a *fan furnace* system or a *central fan furnace* system. A fan furnace system may include air washers and filters.

Wet-Bulb Temperature: The lowest temperature which a water wetted body will attain when exposed to an air current. This is the temperature of adiabatic saturation.

Wet Return: That part of a return main of a steam heating system which is filled with water of condensation. The wet return usually is below the level of the water line in the boiler, although not necessarily so. See *Dry Return*.

ABBREVIATIONS [2]

Absolute	abs
Acceleration, due to gravity	*g*
Acceleration, linear	*a*
Air horsepower	air hp
Alternating-current (as adjective)	a-c
Ampere	amp
Ampere-hour	amp-hr
Area	*A*
Atmosphere	atm
Average	avg
Avoirdupois	avdp
Barometer	bar.
Boiler pressure	bp
Boiling point	bp
Brake horsepower	bhp
Brake horsepower-hour	bhp-hr
British thermal unit	Btu
Calorie	cal
Centigram	cg
Centimeter	cm
Centimeter-gram-second (system)	cgs
Change in specific volume during vaporization	v_{fg}
Cubic	cu
Cubic foot	cu ft
Cubic feet per minute	cfm
Cubic feet per second	cfs
Decibel	db
Degree[3]	deg or °
Degree centigrade	C
Degree Fahrenheit	F
Degree Kelvin	K
Degree Réaumur	R
Density, Weight per unit volume, Specific weight	*d* or ρ (rho)

$$d = \frac{1}{v}$$

Diameter	*D* or diam
Direct-current (as adjective)	d-c
Distance, linear	*s*
Dry saturated vapor, Dry saturated gas at saturation pressure and temperature, Vapor in contact with liquid	*Subscript* $_g$
Entropy (The capital should be used for any weight, and the small letter for unit weight.)	*S* or *s*
Feet per minute	fpm
Feet per second	fps
Foot	ft
Foot-pound	ft-lb
Foot-pound-second (system)	fps
Force, total load	*F*
Freezing point	fp
Gallon	gal
Gallons per minute	gpm
Gallons per second	gps
Gram	g
Gram-calorie	g-cal

[2] From compilations of abbreviations approved by the *American Standards Association*, Z, 10 a, c, f, and i. As a general rule the period is omitted in all abbreviations except where the omission results in the formation of an English word

[3] It is recommended that the abbreviation for the temperature scale, F, C, K, be included in expressions for numerical temperatures but, wherever feasible, the abbreviation for *degree* be omitted; as 68 F.

Head .... H or h
Heat content, Total heat, Enthalpy. (The capital should be used for any weight and the small letter for unit weight.) .... H or h
Heat content of saturated liquid, Total heat of saturated liquid, Enthalpy of saturated liquid, sometimes called heat of the liquid .... h_f
Heat content of dry saturated vapor, Total heat of dry saturated vapor, Enthalpy of dry saturated vapor .... h_g
Heat of vaporization at constant pressure .... L or h_{fg}
Horsepower .... hp
Horsepower-hour .... hp-hr
Inch .... in
Inch-pound .... in.-lb
Indicated horsepower .... ihp
Indicated horsepower-hour .... ihp-hr
Internal energy, Intrinsic energy. (The capital should be used for any weight and the small letter for unit weight.) .... U or u
Kilogram .... kg
Kilowatt .... kw
Kilowatthour .... kwhr
Length of path of heat flow, thickness L
Load, total .... W
Mass .... m
Mechanical efficiency .... e_m
Mechanical equivalent of heat .... J
Melting point .... mp
Meter .... m
Micron .... μ (mu)
Miles per hour .... mph
Minute .... min
Molecular weight .... mol wt
Mol .... mol
Ounce .... oz
Power, Horsepower, Work per unit time .... P
Pressure, Absolute pressure, Gage pressure, Force per unit area .... p
Quantity (total) of fluid, water, gas, heat, Quantity by volume; Total quantity of heat transferred .... Q
Quality of steam, Pounds of dry steam per pound of mixture x
Revolutions per minute .... rpm
Saturated liquid at saturation pressure and temperature, Liquid in contact with vapor .... *Subscript* $_f$
Specific gravity .... sp gr
Specific heat .... sp ht or c
Specific heat at constant pressure .... c_p
Specific heat at constant volume .... c_v
Specific volume, Volume per unit weight, Volume per unit mass v
Square foot .... sq ft
Square inch .... sq in.
Temperature (ordinary) F or C (*Theta* is used preferably only when t is used for Time in the same discussion.) .... t or θ (*theta*)
Temperature (absolute) F abs or K. (Capital *theta* is used preferably only when small *theta* is used for ordinary temperature.) .... T or Θ (*capital theta*)
Thermal conductance[4] (heat transferred per unit time per degree) C

$$C = \frac{1}{R} = \frac{kA}{L} = \frac{q}{t_1 - t_2}$$

Thermal conductance per unit area, Unit conductance (heat transferred per unit time per unit area per degree) .... C_a

$$C_a = \frac{C}{A} = \frac{1}{RA} = \frac{q}{A(t_1 - t_2)} = \frac{k}{L}$$

[4]Terms ending *ivity* designate properties independent of size or shape, sometimes called *specific properties*. Examples are—conductivity and resistivity. Terms ending *ance* designate quantities depending not only on the material, but also upon size and shape, sometimes called *total quantities*. Examples are—conductance and transmittance. Terms ending *ion* designate rate of heat transfer. Examples are—conduction and transmission.

Thermal conductivity (heat transferred per unit time per unit area, and per degree per unit length) .... k

$$k = \frac{\frac{q}{A}}{\frac{(t_1 - t_2)}{L}}$$

Surface coefficient of heat transfer, Film coefficient of heat transfer, Individual coefficient of heat transfer (heat transferred per unit time per unit area per degree) .... f

$$f = \frac{\frac{q}{A}}{t_1 - t_2}$$

(In general f is not equal to k/L, where L is the actual thickness of the fluid film.)

Over-all coefficient of heat transfer, Thermal transmittance per unit area (heat transferred per unit time per unit area per degree over-all) .... U

$$U = \frac{\frac{q}{A}}{t_1 - t_2}$$

Thermal transmission (heat transferred per unit time) .... q

$$q = \frac{Q}{t}$$

Thermal resistance (degrees per unit of heat transferred per unit time) .... R

$$R = \frac{t_1 - t_2}{q} = \frac{L}{kA}$$

Thermal resistivity .... $1/k$
Vaporization values at constant pressure, Differences between values for saturated vapor and saturated liquid at the same pressure .... *Subscript* $_{fg}$
Velocity V
Volume (total) V
Volume per unit time, Rate at which quantity of material passes through a machine, Quantity of heat per unit time, Quantity of heat per unit weight .... q
Watt .... w
Watthour .... whr
Weight of a major item, Total weight .... W
Weight rate, Weight per unit of power, Weight per unit of time .... w
Work (total) .... W

CONVERSION EQUATIONS

Fahrenheit degrees = 9/5 (centigrade degrees + 32).

Centigrade degrees = 5/9 (Fahrenheit degrees − 32).

Absolute temperature, expressed in Fahrenheit degrees = Fahrenheit degrees + 459 6 In heating and ventilating work, 460 is usually used.

Absolute temperature, expressed in centigrade degrees = centigrade degrees + 273.1

Power, Heat, and Work

1 ton refrigeration	= 12,000 Btu per hour 200 Btu per minute
Latent heat of ice	= 143 33 Btu per pound
1 Btu	= 777 5 ft-lb 0 293 watthours 252 02 mean calories
1 watthour	= 2,655 2 ft-lb 3 415 Btu 3600 joules 860.648 mean calories
1 mean calorie	= 0.003968 Btu 3 085 ft-lb 0 0011619 watthours
1 kilowatt (1000 watts)	= 1.3405 horsepower 56 92 Btu per minute 44,252 7 ft-lb per minute
1 horsepower	= 0.746 kilowatt 42 44 Btu per minute 33,000 ft-lb per minute 550 ft-lb per second
1 boiler horsepower	= 33,471.9 Btu per hour

Weight and Volume

1 gal (U. S.)	= 231 cu in. 0 13368 cu ft
1 British or Imperial gallon	= 277 274 cu in.
1 cu ft	= 7.4805 gal 1728 cu in
1 cu ft water at 60 F	= 62 37 lb
1 cu ft water at 212 F	= 59.76 lb
1 gal water at 60 F	= 8.34 lb
1 gal water at 212 F	= 7.99 lb
1 lb (avdp)	= 16 oz 7000 grains
1 bushel	= 1.244 cu ft
1 short ton	= 2000 lb
1 long ton	= 2240 lb

Pressure

1 lb per square inch	= 144 lb per square foot 2 0416 in. mercury at 62 F 2.309 ft water at 62 F 27.71 in. water at 62 F
1 oz per square inch	= 0.1276 in. mercury at 62 F 1 732 in. water at 62 F
1 atmosphere	= 14.7 lb per square inch 2116.3 lb per square foot 33.974 ft water at 62 F 30 in. mercury at 62 F 29.921 in. mercury at 32 F

1 in. water at 62 F	= 0.03609 lb per square inch 0.5774 oz per square inch 5.196 lb per square foot
1 ft water at 62 F	= 0 433 lb per square inch 62 355 lb per square foot
1 in. mercury at 62 F	= 0 491 lb per square inch 7.86 oz per square inch 1 131 ft water at 62 F 13.57 in water at 62 F

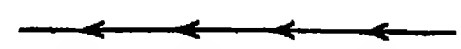
Metric Units

1 cm	= 0.3937 in
1 in.	= 2.54 cm
1 m	= 3.281 ft
1 ft	= 0 3048 m
1 sq cm	= 0.155 sq in.
1 sq in	= 6.45 sq cm
1 sq m	= 10 765 sq ft
1 sq ft	= 0.0929 sq m
1 cu cm	= 0.061 cu in.
1 cu in.	= 16 39 cu cm
1 cu m	= 35 32 cu ft
1 cu ft	= 0.0283 cu m
1 liter	= 1000 cu cm = 0 264 gal
1 kg	= 2 2046 lb
1 lb	= 0 4536 kg
1 metric ton	= 2205 lb (avdp)
1 gram	= 980.59 dynes = 0.002205 lb
1 kilometer per hour	= 0 6214 mph
1 gram per square centimeter	= 0 0290 in. mercury, at 0 deg C 0.394 in. water, at 15 C
1 kg per square centimeter (metric atmosphere)	= 14.22 lb per square inch
1 gram per cubic centimeter	= 0.03614 lb per cubic inch 62.43 lb per cubic foot
1 dyne	= 0 00007233 poundals
1 joule	= 10,000,000 ergs 0.73767 ft-lb
1 metric horsepower	= 75 kg-m per second 0.986 hp (U. S.)
1 kilogram-calorie (large calorie)	= 1000 gram-calories (small calorie) 3.97 Btu
1 kilogram-calorie per kilogram	= 1 8 Btu per pound
1 gram-calorie per square centimeter	= 3.687 Btu per square foot
1 gram-calorie per square centimeter per centimeter	= 1.451 Btu per square foot per inch
1 gram-calorie per second per square centimeter for a temperature graduation of 1 deg C per centimeter	= 2903 Btu per hour per square foot for a temperature graduation of 1 deg F per inch of thickness.

SYMBOLS FOR HEATING, VENTILATING AND AIR CONDITIONING DRAWINGS[5]

1. The objects of this standard set of symbols are to insure the correct interpretation of drawings and to conserve drafting room time by establishing simple and unmistakable symbols for the component parts of the heating and ventilating systems In preparing the list of symbols an effort has been made to follow existing practice in so far as possible but the list cannot be expected to match exactly the existing practice of every drafting room

1. General Piping
2. Steam Piping
3. Condensate Piping
4. Cold Water Piping
5. Hot Water Piping

6 Air Piping 

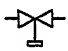
7 Vacuum Piping

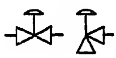
8 Gas Piping

9 Refrigerant Piping 

10. Oil Piping

11. Lock and Shield Valve 

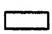
12 Reducing Valve 

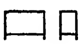
13. Diaphragm Valve 

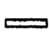
14. Thermostat 

15 Radiator Trap Elevation 

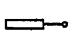
16 Radiator Trap Plan 

17. Tube Radiator Plan 

18 Tube Radiator Elevation 

19 Wall Radiator Plan 

20. Wall Radiator Elevation 

21 Pipe Coil Plan 

22 Pipe Coil Elevation 

23 Indirect Radiator Plan 

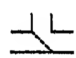
24 Indirect Radiator Elevation 

25 Supply Duct, Section 

26 Exhaust Duct, Section 

27 Butterfly Damper Plan (or Elevation) 

28 Butterfly Damper Elevation (or Plan) 

29. Deflecting Damper Rectangular Pipe 

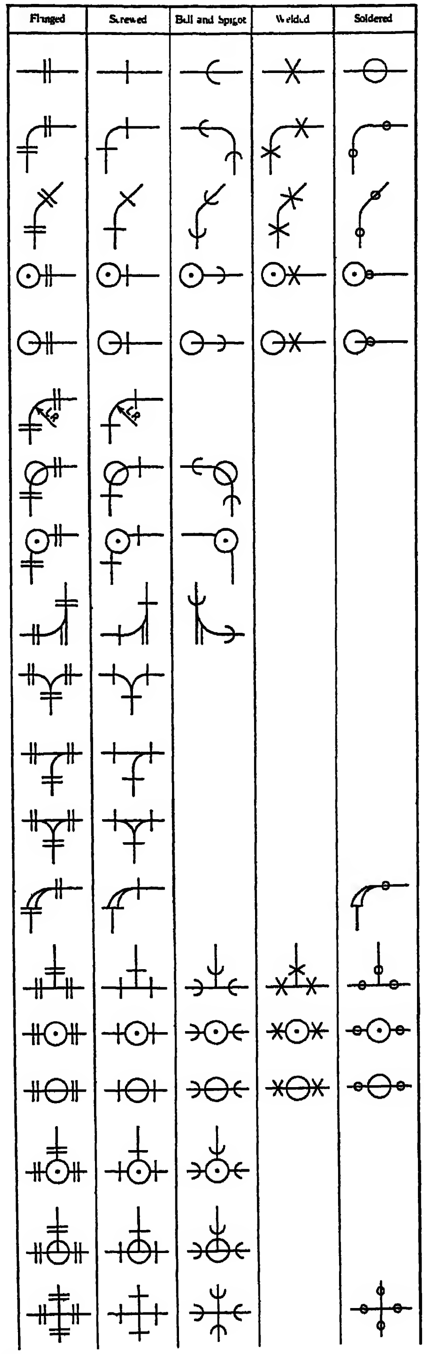
30. Vanes 

31. Air Supply Outlet 

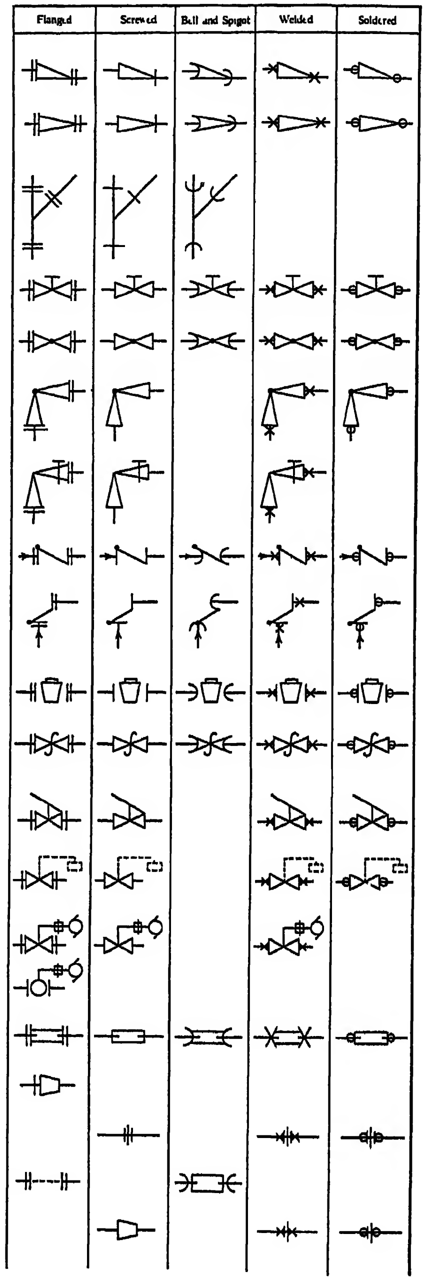
32. Exhaust Inlet 

[5]From A S H V.E. Code of Minimum Requirements for the Heating and Ventilation of Buildings, edition of 1929, and American Standard Drawings and Drafting Room Practice Graphical Symbols (*American Standards Association*, Z14 2—1935)

33. Joint
34 Elbow—90 deg
35 Elbow—45 deg
36. Elbow—Turned Up
37 Elbow—Turned Down
38. Elbow—Long Radius
39 Side Outlet Elbow—Outlet Down
40 Side Outlet Elbow—Outlet Up
41. Base Elbow
42. Double Branch Elbow
43 Single Sweep Tee
44. Double Sweep Tee
45 Reducing Elbow
46 Tee
47 Tee—Outlet Up
48. Tee—Outlet Down
49. Side Outlet Tee—Outlet Up
50. Side Outlet Tee—Outlet Down
51 Cross



52. Eccentric Reducer
53 Reducer
54 Lateral
55 Gate Valve
56 Globe Valve
57 Angle Globe Valve
58 Angle Gate Valve
59 Check Valve
60 Angle Check Valve
61. Stop Cock
62 Safety Valve
63 Quick Opening Valve
64 Float Operating Valve
65. Motor Operated Gate Valve
66 Motor Operated Globe Valve
67 Expansion Joint Flanged
68 Reducing Flange
69. Union
70. Sleeve
71. Bushing



A.S.H.V.E. CODES

The following codes and standards relating to the design, installation, testing, rating, and maintenance of materials and equipment used for the heating, ventilation and air conditioning of buildings, have been adopted by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

SUBJECT	TITLE	WHEN ADOPTED	REFERENCE
Air Cleaning Devices	A.S H.V.E Standard Code for Testing and Rating Air Cleaning Devices Used in General Ventilation Work[a]	June, 1933	A S H V E TRANSACTIONS, Vol. 39, 1933
Air Purity	Synthetic Air Chart	June, 1917	A.S.H.V.E. TRANSACTIONS, Vol 23, p. 607, and THE GUIDE, 1931
Boilers (testing)	Standard and Short-Form Heat Balance Codes for Testing Low Pressure Steam Heating Solid Fuel Boilers (Codes 1 and 2)[a]	June, 1929	A S H.V E. TRANSACTIONS, Vol. 35, 1929
Boilers (testing)	A.S.H.V.E. Performance Test Code for Steam Heating Solid Fuel Boilers (Code 3)[a b]	June, 1929	A.S H.V.E. TRANSACTIONS, Vol. 35, 1929
Boilers—Oil Fuel (testing)	A.S.H.V E Standard Code for Testing Steam Heating Boilers Burning Oil Fuel[a]	June, 1932	A S H.V.E. TRANSACTIONS, Vol. 37, 1931
Boilers (rating)	A S.H.V E. Standard Code for Rating Steam Heating Solid Fuel Hand Fired Boilers[a]	January, 1929 Revised April, 1930	A.S H.V.E. TRANSACTIONS, Vol. 36, 1930, p 42
Concealed Gravity Type Radiation	A S.H.V.E. Standard Code for Testing and Rating Concealed Gravity Type Radiation (Hot Water Section)[a]	June, 1933	A.S H V.E TRANSACTIONS, Vol. 39, 1933
Convectors	A S H V.E. Standard Code for Testing and Rating Concealed Gravity Type Radiation (Steam Code)[a]	January, 1931	A.S.H.V E. TRANSACTIONS, Vol. 37, 1931, p 367
Ethics	Code of Ethics for Engineers	January, 1922	A.S.H V.E. TRANSACTIONS, Vol. 28, 1922, p. 6 (See frontispiece THE GUIDE, 1937)
Fans	Standard Test Code for Disc and Propeller Fans, Centrifugal Fans and Blowers[a]	May, 1923. Revised June, 1931	A S.H.V.E. TRANSACTIONS, Vol. 29, 1923, p. 407[c]
Garages	Code for Heating and Ventilating Garages	June, 1929 Revised January, 1935	A.S.H.V.E. TRANSACTIONS, Vol. 35, 1929, p 355 A S.H.V.E Reprint

[a]Reprints available
[b]Originally adopted by the *National Boiler and Radiator Manufacturers Association.*
[c]Also, see *Heating, Piping and Air Conditioning*, August, 1931, p 713

SUBJECT	TITLE	WHEN ADOPTED	REFERENCE
Heat Transmission Through Walls	Standard Test Code for Heat Transmission through Walls[a]	January, 1927	A.S.H.V.E. TRANSACTIONS, Vol. 34, 1928, p. 253
Minimum Requirements	Code of Minimum Requirements for Heating and Ventilation of Buildings, Edition–1929	June, 1925	A.S.H.V.E. Codes
Pitot Tube	Code for Use of Pitot Tube	January, 1914	A.S.H.V.E. TRANSACTIONS, Vol. 20, 1914, p. 211
Radiators	Code for Testing Radiators[a]	January, 1927	A.S.H.V.E. TRANSACTIONS, Vol. 33, 1927, p. 18
Unit Heaters	Standard Code for Testing and Rating Steam Unit Heaters[a d]	January, 1930	A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930, p. 165
Unit Ventilators	A.S.H.V.E. Standard Code for Testing and Rating Steam Unit Ventilators[a]	June, 1932	A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932, p. 25
Vacuum Heating Pumps	A.S.H.V.E. Standard Code for Testing and Rating Return Line Low Vacuum Heating Pumps[a]	June, 1934	A.S.H.V.E. TRANSACTIONS, Vol. 40, 1934
Ventilation	Report of Committee on Ventilation Standards[a]	August, 1932	A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932, p. 383

The following Codes and Standards have been endorsed or approved by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS:

SUBJECT	TITLE	SPONSORED BY	REFERENCE
Air Conditioning Equipment[e]	Standard Method of Rating and Testing Air Conditioning Equipment	*American Society of Refrigerating Engineers*	*American Society of Refrigerating Engineers*, New York, N. Y.
Chimneys	Standard Ordinance for Chimney Construction	*National Board of Fire Underwriters*	Chapter 14, THE GUIDE, 1931
Condensing Units[e]	Standard Method of Rating and Testing Mechanical Condensing Units	*American Society of Refrigerating Engineers*	*American Society of Refrigerating Engineers*, New York, N. Y.
Piping Systems	Identification of Piping Systems[f]	*American Society of Mechanical Engineers*	*Heating, Piping and Air Conditioning*, July, 1929
Warm Air Furnaces	Standard Code Regulating the Installation of Gravity Warm Air Furnaces in Residences	*National Warm Air Heating and Air Conditioning Association*	*National Warm Air Heating and Air Conditioning Association*, Columbus, Ohio

[d]Adopted jointly by the *Industrial Unit Heater Association*, and the A.S.H.V.E.
[e]Proposed code prepared by Joint Committee of *American Society of Refrigerating Engineers*, AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, *Refrigerating Machinery Association*, *National Electric Manufacturers Association* and *Air Conditioning Manufacturers Association*
[f]Adopted November, 1928, Sponsored by (1) *American Society of Mechanical Engineers*, (2) *National Safety Council*

CATALOG DATA SECTION

The AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS GUIDE 1937

INDEX TO ADVERTISERS

PAGE 811

INDEX TO MODERN EQUIPMENT

PAGE 1133

INDEX

TO

ADVERTISERS

AMERICAN SOCIETY *of* HEATING *and* VENTILATING ENGINEERS GUIDE, 1937

A

B

C

D

D

E

F

G

H

I

J

K

L

M

N

O

P

R

S

T

U

V

W

Y

Z

American Moistening Company

ESTABLISHED 1888

ATLANTA GA
BOSTON MASS

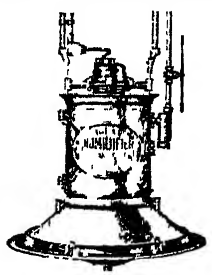
Providence, R. I.

CHARLOTTE, N C
GREENVILLE S C

AMCO

UNIT HUMIDIFYING AND AIR CONDITIONING EQUIPMENT

The Amco line of devices for the supply, maintenance and control of humidity is complete in its ability to meet any presented problem of applied humidification Used independently or as an adjunct to Central Station equipment, these devices automatically maintain any required humidity condition in a capable uniform performance

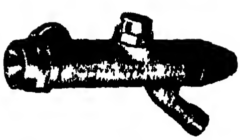


IDEAL HUMIDIFIERS—Senior Type

A high capacity unit for use where conditions require a great amount and good distribution of moisture Motor driven fan gives wide distribution of atomized spray. Amco heads serve the triple purpose of humidifying, air washing and cooling

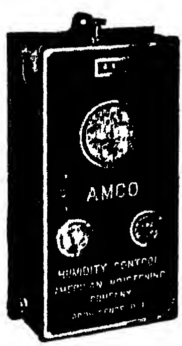
IDEAL HUMIDIFIERS—Junior Type

Similar in construction to Senior Type. Used where medium capacities are required.



AMCO ATOMIZER—No. 4

Quality and quantity of spray are maintained even under adverse conditions because this atomizer is automatically self-cleaning. When the compressed air supply is shut off, either manually or in response to a humidity control, *both* air and water nozzles are thoroughly cleaned



AMCO HUMIDITY CONTROLS

Compressed Air Operated

An extremely accurate and active device operated by compressed air which assures a regulation of humidity within exceedingly close ranges

AMCO HUMIDITY CONTROL

Electrically Operated

Similar in principle to the Compressed Air Type except that the hydroscopic element operates electrical contacts which control the units.

A few of many AMCO products with a Long Record of Dependable Performance

Sectional Humidifiers.
Amtex Humidifiers
Hand Sprayers
Mine Sprays
Fabric and Paper Dampeners.

Mechanical Psychrometers.
Electro Psychrometers
Sling Psychrometers.
Hygrometers

Air Devices Corporation

(Subsidiary of Automatic Products Corporation)

Manufacturers of Air Conditioning, Heating and Ventilating Equipment

Factory and General Sales Offices **64 East 25th Street, Chicago, Ill.**

Telephone CALumet 6141

Products:

Air Conditioning (room cooling, dehumidifying, ventilating, filtering and circulating) units—self-contained—portable floor type and portable window type—both air and water cooled. Commercial and Industrial Refrigeration equipment. Blast Coils for heating or refrigeration. Cooling Units and Heating Units—various capacities. Humidifying Units—automatic, for industrial or residential application. Drying Units—steam or electric. Ventilating Fans—for industrial or residential use.

New ADCO ¾-ton Air-Cooled Air Conditioner

Newly developed to fit the summertime requirements of average size hotel rooms, offices or residential chambers, the ADCO ¾-ton Air Conditioner and Dehumidifier is a complete, self-contained unit. It requires no plumbing. It occupies no more space than an average radiator. It draws in fresh air through a window duct, filters it, cools it and circulates it. When desired, circulating action may be reversed, to exhaust stale air, odors and smoke from the room. This feature, of course, does not affect action of refrigerating units. Designed around the extremely efficient and permanently quiet ADCO "V-8" compressor, direct-driven by a shaft integral with the motor armature. The cabinet is similar in style, although somewhat larger than the ½-ton size pictured. Consult factory for current data on exact dimensions, color and electrical characteristics.

Air Devices Corporation

(Subsidiary of Automatic Products Corporation)

AIR DEVICES CORPORATION
ADCO
CHICAGO ILLINOIS U.S.A.

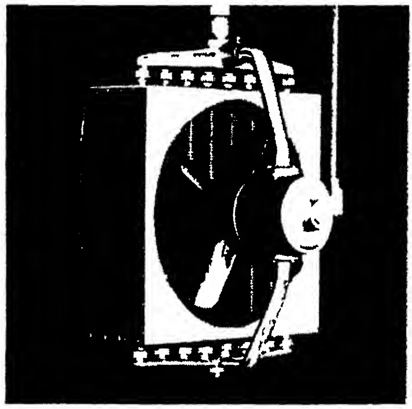
Manufacturers of Air Conditioning, Heating and Ventilating Equipment

Factory and General Sales Offices **64 East 25th Street, Chicago, Ill.**

Telephone CALumet 6141

ADCO ½-ton Window Type Air-Cooled Air Conditioner

Complete, self-contained—requires no water connections Simply mounted in place in any double-hung window as illustrated Draws fresh air from outside and a fixed percentage of air from inside, filters it, cools it, de-humidifies it, circulates it and discharges it through the same window Room capacity 1600 cu ft—½-ton melting ice equivalent ADCO "V-8" eight-cylinder compressor unit



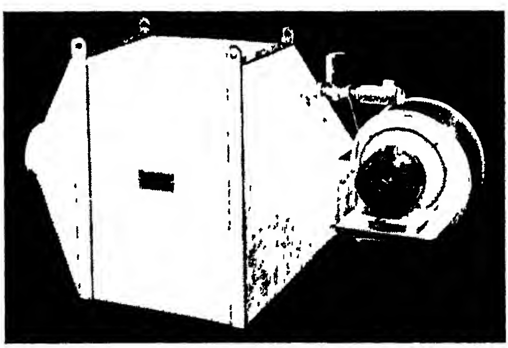
ADCO Industrial Unit-Heaters and Coolers

Made in three standard sizes, ranging in capacity from 30,000 to 450,000 Btu Leakproof, highly frost-resistant, non-electrolytic and non-galvanic Guaranteed to 250 lb pressure

ADCO Unit Coolers are built in five sizes, three types, i e, direct expansion, flooded or circulated operation, using Freon, Ammonia, Calcium Chloride Brine or Water Capacities ½-ton to 8 tons, m i e

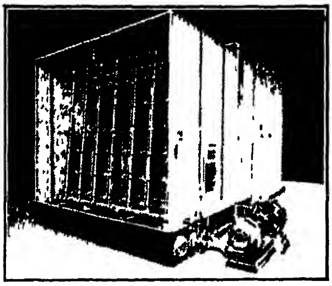
ADCO Automatic Humidifier

For use in conjunction with steam, hot water and vapor heating systems Made in five sizes to fit virtually any industrial or residential requirements Will automatically maintain any desired percentage of relative humidity Sturdy, compact, using ADCO all-aluminum heat-transfer elements as in ADCO Unit Heaters and Coolers Fully automatic, being governed by handset humidistat



American Blower Corporation-Detroit
Canadian Sirocco Co., Ltd.-Windsor

Division of American Radiator and Standard Sanitary Corp.



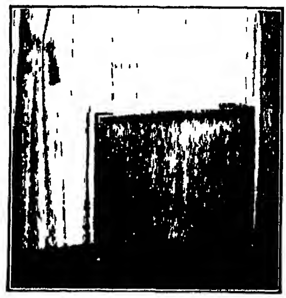
American Blower Dehumidifiers and Washers

American Blower Dehumidifiers

This line of dehumidifiers, air washers, scrubbers and purifiers, commonly referred to as the American Blower Washer, is available in standard sizes and designs, constructed of galvanized iron casing and eliminators with welded iron tank Capacity, 1000 to 300,000 cu ft of air per minute These washers are built in lengths from 4 ft 0 in to 24 ft 0 in. with one to six stages of sprays depending upon the duty to be performed Also built of special metal when required Bulletin No 3523

American Blower Conditioners Series "OW"

Series "OW" American Blower Conditioners are made in various sizes for comfort cooling of restaurants, shops, hospitals, offices, etc Units are provided with three-speed switch automatically controlled by means of a thermostat, and can be equipped with wall box for outside air intake Air filter and humidifier furnished extra For winter heating, units are connected to hot water boiler Complete data in Bulletin No 4327



Series "OW" Conditioner for cooling and heating offices, restaurants, hospitals, etc

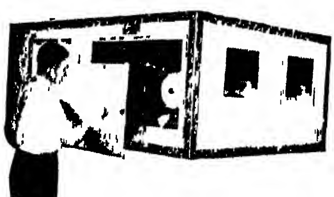
Series "RH" Conditioner

American Blower Conditioner Series "RH"

American Blower Series "RH" Conditioner for small commercial establishments For installations requiring only heating humidification and air cleaning during the heating system and where the summer requirements do not extend beyond the cooling effect resulting from air motion Units are equipped with steam or hot water heating coils only. Bulletin No 4127

American Blower Conditioner Series "B"

The American Blower Conditioner Series "B" is a unit conditioner and when used with a Decalorator (steam vacuum refrigeration) or mechanical refrigeration, constitutes a simple and complete air conditioning system for department stores, restaurants, coffee shoppes, offices and similar applications Bulletin No 3527



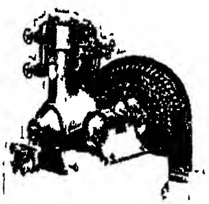
American Blower Conditioner Series "B"

Baker Ice Machine Co., Inc.

1518 Evans Street, **Omaha, Nebr.**

MANUFACTURERS OF INDUSTRIAL AND COMMERCIAL REFRIGERATION AND AIR CONDITIONING

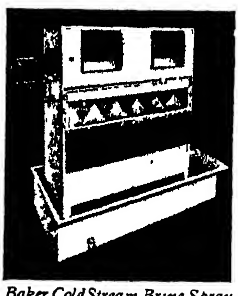
Get specifications for your requirements from the Baker line of equipment. Every machine and cooling assembly has been precision-manufactured and designed to offer maximum service, dependability and economy per dollar invested.



Baker Ammonia Compressor

Baker Ammonia Compressors

Available to 100 tons capacity, with synchronous, direct-connected or V-belt drive. Baker Compressors may be arranged in duplex or multiple installations for any desired capacity. Also equipped with double-suction, capacity reduction where conditions require utmost economy of operation. Available in automatically controlled self-contained units ranging from 1 to 25 tons capacity, 2 and 4 cylinder types.

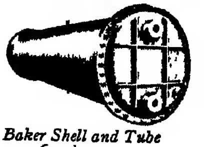


Baker ColdStream Brine Spray Unit—forced draft type

Baker Coldstream Brine Spray Units

Designed for applications requiring uniform control of temperatures and relative humidity. Equipped with slow-speed, blowers mounted on ball bearings. Fan speeds may be changed to suit air velocity requirements. Housing is of boiler plate construction.

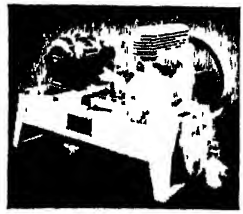
Baker Shell and Tube Condensers



Baker Shell and Tube Condenser

Baker Condensers and Liquid Coolers are made in all sizes up to 2500 sq ft of effective cooling service. Vertical, horizontal, multipass or single-pass types available, with diameters and tube lengths to fit any specification.

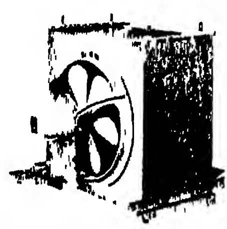
Baker Freon or Methyl Chloride Units



Baker Freon or Methyl-Chloride Unit

Baker offers a complete line of 77 models assembled in both single and dual mounted self-contained automatic units from 1/4 hp to 60 hp capacity, two and four cylinder types. Mechanical features include double-trunk type, semi-steel pistons, full force feed lubrication, Timken Bearings and shell and tube condensers. Both air cooled and water cooled models available.

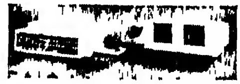
Baker Fan Type Coldstream Units



Baker Fan Type ColdStream Unit

Rigidly constructed for compact, high-capacity, heavy duty service in refrigerating and air conditioning foods and other perishables requiring positive control of temperatures above the freezing point. Finned coil surfaces and air velocity designed for a correct combination of temperature and relative humidity.

Baker Ceiling Type Coldstream Blower Units



Baker ColdStream Blower Unit—ceiling type, front and rear views

Designed for comfort cooling or commercial and industrial air cooling. Equipped with finned or galvanized bare pipe coil, fans direct connected or V-belt driven. Also made in **Floor Type Units** ranging in size from 2 to 16 tons refrigerating, 2000 to 16000 cfm air capacity.

CARBONDALE

Carbondale

Division

Worthington Pump and Machinery Corporation

WORTHINGTON

General Offices HARRISON, NEW JERSEY

Branch Offices and Representatives in Principal Cities CA7-1

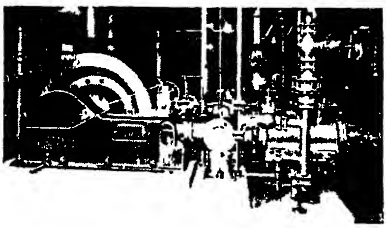
REFRIGERATION SYSTEMS FOR AIR CONDITIONING IN COMFORT COOLING OR INDUSTRIAL PROCESS

Complete refrigerating systems for use with Freon (F-12), Methyl Chloride, Ammonia, or Carbon Dioxide, either direct-expansion or water cooling applications A complete line of refrigeration compressors, permitting impartial recommendations A nationwide organization of Dealer-Distributors in major cities to provide sales and engineering service and plan complete air conditioning systems of the central or unit type Architects, Engineers, and Contractors are invited to consult with us Write to Harrison, New Jersey for bulletins covering products illustrated below

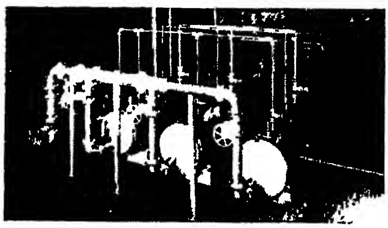
Air Conditioning Units vertical and horizontal 500 to 15,000 c f m

Freon 12 or methyl chloride self-contained commercial units with motors up to 25 hp and ratings up to 25 tons

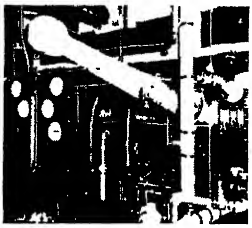
Shower Condensers for Freon 12 or methyl systems—5 to 50 tons refrigeration



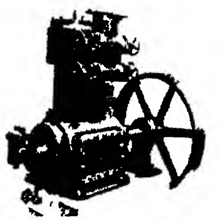
Horizontal ammonia compressors, single and duplex, belt, direct motor and steam drive, 60 to 100 tons Capacity control features available



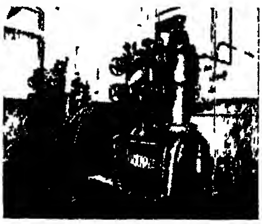
High and low side equipment Coils, coolers, condensers, receivers, controls, pumps, valves, connections, fittings



Steam-jet vacuum cooling equipment Chilled water 35°F or over 2 to 1000 tons at 50°F Also centrifugal water vapor units, 50 to 300 tons



Vertical ammonia compressors, pressure-lubricated roller main bearings, safety heads, patented Feather Valves, 2 to 200 tons



Special vertical Freon Compressors, pressure-lubricated, roller main bearings, safety heads, up to 150 tons

Compressor requirements above 250 tons are best met by single or duplex horizontal machines Units up to 750 tons are available, proportioned for Freon 12 and equipped with three-step stuffing boxes

Clarage Fan Company

Kalamazoo, Michigan

Sales Engineering Offices in All Principal Cities (Consult Telephone Directory)

CLARAGE

AIR HANDLING AND CONDITIONING EQUIPMENT

For Nearly a Quarter Century Clarage has been a leading manufacturer of air handling and conditioning equipment There is a Clarage fan, unit or system to meet every need, from the simplest ventilating or cooling job to the most exacting temperature and humidity control installation

Whatever your ventilating, unit heating, cooling, air cleaning, humidifying, dehumidifying or complete air conditioning problem, Clarage can meet your requirements successfully and economically.

The Experience of Clarage Engineers covers almost every conceivable type of installation, commercial, industrial and public building Clarage equipment is used in the largest industrial plants, office buildings, auditoriums, theatres, hotels, restaurants, retail stores, hospitals, churches and schools.

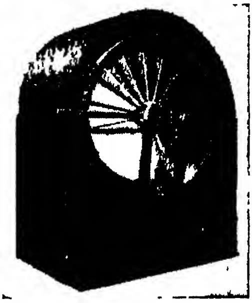
Clarage Vortex Control System is an important development in air conditioning. It delivers only the volume of conditioned air required by the sensible heat load existing at any given time. Compared to the conventional system, first and operating costs are definitely lower.

Architects, Engineers and Contractors will find our service specially helpful This Company is an independent manufacturer, and to specify and use our apparatus does not involve any binding agreements or license fees whatsoever Your inquiry for complete data on any Clarage product is invited

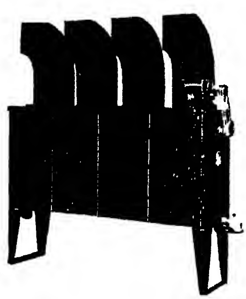
Vortex Control System for commercial and industrial conditioning

Moditherm Unit, complete conditioning plant designed for ceiling or wall installation

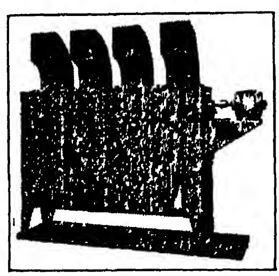
Duotherm Unit, complete conditioning plant for fine homes, installed in basement



Type HV Ventilating Fan with Clarag Vortex (constant speed) Volume Control



Unitherm Unit Heater with Syncrotherm Control as standard equipment

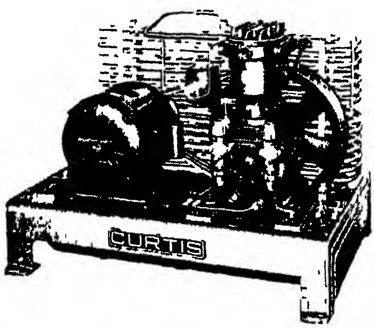


Unitherm Unit Cooler for product cooling and refrigeration

Curtis Refrigerating Machine Company

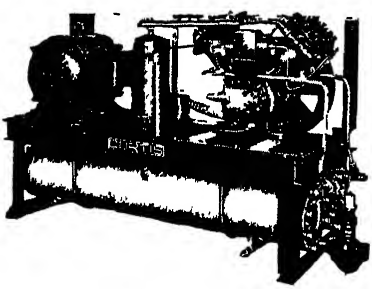
***Division of* Curtis Manufacturing Company**

St. Louis, Missouri



Double Tube Type

High efficiency, double tube counter flow condenser, multi-banked for low pressure drop



Shell and Tube Type

Dual shell and tube condensers permitting series of parallel operation to suit every condition

SOME FEATURES OF CURTIS UNITS

The curtis "Centro-Ring Positive Pressure" oiling system assures proper lubrication without dependence on gear or plunger type oil pump Only one simple moving part

Timken tapered roller bearings

Water jacketed heads and cylinders

Each unit equipped with built-in oil separator with automatic return

Balanced sylphon bellows seal

Efficient drop-forged, heat-treated crankshaft and connecting rod construction

Stainless steel disc type suction and discharge valves

Automatic water valves

Extra large water-cooled condensers

Compressors of proven "V" type radial design

CURTIS builds a complete line of condensing units for every air conditioning and refrigeration requirement, up to 30 tons inclusive Every Curtis unit reflects 83 years engineering, designing and manufacturing experience, over 43 years of this specializing in building of fine compressors

CURTIS also builds dual compressor units in capacities above 20 hp , providing automatic capacity control for variable air conditioning loads

There is a Curtis unit for every purpose—from fractional tonnage capacity up to 30 tons, for Freon or Methyl Chloride

ALBANY
ATLANTA
BALTIMORE
BOSTON
BUFFALO
CHARLOTTE
CHICAGO
CINCINNATI
CLEVELAND
DALLAS
DETROIT
DES MOINES

Frick Company

(Incorporated)

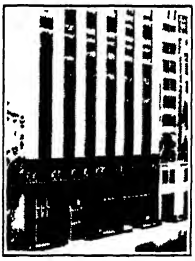
Air Conditioning, Refrigerating and Ice-Making Equipment

Waynesboro, Penna.

KANSAS CITY
LOS ANGELES
MEMPHIS
NEW ORLEANS
NEW YORK
OKLAHOMA CITY
PALATKA
PHILADELPHIA
PITTSBURGH
ST LOUIS
SEATTLE
WASHINGTON

Distributors in 100 Principal Cities

AIR CONDITIONING WITH FRICK REFRIGERATION



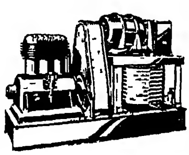
Fourteen Stories of this Building at 22 East 40th Street, New York City, are Air Conditioned with Frick Freon-12 Refrigeration

Is the term applied to our service in supplying refrigerating equipment and engineering assistance for complete air conditioning jobs, which are handled by Frick Branches and Distributors — located in principal cities throughout the world

Estimates cheerfully submitted Get data on special Frick air conditioning systems arranged for automatic direct-expansion operation Economical, safe, highly satisfactory let us refer you to typical installations Ask for Bulletins 504 and 512

FRICK FREON-12 REFRIGERATION

Includes the most complete line of enclosed type Freon-12 compressors Large capacity, ample gas passages, pressure lubrication from internal pump, patented FLEXO-SEAL at shaft Coils, coolers, condensers and controls for Freon-12 systems Bulletin 508



15-Ton Freon-12 Unit for Air Conditioning Work

AMMONIA REFRIGERATION

Machines in all capacities from 1½ ton up Combined units, vertical enclosed type compressors, horizontal machines complete high and low sides Widely used for air conditioning Bulletins 102 to 138

The Standard Carmel Company of Lancaster, Penna, has used Frick Ammonia Refrigeration for Air Conditioning Since 1910

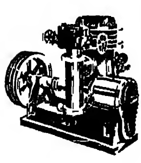
CARBON DIOXIDE REFRIGERATION

Six sizes of enclosed CO_2 compressors smooth running, efficient and reliable machines, condensers, coolers, etc Bulletins 118 and 124

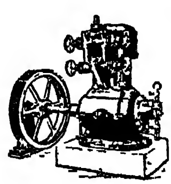
LOW PRESSURE REFRIGERATION

The World's Largest Air Conditioned Storage Room, at Vernon, Calif, is Cooled with Frick Refrigeration

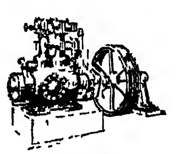
Commercial units in more than 50 sizes and types, with motors of ¼ to 30 hp Charged with Freon-12 or methyl chloride Air and water cooled condensers Finned coils, fan and blower units, ice cube and beverage coolers, etc Bulletins 97 and 98



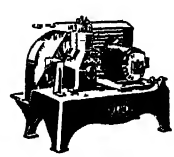
Frick Enclosed Freon-12 Compressor



Enclosed Type Ammonia Compressor



Enclosed Compressor for Carbon Dioxide

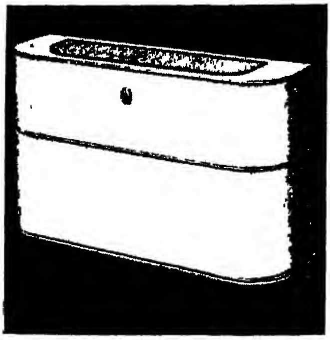


Low Pressure Refrigerating Units

Fairbanks Morse & Co.

Air Conditioning

900 S. Wabash Ave. **Chicago, Illinois**

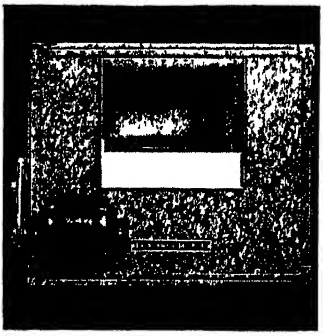


Floor Type Models

Fairbanks-Morse "Ortho-Clime" air conditioners are made in a wide range of attractively designed and efficiently engineered models in capacities from 1 to 20 tons

Floor type units, of smart modernistic design, in 1 and 2 tons capacity are available for installation in the conditioned space May be had for cooling service only, using Freon or water as the refrigerant, or for combination cooling and heating service to replace the ordinary room radiator for year 'round conditioning including automatic humidification in the winter time

Cabinets are attractively finished in rich cream enamel or grained walnut or mahogany with brushed-aluminum trim



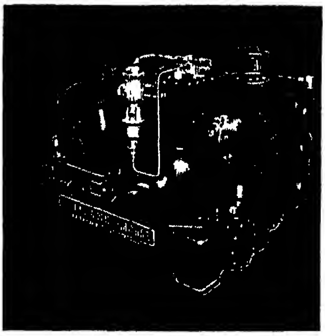
Model 760 Central Station Unit

The Model 760 Central Station Unit, which has a capacity range of from 3 to 5 tons, is available either with duct connection for remote installation or with diffusing grille for installation in the conditioned space May be had with coils for either Freon or cold water operation on the cooling cycle Heating coils with automatic humidification attachment are available with either type of cooling coil to provide year 'round conditioning. This unit may be suspended from the ceiling or may be installed on a door or window deck to provide convenient transmission of air to the conditioned space with a minimum amount of installation expense.

Compressor Units

Fairbanks-Morse Freon condensing units or high sides are available in capacities from 1 to 20 tons

The compressor is provided with water-cooled automotive type cylinder head to insure long valve life and low water consumption Oversized discharge and suction valves insure durability and high efficiency Balanced pressure seal, which has proved its dependability in more than ten years of service in commercial refrigeration, effectively prevents escape of gas at the crank shaft Detailed specifications and capacity sheets available on request.

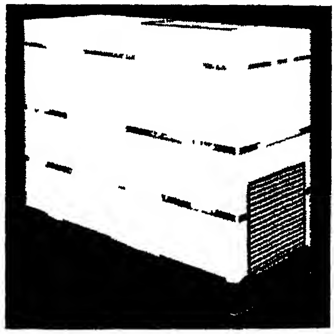


Fairbanks Morse & Co.

Air Conditioning

900 S. Wabash Ave. **Chicago, Illinois**

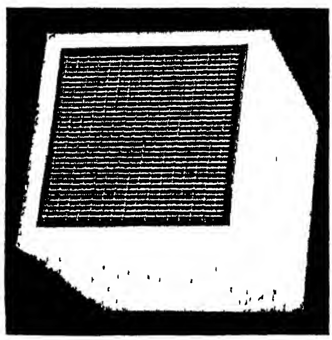
Unit Room Cooler



The Fairbanks-Morse 1937 Model Unit Room Cooler is a complete, self-contained unit of the plug-in type It has a capacity of 10,000 Btu per hour or a cooling effect equivalent to more than 1600 lbs of ice per day Furnished for either A C or D C current Easy to install—no water connections to make Exceptionally quiet in operation Carefully balanced for smooth operation Attractively finished in grained walnut

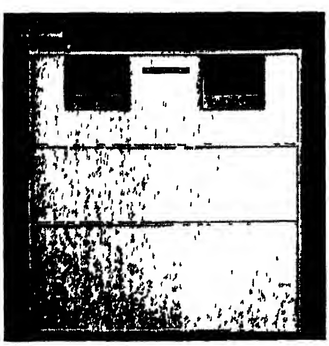
These units have a wide application for the air conditioning of offices, hotels, apartments, hospital rooms, residences, etc.

Model 715 Ceiling Type



This unit, which has a capacity of approximately 1½ tons, is designed for installation in the conditioned space Its smooth exterior lines and attractive finish makes it practical for installation in offices, stores, shops, fitting rooms, etc , where floor space is at a premium and appearance is a vital consideration

Standard finishes are rich cream enamel or grained walnut or mahogany Dimensions, 21 in high, 21 in wide and 26½ in deep Designed for ceiling suspension but may be installed on any convenient door or window deck



Central Plant Units

These plants are available in capacities from 7 to 20 tons in either vertical or horizontal type Either design may be had for horizontal or vertical discharge of air

These plants will perform all the functions of summer cooling and dehumidifying, winter heating and humidifying, or both

Vertical types are designed for floor mounting, horizontal types for ceiling suspension or installation on a door or window deck Carefully balanced fan Quiet in operation Air capacities from 2500 to 6500 cfm

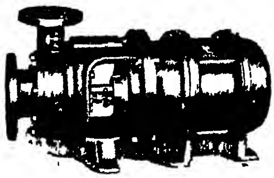
Ingersoll-Rand Company

11 Broadway, **New York City**

Branches or Distributors the World Over

Birmingham	Chicago	Detroit	Knoxville	Picher	Scranton
Boston	Cleveland	Duluth	Los Angeles	Pittsburgh	Seattle
Buffalo	Dallas	El Paso	Newark	Pottsville	St Louis
Butte	Denver	Hartford	New York	Salt Lake City	Tulsa
		Houston	Philadelphia	San Francisco	Washington

CAMERON PUMPS

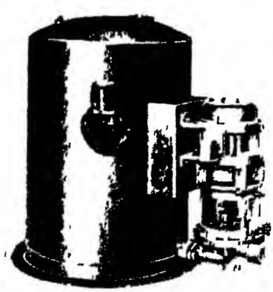


Cameron Motorpump

There is an efficient, reliable Cameron pump for every purpose Single-stage centrifugal units range from 40 to 100,000 g p m, and multi-stage units (2 to 8 stages) range from 125 to 3000 g p m for pressures up to and over 1700 lbs

The Cameron "Motorpump" is ideal for general service everywhere, ranging in sizes from ¼ to 40 hp Pumps and motors are built as one compact and efficient unit on a single shaft They can be mounted in any position on the floor, wall or ceiling Capacities range from 5 to 1000 g p m for heads as high as 240 ft single-stage, and from 20 to 275 g p m for heads up to 500 ft two-stage Motors for any common current conditions, open, splash-proof, totally-enclosed, or explosion-proof types

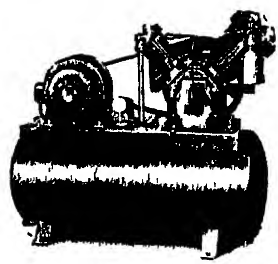
CONDENSATE RETURN UNITS



"Motorpump" Condensate Return Units are designed to return condensation from radiation systems, heaters, steam coils, etc

The units consist of standard "Motorpumps" mounted on 15-, 30-, or 60-gal tanks and controlled by a float switch They are suitable for use in schools, apartment houses, greenhouses, industrial plants, chemical plants, etc They are often used to replace inefficient steam traps

AIR COMPRESSORS



Type "30" Two-Stage Compressor

A "Type 30" Air Compressor is an excellent source of compressed air for use with regulating devices or control equipment, and for many other applications requiring small capacities It is a self-contained plant, consisting of a compressor, driving motor, air receiver and automatic pressure control switch The compressor is either single- or two-stage, using I-R plate valves Cylinders and intercooler are air cooled. Capacities range from 1 2 to 82 cu ft per min Pressures range from 5 to 1000 lb per sq in

Ingersoll-Rand manufactures more than 1000 sizes and types of compressors including a complete line of ammonia compressors

OTHER I-R PRODUCTS

Surface condensers, counter-current and ejector-jet barometric condensers, steam-jet ejectors, vacuum pumps, centrifugal blowers, air aftercoolers and receivers, Diesel and gas engines, air and electric hoists, rock drills, and pneumatic tools of many kinds

WATER-VAPOR REFRIGERATION

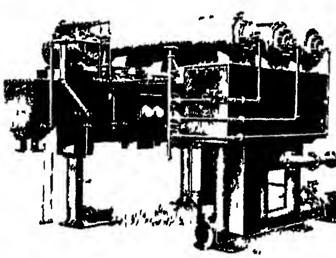
I-R Water-Vapor Refrigeration uses water as the only refrigerating medium There is no refrigerant to purchase or replace The system operates entirely under vacuum, and may be opened up for inspection whenever desired

Water-Vapor Refrigeration gives low operating costs by saving refrigerant replacement and by economical operation at light as well as full loads Of special value is its unusual ability to handle overloads Full capacity is retained for the life of the machine

Two types of units—Steam-Jet Cooler and Centrifugal—permit the selection of the most economical type of equipment to meet widely varying conditions of steam or electric power, cooling water supply and refrigerating requirements

I-R Units are ordinarily sold through reliable contractors for field installation as a part of air conditioning systems

STEAM-JET COOLER TYPE

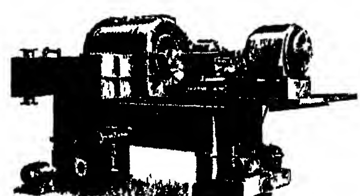


Cooling is effected by direct evaporation of water at high vacuum Steam-jet boosters maintain the vacuum and discharge the evaporated water vapor into a surface condenser (When desired, barometric type condenser units can be furnished) The heat carried away by the evaporation of the water vapor chills the main body of water as it is circulated through the unit

The I-R patented design has been specially developed to insure reliability and simplicity of operation Steam-Jet Coolers are free from noise and vibration, and are easy to install and maintain

Standard sizes are available for capacities of approximately 10 tons upward for chilled water temperatures of 35° F and higher Units are built for steam pressures of 2 lb gauge and upward.

CENTRIFUGAL TYPE



The centrifugal water-vapor refrigerating unit likewise cools water by direct evaporation of water at high vacuum A centrifugal type compressor maintains the vacuum by compressing the water vapor and discharging it into the condenser

Units can be furnished with electric-motor or highly efficient steam-turbine drivers Turbines may be either of the low-pressure or high-pressure type, and may be designed for back pressure or for condensing operation

An important feature is self-regulation, units simply float on the load, giving inherently reduced power consumption with decreasing demand The capacity is sustained for life due to the elimination of wearing parts

Standard sizes are available for capacities of approximately 100 tons upward

Niagara Blower Company

AIR ENGINEERING EQUIPMENT AND SYSTEMS

General Sales Office: 6 East 45th Street, **New York City**

Buffalo Rochester Boston Philadelphia Cleveland Toledo
Pittsburgh Chicago St Louis Seattle San Francisco

15 Years experience in the engineering, design and installation of complete air conditioning

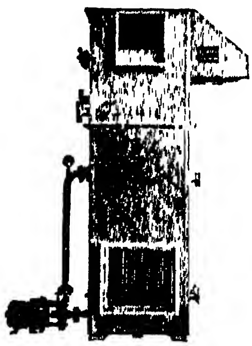
PRODUCTS—Exact Control Air Conditioning, Humidifying, Dehumidifying, Drying, Moistening, Chilling, Comfort Systems, Niagara Air Conditioners, High Humidity Spray Coolers, Fan Coolers, Fan Heaters, Cooling Coils, Heating Coils, Evaporative Aero Condensers.

NIAGARA AIR CONDITIONING SYSTEMS

For human comfort and for all industrial applications requiring controlled climatic conditions of temperature, relative humidity, air purity and air movement

NIAGARA AIR CONDITIONER, TYPE A

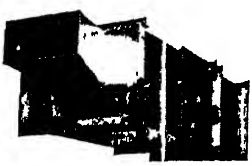
Maintains constantly, or makes any change required in temperature and relative humidity, dries or moistens within tolerance of 1 degree F and 2 per cent R H in processing hygroscopic materials, cleans air effectively, secures saturation for dehumidifying Seven sizes Available both in floor-mounted and space-saving suspended types.



Niagara Air Conditioner, Type A manufactured in 7 sizes, both floor mounted and ceiling suspended models

NIAGARA AIR CONDITIONER TYPE C

Uses Surface Cooling Coil method for cooling and dehumidifying Manufactured in sections so that any desired combination of air conditioning functions may be obtained, also solves installation problem in existing buildings as its sections can be brought thru any doors and erected in confined spaces



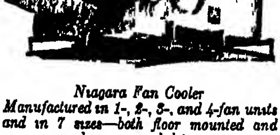
Niagara Type C Air Conditioner Niagara All Aluminum Surface Coil Air Conditioner, manufactured in 7 sizes both floor mounted and ceiling suspended models

NIAGARA SURFACE COOLING COIL METHOD

Air conditioning for human efficiency and comfort A year around operating system providing winter heating and humidifying and summer cooling and dehumidifying for offices, stores, restaurants and all places where people gather. An advanced engineering method introducing simplified apparatus and controls, for superior operating results, economy and long life, which cut operating costs

NIAGARA FAN COOLER

Recommended for comfort cooling, process cooling, low temperature storage for dairies, fruits, meats, food products, fur storage vaults, etc. Gives complete circulation of air at desired temperature with even temperature at all points Manufactured in seven sizes



Niagara Fan Cooler Manufactured in 1-, 2-, 3-, and 4-fan units and in 7 sizes—both floor mounted and ceiling suspended types

NIAGARA DISK FAN COOLER

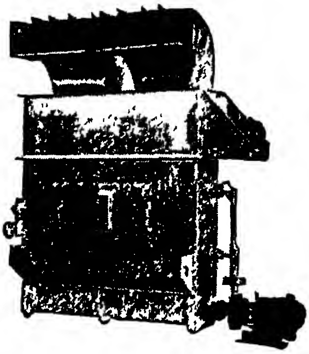
For overhead suspension, saves space and provides the efficiency of moving air cooling for small storage areas, market coolers, etc

Niagara Disk Fan Cooler

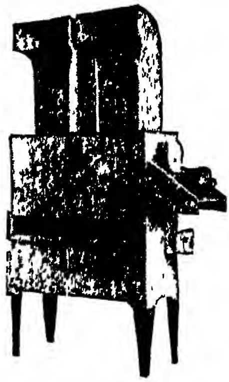
NIAGARA SPRAY COOLER

For all cooling applications requiring high humidity or high capacity in small space such as food product applications, especially meat chilling and storage pre-cooling and storage of fruits and vegetables Prevents drying out, wilting Also used in industries requiring freezing temperatures such as ice-cream manufacture and frozen food storage.

Niagara Spray Cooler is built with cooling coils in a constant brine or water spray Maintains constant relative humidity as required Available both in floor-mounted and space-saving suspended types



Niagara Spray Cooler
Illustration of 2-fan unit Also manufactured in 1-, 3-, and 4-fan units and in 7 sizes, both floor mounted and ceiling suspended models



Niagara Fan Heater
Illustration of 2-fan unit Also manufactured in 1-, 3-, and 4-fan units and in 7 sizes, both floor mounted and ceiling suspended models

NIAGARA EVAPORATIVE AERO CONDENSER

Saves power and water cost utilizing atmospheric air to remove heat of condensation, consuming only the small amount of water evaporated by the air c.rculated Seven sizes

NIAGARA FAN HEATERS

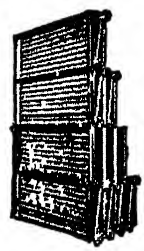
For the heating and ventilating of large areas, Niagara Fan Heaters put the heat immediatly where needed in the working zone, give quicker heating up to working temperatures Definitely built to the highest possible standards, welded alummimun heating coils

NIAGARA HUMID HEATER

Recommended for industrial applications where heat and humidity are required as in manufacture of textiles, cordage, printing and paper-converting plants

NIAGARA COOLING COILS

High pressure tested cooling coils are used in Niagara Fan Coolers and Niagara Surface Method air conditioning Encased 13 standard lengths for blast cooling installations, in both 20-in and 30-in widths Manufactured in aluminum and copper

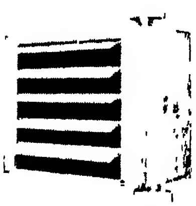


Niagara Aluminum Heating Coils

NIAGARA ALUMINUM HEATING COILS

For use with fan heating systems giving the advantage of aluminum, light weight and resistance to corrosion Manufactured in two widths, 20-in and 30-in , and in various lengths, giving a complete range of sizes 150 lbs working steam pressure

Niagara Aluminum Booster Heaters are used for reheaters to control room temperature independently of fan system



Niagara All Aluminum Disk Fan Heater

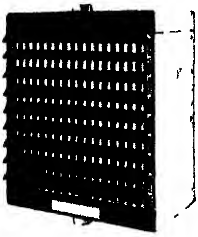
NIAGARA DISK FAN HEATERS

Operate with low discharge temperature, cut down roof and wall losses.

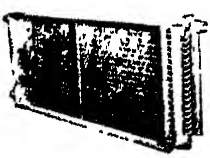
McQuay, Inc.

1600 Broadway, N.E., **Minneapolis, Minn.**

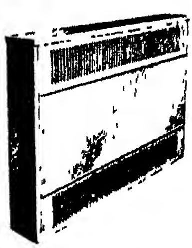
UNIT HEATERS — BLAST HEATERS — CONVECTION RADIATION AIR CONDITIONING COILS—EVAPORATOR COILS—UNIT COOLERS



Series "H" Unit Heater



Air Conditioning Coil



Copper Convector

The McQuay extensive line of **Heating, Cooling,** and **Air Conditioning** units can best be understood by analysis of the special catalogues which describe fully the various products

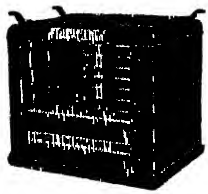
McQuay Unit Heaters with full floating all-copper heating elements are available in thirty-four sizes

McQuay Extended Surface Coils for **Central Fan Heating and Cooling,** and for all Commercial drying purposes are available in a wide range of sizes to meet any Air Conditioning application

McQuay Concealed and Cabinet Convectors are available in all enclosure types, exposed floor and wall hung, fully or partially recessed with removable panels, and totally concealed

McQuay Unit Coolers available in seven sizes for Walk-In Storage Rooms, Truck Refrigeration, etc For all types of refrigerants including ammonia

McQuay Comfort Coolers available for water or brine and direct expansion refrigerants Numerous sizes with capacities to fit your requirements



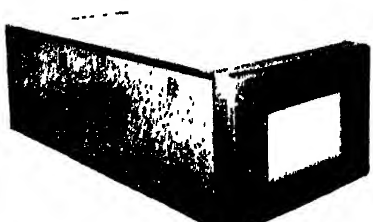
Unit Cooler

Ice Cube Maker

Evaporator Coil

McQuay Evaporator Coils designed for all phases of commercial refrigeration work Fins with special spun collars with tubes hydraulically expanded into the fins All joints brazed assuring no leaks McQuay method of manufacturing makes possible a flexible line of special coils tailor-made to fit your specific requirements Coils for ammonia installations available for all purposes

McQuay Suspended, Centrifugal Fan Type Cooling and Heating Units available in 3 - 5 - 7½ - 10 - 15 and 20 tons capacity Units are adaptable for many Air Conditioning applications



Winter Air Conditioning Unit

McQuay Winter Air Conditioning Unit. A low cost residential Air Conditioning unit for use with radiator heating systems A suspended type basement unit which filters, humidifies and circulates the air

Bulletins descriptive of all McQuay products available upon request

Parks-Cramer Company

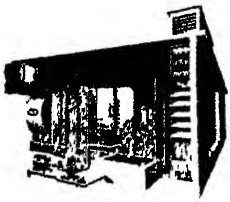
Fitchburg, Mass. **Charlotte, N. C.**

CERTIFIED CLIMATE

Complete Air Conditioning Systems including Heating, Cooling, Humidifying, De-humidifying, Ventilating, Refrigeration, Air Filtering and Air Washing

AUTOMATIC REGULATION

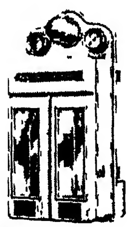
Industrial Heating by Oil Circulation with Merrill Process



Central Station

Central Station Air Conditioning

A centrally located apparatus supplying maximum moisture needed with positive pre-determined air change Usually includes indirect radiation for heating —may include refrigeration and cooling All air is washed Where absolute and centralized control is desirable — helps in many industries, notably, Textiles (Cotton, Wool, Worsted, Silk, Rayon, Jute), Printing and Lithographing, Cigar, Cigarette and Tobacco, Clothing, Paper and Envelope, Leather and Shoes, Wood Products, Cereals, Storage of Perishables, Ceramics, Celluloid, Glassine Paper, Starch and Dextrine, Cement Installations similar in design are effective in Hospitals, Art Galleries, Auditoriums and Restaurants



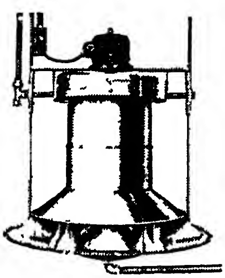
Psychrostat

Automatic Airchanger (Not Illustrated)

Insures maximum evaporative cooling in hot rooms in summer Conserves heat in winter Positive air circulation and uniformity of humidity and temperature Mostly for industrial application

Automatic Regulation

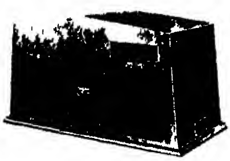
The Psychrostat for accuracy, durability, sensitivity Hygrostat (not illustrated) where requirements are not so exacting Psychrostat employs the principle of the Sling Psychrometer, used by U S Government in all Weather Bureau Stations An Air Conditioning System is no better than its Regulation



High Duty Humidifier

High Duty Humidifier

Water under pressure generates spray Excess water returns to filter tank and recirculated Evaporation per unit high, two sizes of heads each with two sizes of nozzles give flexible capacity for varying conditions Circulation increased by individual motor-driven fan Spray thoroughly diffused and distributed over wide area



Pettifogger

The Pettifogger

A compact humidifier for offices, stores, storerooms, testing laboratories, and other isolated departments Entirely self- contained in attractive lacquered copper casing Permanently though flexibly connected to water and electrical supplies Automatic control Adjustable capacity Reduces dust Neutralizes drying effect of heating Conditions textiles and other hygroscopic substances for testing purposes Safeguards health Increases bodily comfort Inexpensive.

Servel, Inc.

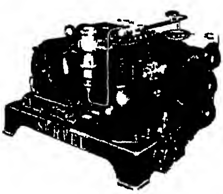
Electric Refrigeration and Air Conditioning Division

Evansville, Indiana

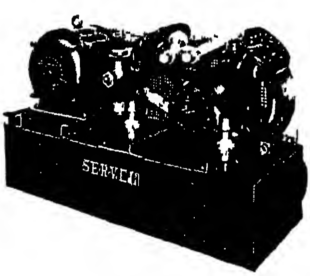
AIR CONDITIONING

Servel offers a complete line of refrigerating machines for air conditioning duty, developed along the same basic lines as the Servel commercial machines which have been serving users for 15 years All are of the reciprocating compressor type with sealed crankcase especially suited to F12 and similar hydrocarbon refrigerants All models may be used with city water, cooling towers or atmospheric condensers Copper lead bearings, high volumetric efficiency, true counter-flow condensers and other important design features, combined with precision workmanship, insure low operating and maintenance cost

WAH Series



WAO Series



WAV Series

Complete Coverage

Seven series of machines, ranging from 1/3 ton to 20 tons, meet every practical requirement of the commercial and residential air conditioning field

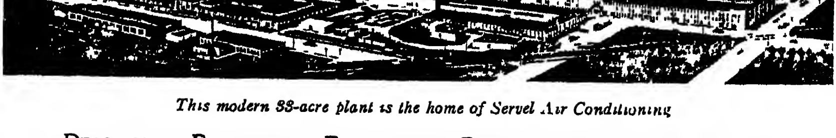
16 Standard Models

The WAV four-cylinder series includes 15-ton and 20-ton models The twin-cylinder WAG series offers 7 and 10-ton capacity The WAO series, developing 4 to 6 tons, serves a broad commercial market Small commercial and residential requirements of 2 to 3 tons are met by the WAE series Private offices and smaller homes find the WAD series of 1 to 1½ ton ratings practical and economical For individual rooms in homes and for small self-contained conditioners, the WAH and WAC series, developing from ⅓ to ¾ ton, afford high capacity per unit of space

Factory Cooperation

Architects and design engineers are invited to communicate with our applications department for details and advice

Our engineers will gladly give assistance in the selection of equipment and the proper balancing of evaporators and auxiliary apparatus



This modern 88-acre plant is the home of Servel Air Conditioning

DESCRIPTIVE FOLDERS AND ENGINEERING DATA WILL BE SENT ON REQUEST

Thermal Units Manufacturing Company

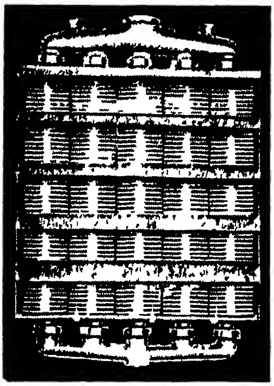
Subsidiary of Air Devices Corporation

64 East 25th Street, **Chicago, Ill.**

Representatives in All Principal Cities

PRODUCTS—Heaters, Unit; Coolers, Unit; Humidifying Units; Coils, Heating and Cooling, Fans and Automatic Refrigerating Compressors for Commercial and Air Conditioning Work

THERMAL UNIT HEATERS—Six Sizes



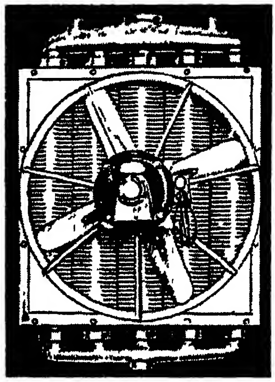
With permanent one piece—integrally cast—aluminum element Capacities from 15,000 to 500,000 Btu per unit

No joints, welds, brazed or soldered connections

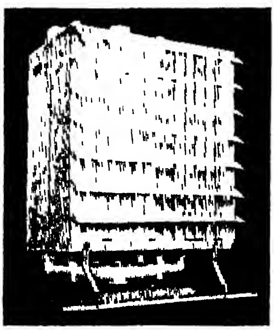
Leakproof, Freezeproof, indefinite life without servicing

Maximum air delivery per horsepower, with low outlet air temperature

Built for troubleproof service and long life



THERMAL UNIT COOLERS—Five Sizes

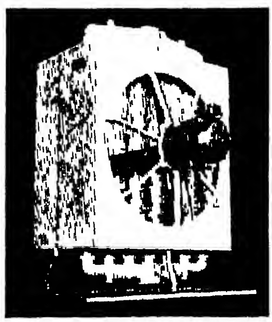


For Ammonia, Brine, Freon, water and other refrigerants Three types Circulating, Flooded or direct Expansion Capacities from ½ ton to 8 tons refrigerating effect

Employs the same sturdy heat transfer element as our Unit Heater above

Automatic operation, Automatic defrosting, with controls

Eight years of successful applications, of all types



THERMAL UNIT "V8" AUTOMATIC REFRIGERATING COMPRESSORS

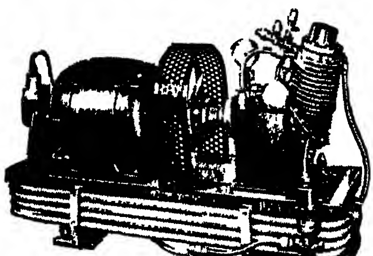
Simplicity - Efficiency - Economy—Using Freon or Methyl-Chloride

High Volumetric Efficiency - Low Cost - Trouble-Free Operation

Eight cylinders and pistons for smooth operation.

Silent operation for air conditioning

Light weight, more compact for delivered refrigeration tonnage



Direct Motor drive, vibrationless operation

Pressure lubrication, for longer life and fewer wearing parts

Economical operation with increased efficiency

Catalogs, information, and Engineering data furnished on request

UNIVERSAL COOLER CORPORATION

Automatic Exclusively — *Refrigeration Since 1922*

Detroit, Michigan

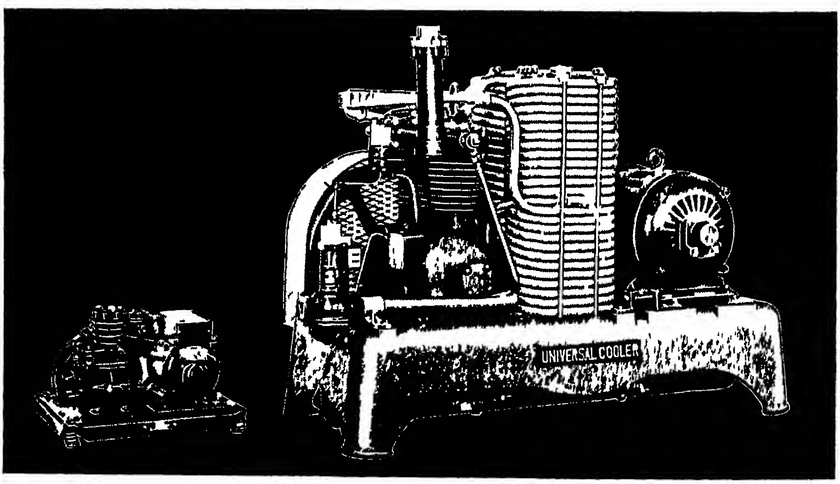
In Canada
Brantford, Ont

Condensing Units by Universal Cooler form a completely overlapping line of refrigeration equipment for all automatic commercial applications: Store Case Refrigeration, Walk-in Coolers, Liquid Cooling, Air Conditioning, Ice Cream Freezers, Ice Cream Cabinets, Truck Refrigeration and all types and uses of automatic cooling whether operated by electricity or gasoline engines

Universal Cooler condensing units have been developed by engineers fully conversant with the requirements of the air conditioning industry During the past few years a great many machines have been installed, and the 1937 line reflects the experience gained on applications of every kind

Regardless of your needs send a detail of your requirements and learn what Universal Cooler offers A total of 200 units from which to choose

Ratings of Universal Cooler Condensing Units are certified to the *Refrigeration Machinery Association* and the *National Electrical Manufacturers Association* under *American Society of Refrigeration Engineers'* standards



The Vilter Manufacturing Company

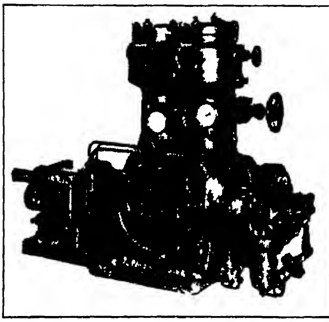
Since 1867

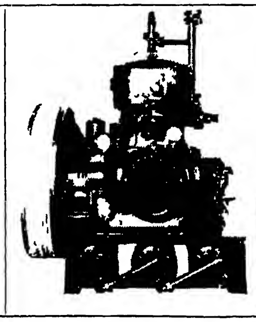
Milwaukee, Wisconsin

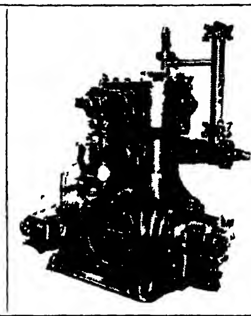
AIR CONDITIONING EQUIPMENT FOR INDUSTRIAL OR COMFORT COOLING

COMPRESSORS OF MODERN DESIGN

Ammonia **"Freon 12"** or **Methyl Chloride**







Compressors by Vilter are the result of nearly seventy years of research, development and experience gained through thousands of installations in sizes from one ton to several hundred tons These installations cover a range of applications from the most simple comfort cooling installation to the exacting requirements of temperature and humidity control necessary to the successful operation of certain industries

The new Vilter "Freon 12" Compressors embody many outstanding new features two of which are of major importance and exclusive A new, Vilter developed, shaft seal which prevents leakage, and the elimination of certain friction factors which result in extremely low relative horsepower per ton

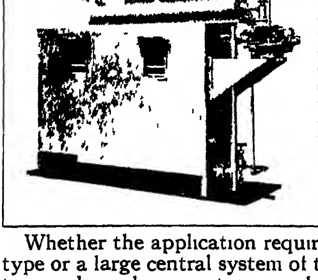
Vilter Air Washers—designed for industrial air conditioning Positive control of humidity, temperature, and circulation of air Automatic or hand operated Eliminators are incorporated for entrained moisture removal Odors and dust are removed by water sprays No filter replacement Low cost operation Good for low or high temperature cooling

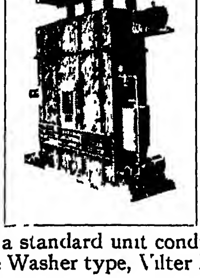
Vilter Mono-Unit Air Conditioners—are built in a complete range of sizes and types from small ceiling units to large floor units

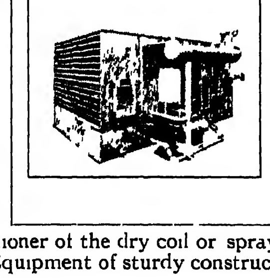
For refrigeration and air conditioning projects Vilter can supply the equipment

UNIT AIR CONDITIONERS

Dry Coil **Spray Type** **Air Washer**







Whether the application requires a standard unit conditioner of the dry coil or spray type or a large central system of the Washer type, Vilter Equipment of sturdy construction and ample capacity is available

Bulletins describing this Vilter Equipment are available for the asking.

Westinghouse Electric & Manufacturing Co.

Mansfield, Ohio

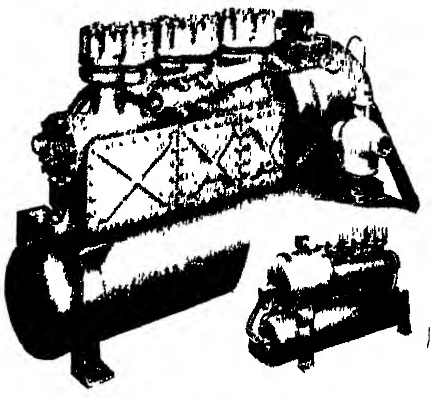
Sales, engineering and service facilities available through local distributors in principal cities

HERMETICALLY-SEALED CONDENSING UNITS

The newest development of Westinghouse Research and Design Engineers is a complete line of *Hermetically-sealed* Condensing Units for air conditioning—from 1 to 23 tons capacity These new machines are forty per cent smaller in size, as much as fifty per cent lighter in weight They permit great flexibility in the design of systems, save money in original installation costs and upkeep

Entire operating mechanism is enclosed in one solid casting, eliminating protruding shafts and troublesome "stuffing boxes" Direct drive of crankshaft by motor located *inside* the housing eliminates the necessity of a heavy supporting chassis, while greatly reducing noise and vibration The complete mechanism is water-cooled, *including the motor*, permitting installation in unventilated locations

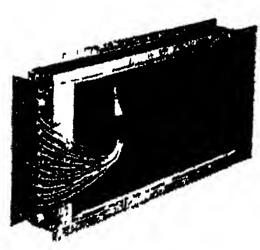
By removing the side plates of the crankcase casting, the entire operating mechanism is accessible for adjustment and service No refrigerant or water lines need be disconnected

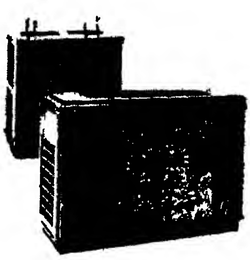


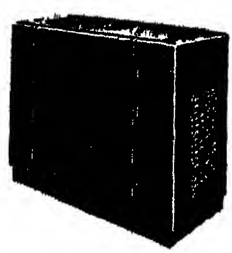
Westinghouse Hermetically-sealed Unit, Type CLS-706 Rated capacity 266,400 Btu/Hr Weight 2,000 lbs

In all, seventeen important improvements have been introduced in these new Westinghouse Condensing Units for air conditioning

NEW EVAPORATORS . . . CONDITIONING UNITS . . . MOBILAIRE







WE Evaporators—A new line of extended surface Freon Evaporators exactly matching in capacities the new Westinghouse Hermetically-sealed Condensing Units

Air Conditioning Units—In wall or ceiling mounted units, or in floor cabinets Available for summer conditioning, winter conditioning, or complete year around service

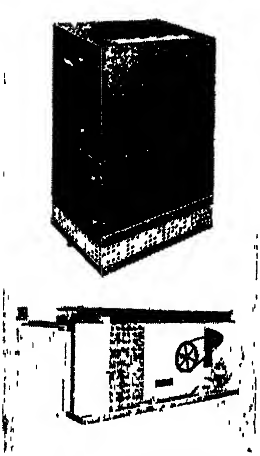
Standard Water-cooled Mobilaire—A self-contained room cooler, the Westinghouse *Mobilaire* provides complete summer air conditioning by cooling, de-humidifying, filtering and gently circulating the air Easily installed in any room where electrical and water connections are available It is powered by the famous Westinghouse Hermetically-sealed Mechanism, which is backed by a 5-year warranty

York Ice Machinery Corporation

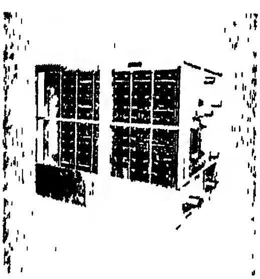
General Offices: **York, Pennsylvania**

Direct Factory Branches in 70 U. S. Cities

Complete Air Conditioning and Refrigerating Systems for maintaining proper atmospheric conditions for industrial processes and human comfort. Available in central and unit systems . . . from fractional tonnage up to any capacity required.



York Air Conditioners



York Standard Dehumidifier with Coils

York Water Cooling System

Air Conditioners—Horizontal and Vertical Models: Capacities up to thirty-five tons, unit air conditioners for horizontal or vertical installation, with or without ducts, with complete or partial automatic control, for complete year-round air conditioning—summer cooling and dehumidification, winter heating and humidification, year-round circulation, filtering and ventilation, including a definite supply of outside air introduced under the positive pressure of the supply fan of the conditioner

Standard Dehumidifiers—With or Without Coils: York air dehumidifiers have been developed to obtain unusually efficient performance when compared with conventional air washer design—accomplished by a refined system of water distribution and improved inlet deflector and eliminator arrangement Can be furnished in all sizes and capacities, seven feet standard length, with and without cooling coils depending upon requirements When evaporator coil surface is included in the spray chamber of an air dehumidifier the usual water cooler furnished with the refrigeration system can be eliminated Both types of air dehumidifiers regularly furnished as a part of central station air conditioning systems being so arranged with fans, heating equipment, automatic controls, air distributing ductwork, and accessories to provide year-round air conditioning

York Water Cooling and Condensing Systems· For dehumidification duty York offers a complete line of water cooling systems built in standard sizes up to eight hundred tons capacity Supplied for both industrial as well as comfort cooling applications

Long Life—made possible by the use of non-ferrous heat transmission surfaces in condenser and cooler—slow speed positive displacement reciprocating compressors

Safety—made possible by the use of the safe refrigerant Freon-12

High Efficiency—made possible only by the careful proportioning of condensing and cooling surface against refrigeration compressors of the vertical single acting reciprocating type Economical operation obtained at all conditions of load through manual or automatic capacity reduction bypass valves built in the compressor

Space Requirements—50 per cent less than steel tube designs—arranged for easy access and maintenance. For applications of direct expansion cooling the water cooling systems are furnished less the water cooler, thereby making possible the same advantages in Condensing Systems

The York Economizer, a combined forced draft cooling tower and refrigerant condenser Replaces usual shell and tube condenser when water rates are prohibitive, where municipal ordinances restrict the use of water or where drainage systems or sewers are inadequate

York Engineering Service: In York branches throughout the world, trained engineers cooperate in preparation of plans for all types of Air Conditioning Applications

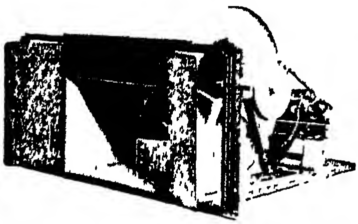
Air Controls, Inc.

Div. of

The Cleveland Heater Co.

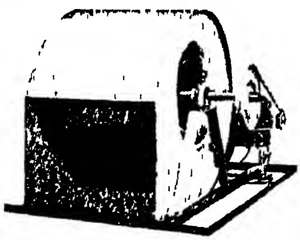
1937 West 114th Street, **Cleveland, Ohio**

Manufacturers of REX AIR-PAK Forced Air Heating and Air Conditioning Units, A. C. Blowers and REX Airate Air Circulators



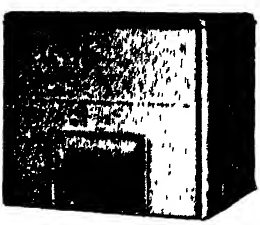
A. C. BLOWER WITH LOUVRES

A high efficiency blower for forced air heating and air conditioning installations Automatic Louvres prevent damage to furnace due to overheating and are essential for safety on oil, gas and coal fired furnaces.

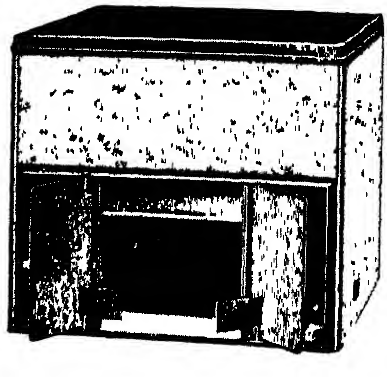


A. C. BLOWER WITHOUT LOUVRES

A. C Blowers are quiet and are designed to deliver proper amount of air whether filters are dirty or clean They occupy small space but deliver large volume of air. Send for bulletin describing them



As shown above the REX AIR-PAK can be obtained without louvres when so desired Send for Simplified Selector Chart and Performance Data

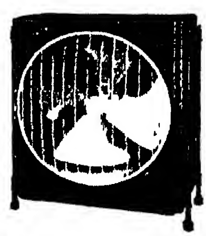


THE REX AIR-PAK

The REX AIR-PAK blower-filter unit for forced air heating and conditioning is encased in an attractive red casing with black top and trimmings Casing comes in panels and can be assembled in 10 min without screws, bolts or slip joints Filters have high cleaning capacity and low resistance Blower is quiet and free from vibration All moving parts are rubber mounted Self-aligning bearings require oiling once a year Casing has removable back which permits cool basement air to be drawn through filters for summer use

The REX AIR-PAK is equipped with the famous Patented Automatic By-pass Louvres which permit unobstructed circulation by gravity when blower is not running

THE REX AIRATE



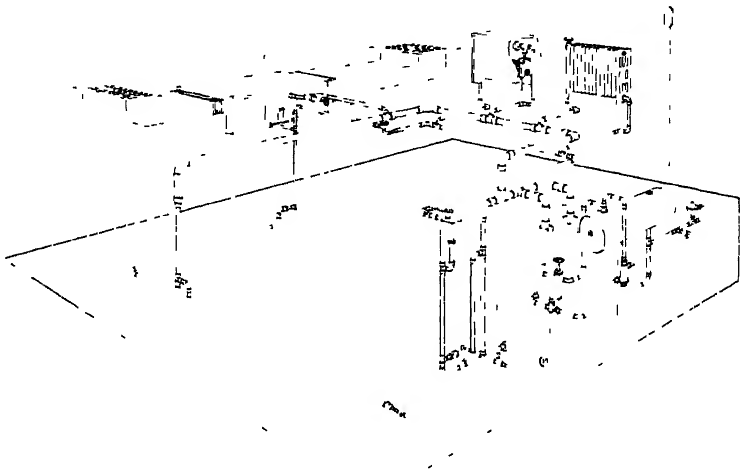
A large volume circulator for commercial or attic ventilation Attractive, quiet, powerful and economical Write for bulletins

AMERICAN RADIATOR COMPANY

DIVISION OF AMERICAN RADIATOR & STANDARD SANITARY CORPORATION

40 West 40th Street, New York, N. Y.

NEW AMERICAN RADIATOR CONDITIONING SYSTEMS



In this new kind of home conditioning presented by American Radiator Company, the heating operates independently of the other functions of air conditioning This permits operation of the conditioning even when the heating is off or of heating alone when no conditioning is necessary It simplifies duct work, too, since ducts do not carry the heating load

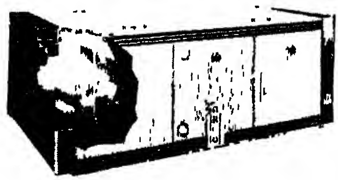
The illustration shows a typical system The Conditioning Unit is suspended from the ceiling of the basement on rubber dampers Air is filtered as it enters, then brought to a comfortable temperature by tempering coils A spray humidifier provides correct moisture content A Sirocco Blower in the unit silently forces the conditioned air through the house

A radiator system—steam, hot water or vapor—provides heat There are new controls, new valves, new vents which improve heat distribution Arco Pipe and Fittings of pure wrought copper connect boiler, radiators and hot water supply system Because of low conductivity they cut down heat loss, they are rustproof, cakeproof, and corrosion resisting

The system is simple to install There is a minimum of ductwork outlets and return grilles being generally recommended only for the first floor, in as few rooms as is advisable Introduced even at only one point, the conditioned air will naturally permeate the entire house

Another advantage of the system is in modernization work as the air conditioning unit is easily added to existing radiator systems

NEW ARCO AIR CONDITIONER



Used with radiator heating, it provides Fresh Air Ventilation, Humidification, Air Cleaning, Circulation It is easily added to existing radiator systems to add the benefits of air conditioning to the benefits of radiator heat

ARCO AIR FILTER

Replacement type Viscous coated Provides 90 deg change in air flow Available in many sizes Uniform construction insures thorough filtering of the air

(See American Radiator Company pages 898-901, 974 and Subsidiaries)

Airtemp Incorporated

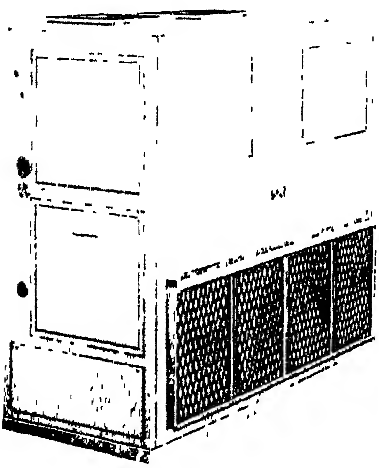
Dayton, Ohio

Subsidiary of

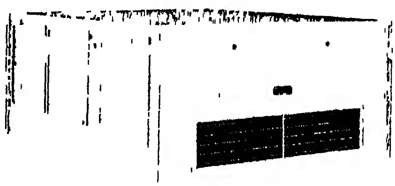
Chrysler Corporation

Complete Air Conditioning Systems for all Types of Buildings

AIRTEMP COMMERCIAL AIR CONDITIONERS



Airtemp Vertical Conditioner



Airtemp Suspended Conditioner

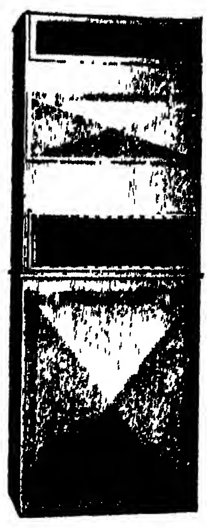
Airtemp Units are especially designed for quiet operation, employing new principles of rubber mounting and acoustic treatment Frames are of welded construction Designed for either free discharge or ducts

AIRTEMP AIR CONDITIONERS
S and V Series Units

Available for either vertical floor mounting or horizontal suspension, are designed to provide all the functions of year round air conditioning or may be furnished to provide only summer or winter conditioning Standard units in capacities to 4500 cfm in various sizes, including automatic temperature and humidity control

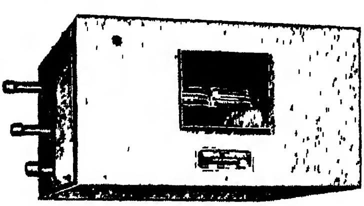
The 3 SC—"All-in-One" Air Conditioner

A heavy duty conditioner of 3 hp capacity, employing Airtemp's new sealed, radial compressor, enclosed in an attractive casing unique in design and assembly Factory assembled, tested and shipped ready for installation, requiring merely simple electric service and water connections in the field for operation



Individual Room Conditioner—Type C-1

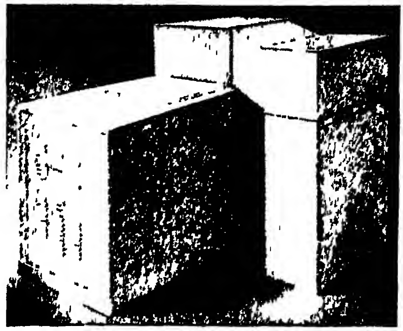
A small compact conditioner designed especially for hotel rooms, apartment hotels or office building conditioning Conditioning surface suitable for direct expansion, Freon or cold water circulation. Equipped for automatic temperature control (Flexibility in its application to new or existing structures, makes this equipment suitable for multiple installation in hotel guest rooms) Nominal 1 Ton 300 cfm capacity

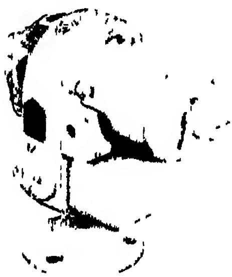


AIRTEMP RESIDENTIAL AIR CONDITIONERS

Airtemp Direct Oil and Gas Fired Conditioners with Self-Contained Package Type Cooling Unit

A compact system that provides complete Winter and Summer Air Conditioning The heating element is designed for high efficiency The cooling unit contains Airtemp's radial sealed compressor of new and compact design, cooling coil, complete automatic controls, etc Capacities to suit normal requirements





Airtemp Oil Burner

A high pressure gun type oil burner used in Airtemp Winter Conditioners and Boiler Burners, or as Conversion Burners Has oversized motor, shielded transformer protection against radio interference, completely enclosed wiring, self-aligning coupling Uses No 3 fuel oil. Adjustable oil pressures, capacities, etc

Airtemp Gas Fired Boilers

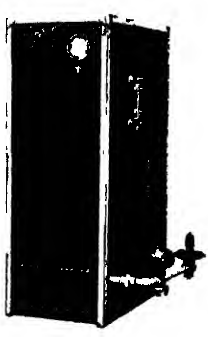
Cast iron external water tube construction Jacket in Stratosphere blue with chrome trim Completely automatic Available in nine sizes Natural, mixed or manufactured gas Approved by A G A

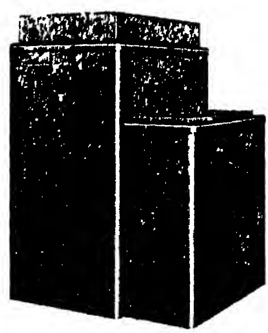
Airtemp Boiler Burner

Cast iron sectionalized wet base construction Jacket of automotive steel, finished in Airtemp blue For steam or water, complete with controls Capacities 600 to 1200 sq ft steam, in three boiler sizes

Airtemp Air Conditioner for Split System Operation

A complete Air Unit consisting of fan, filters, humidifier and steam coil Enclosed in an attractive cabinet, finished in Airtemp blue Can be furnished with cooling coil for summer conditioning





Electrol Incorporated

FINE OIL HEATING EQUIPMENT EXCLUSIVELY SINCE 1918

934 Main Avenue, **Clifton, N. J.**

Conversion Burners, Boiler-Burner Units, Air Conditioning Units

ELECTROL Air Conditioning Units

F. C. Series, Blower Units[1]—Attached to an existing warm air furnace Contains circulating fan and filters Sturdy in construction, simple and economical in operation Fan draws air through an efficient filter that removes all dust and impurities, circulates warm, clean and moist air evenly and without drafts throughout the house Humidifier spray is located in jacket of furnace Amount of moisture injected into air is controlled by Humitrol in living quarters In Summer, Blower Unit can be used to circulate filtered air

Dimensions and Capacity Tables
Electrol Gravity Warm Air Conversion Units—Series F. C.

Model	Height	Width	Length	Outlet Length	Outlet Width	Inlet Length	Inlet Width	Approximate Shipping Weight	Fan Speed			Furnace Data		
									CFM	S P 1/4"	Motor Hp	Furnace Size	Gravity Pipe Area Rating Sq In	Btu per Hour
FC 1	31½"	28"	34½"	14½'	12'	24'	18"	170	1000	555	1/6	22'-24"	450-550	61000-75000
FC 2	34½"	30½"	34½"	14½"	13⅝'	24"	18"	250	1400	530	¼	25"-23'	700-820	95000-112000
FC 3	37½"	33½"	40"	17½"	15⅜"	30"	18"	350	2000	455	⅓	29"-30'	990-1150	135000-157000
FC 4	37½"	33½"	42½"	19½"	18⅝"	30"	24"	400	2500	375	⅓	32'-34"	1270-1400	172000-190000

S-S Series, Split-System Air Conditioner[1]—Used in conjunction with existing boiler Consists of circulating fan, air filters, humidifier and heating coil Radiators are removed from rooms to be conditioned and duct work is installed The Unit circulates moist, warm and clean air efficiently No danger of drafts Blower unit can be used for ventilation in Summer

Capacity Table—Electrol Split System Air Conditioner Units - Series SS

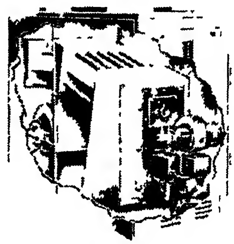
Model	Heating Capacity Btu per Hour		Air Capacity				Btu per Hour Cooling Capacity
	Hot Water 180°	Steam 1 lb Pressure	CFM	WGS	Fan Speed	Motor H P	50° Water
SS 1	86000	87500	1120	⅜"	716	¼	22000 6GPM
SS 2	155000	162000	2000	⅜"	558	⅓	40500 12GPM
SS 3	206000	217000	2700	½"	656	½	54400 10GPM
SS 4	240000	307000	3600	½"	630	¾	61400 12GPM

Dimensions

Model	Height	Width	Length	Outlet Length	Outlet Width	Inlet Length	Inlet Width	Approximate Shipping Weight
SS 1	48½	41¾	35½	26	10½	26	12	600
SS 2	60½	58¾	35½	26	20	26	20	900
SS 3	60½	56⅜	51½	40	20	40	20	1200
SS 4	60½	63⅝	59¼	60	21¾	60	21½	1500

Height of outlet and inlet collars for Duct Connections 1½ in
[1] Provision made for addition of de-humidifying and cooling equipment

See also Page 927



D. F. Series, Direct Fired Furnace Conditioner*—For new construction or homes that are being remodeled Heats air and automatically maintains temperature for which it is set Thoroughly cleanses the air by means of efficient renewable filters Positively circulates the air by means of a motor operated blower of ample capacity Humidifies by means of patented spray device Fired by Electrol Oil Burner Guaranteed 80 per cent efficient when operated at rated capacities Low grille temperatures, low velocity air Beautifully jacketed

Capacity Table—Electrol Warm Air Furnaces—Series D. F.

Model	Btu per Hour at Bonnet	Btu per Hour at Grilles	Air Temp at Bonnet	Oil Rate Lbs 1 Hr	Heating Surface			Air Capacity			
					Primary	Secondary	Total	CFM at 70°	S P	Fan Speed	Motor H P
DF 1	120,000	100,000	145-165	8 65	24	54	78	1450	1/4"	603	1/4
DF 2	150,000	125 000	145-165	9 70	24	57	81	1650	1/4"	500	1/3
DF 3	175,000	145,000	145-165	11 90	24	61	85	1850	1/4"	513	1/2
DF 4	225,000	185 000	145-165	14 40	26	86	112	2450	1/4"	423	3/4

Dimensions

Model	Height	Length	Width	Outlet Width	Outlet Length	Inlet Width	Inlet Length	Approximate Shipping Weight
DF 1	56½	92½	33½	24	30	24	20	1400
DF 2	56½	92½	33½	24	30	24	24	1600
DF 3	56½	92½	33½	24	30	24	24	1800
DF 4	60½	97½	35½	24	32	24	28	2000

Height of outlet and inlet collars for Duct Connections 1½ in

Series SSJ, Room Conditioner*—Will circulate, filter and humidify the air in one or more rooms Contains blower unit, spray type humidifier and extended surface heating coil Takes the place of an existing radiator

RC Series, Room Cooling Unit—For cases where Summer conditioning only is desired Consists of water cooled unit for cooling and de-humidifying, filters and fan Refrigerating mechanism and fan operate independently of each other

Capacity Table (Per Hour) Series SSJ

Heating and Humidifying	Cooling
Steam 1 lb Press 39,200 Btu 4 gals	Air Capacity 350 CFM Fan Speed 1140 RPM Motor H P 1/30
Hot Water 120° 14,200 Btu 1 gal 150° 21,600 Btu 2 gals 180° 31,000 Btu 3 gals	Water at 40°(2 gpm) 8000 Btu Water at 50°(3 gpm) 4800 Btu

Electrol Unit Room Cooler—Series R C.

Model	Height	Length	Width	Compressor Speed	Btu Extn Total Net	Motor, Hp	CFM
RC1	34	36	18	650	25031	1/4	480-520
RC2	34	36	18	500	6379	1/2	500-575
RC2A deluxe	33	36	19	680	7045	1/2	600
RC3	37	45	21	695	9949	3/4	1600
RC4	37	45	21	465	12,980	1	1600

Cooling requirements vary with exposure number of occupants and other local conditions and should be correctly calculated in every case to determine size of unit required

*Provision made for addition of de-humidifying and cooling equipment

The Fox Furnace Company

Elyria, Ohio

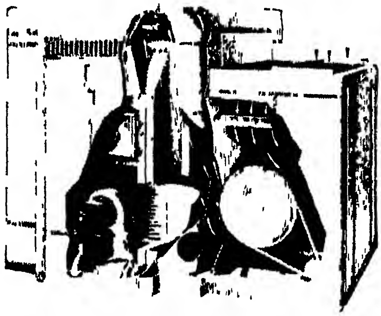
Division of American Radiator and Standard Sanitary Corporation

Sales Offices:
CLEVELAND, OHIO
NEW YORK, N Y
BOSTON, MASS
SAN FRANCISCO, CALIF

SUNBEAM
AIR CONDITIONING

Jobbers in principal cities. Engineering layouts provided by factory

Sunbeam Oil Fired Air Conditioning Units Series No. 200 and 400



Series No 200 interior view, showing attractive cabinet, integral oil burner, inner casing, heating element, humidifier (at top of heating element), blower, motor and filters

Like gas and coal fired Sunbeam Units shown herein, this series is designed to air condition residences and small stores, churches and other buildings Each Air Conditioner heats, filters, humidifies and circulates the air in winter and provides circulation of filtered air in summer Mechanical cooling can be added

Exterior Cabinet—20 gauge furniture steel, modern design, bolts and screws concealed, finished in two tones of green glossy enamel

Heating Element—7 gauge boiler plate, riveted and welded. Radiator made of 12 gauge boiler plate

Blower and Motor—Blower is of double inlet type with forward curved, closely spaced blades Moves large volume of air at low speeds Equipped with self-aligning bearings

Motor is of Capacitor type and is equipped with fuse device to prevent damage from overloading

Humidifier—Either spray or drip type humidifier can be furnished Spray humidifier is regulated by a Humidistat

Minneapolis-Honeywell Controls which regulate temperature and humidity are provided

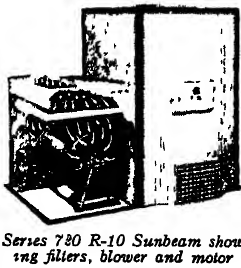
Filters—Sunbeam Filters provide high cleaning efficiency and long life Are treated with large amount of adhesive without clogging the air passages

Oil Burner Series 200—Burner is integral part of this unit and is placed inside cabinet as illustrated Coordinated Air Conditioner and burner results in high efficiency and economical fuel consumption

Series 400—Burner is **not** furnished with this unit A space 40 in wide, 22⅛ in high provided in rear will accommodate practically any gun type conversion oil burner

See next page for Capacities and Dimensions

Series 720 R-9 and 720 R-10



Series 720 R-10 Sunbeam showing filters, blower and motor

This series, which performs the same air conditioning functions as the series described above, is moderately priced, designed for small and average size homes It is compact and requires little space

The 720 R-10 is for installation in the basement.

The 720 R-9 is specially designed for homes or small buildings having no basement.

Exterior Cabinet—Finished in crystalline baked enamel

Heating Element—Heavy boiler plate steel, riveted and welded Combustion chamber is of 8 gauge steel, radiator of 12 gauge

Inner Casing—Galvanized iron Air circulating between inner and outer casing keeps cabinet cool

Oil Burner—Rotary wall flame type, completely and ingeniously assembled at the factory in cast iron frame, ready for insertion in the heating element.

Equipment—Blower, Motor, Filters, Controls and Humidifiers same as in Series 200

Capacities—Blower-Motor-Filter Specifications—Series No. 200—400—720-R-9—720-R-10

No	†Cap at Reg (Oil-Fired)	Blower No	*Max C F M Required at 65° F	*Approx Blower R P M	Dia of Blower Wheel	No of Blowers	Motor H P	Motor R P M	Number Filters
224 and 424	177,000 to 186,000	1-15	1827	380	15"	1	¾	1725	3 (16" x 25")
224 and 424	156,000 to 176,000	1-15	1725	375	15"	1	½	1725	3 (16" x 25")
224 and 424	124 000 to 155 000	1-15	1522	350	15"	1	⅓	1725	3 (16" x 25")
224 and 424	100,000 to 123,000	1-12	1218	482	12"	1	¼	1725	3 (16" x 25")
720-R-9	100,000	1-9	986	655	9¼"	1	¼	1725	2 (16" x 25")
720-R-9	75,000	1-9	740	576	9¼"	1	¼	1725	2 (16" x 25")
720-R-9	50,000	1-9	493	510	9¼"	1	¼	1725	2 (16" x 25")
720-R-10	100,000	1-10	986	515	10⅝"	1	¼	1725	2 (16" x 25")
720-R-10	75,000	1-10	740	486	10⅝"	1	¼	1725	2 (16" x 25")
720-R-10	50,000	1-10	493	480	10⅝"	1	¼	1725	2 (16" x 25")

*Air heated from 65° to 165° increases 19 per cent in volume Therefore warm air ducts should have capacity for 19 per cent greater C F M than listed above R P M based on ¼ in S P in Series No 200 and 400 and on 3/16 in S P in Series No 720-R

†Combustion rate of oil burner must conform to the heating requirements of the installation

Dimensions—Series No. 200—400—720-R-9—720-R-10

No	Overall Width	Overall Depth	Ht Heating Compartment	Ht Blower Compartment	Air Discharge Opening	Air Intake Opening
224 and 424	*76¼"	†57⅝"	58⅛"	43¾"	24" x 26"	13" x 42"

*Allow 2 ft at side of blower compartment for removal and replacement of filters
†Allow 2 ft in front for opening of doors †Allow 2 ft in rear for flue outlet

720-R-9	42"	▲†67⅝"	60"	18¾"	18" x 18"	2 (4½" x 24½")

†Allow 24 in at front for access to burner Allow 24 in at rear for access to blower, motor and filters

720-R-10	*67"	†▲40⅛"	60"	37¼"	18" x 18"	12" x 35½"

†Allow 24 in in front for access to burner ▲Allow 17 in in rear for smoke outlet *Allow 24 in on side for removal of filters

Sunbeam Coal Fired Air Conditioning Units

Series No 80 View of interior showing blower, motor, filters and spray humidifier Arrows indicate air travel

Coal fired models have heavy, sturdy heating elements designed to extract a maximum of heat from the fuel burned They are long lived and require a minimum of attention and servicing The duplex type of grate is standard equipment

Series No. 80

Exterior Cabinet—Finished in attractive crystalline baked enamel

Heating Element—7 gauge boiler plate, riveted and welded Radiator made of 12 gauge boiler plate

Blower and Motor—Blower is of double inlet type with forward curved, closely spaced blades Moves large volume of air at low speeds Equipped with self-aligning bearings

Motor is of Capacitor type and is equipped with fuse device to prevent damage from overloading

Humidifier—Either spray or drip type humidifier can be furnished Spray humidifier is regulated by a Humidistat

Minneapolis-Honeywell Controls which regulate temperature and humidity are furnished

Filters—Sunbeam Filters provide high cleaning efficiency and long life Are treated with large amount of adhesive without clogging the air passages

Stoker Fired and Oil Fired Models—The Series No 80 is available in stoker and oil fired models *See table on following page for capacities*

THE FOX FURNACE COMPANY

SUNBEAM AIR CONDITIONING

Sunbeam Coal Fired Air Conditioning Units (Con'd.)

Series No. 20

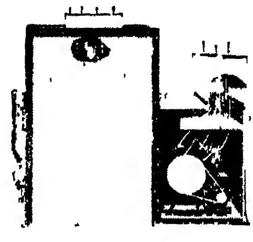
This series meets the demand of the average home owner who desires the benefit of air conditioning and is seeking a moderately priced unit to install in a new home or to replace an inefficient furnace Features that assure years of satisfactory and healthful service are incorporated in this unit

Exterior Cabinet—Finished in crystalline baked enamel

Heating Element—Cast iron Has large area of heating surface Is long-lived and gas-tight

Inner Casing—Galvanized iron Air circulating between inner and outer casing keeps cabinet cool

Equipment—Blower, Motor, Filters, Humidifiers and Controls same as in Series 80



Interior view of Series No 20 Unit showing spray humidifier, filters, blower and motor

Capacities—Blower-Motor-Filter Specifications—Series No. 80 and 20

No	Btu Capacity at Register		Blower No	Coal Hand Fired		Oil or Stoker Fired		Diam of Blower Wheel	Number Motor Blowers	Motor H P	Motor R P M	Number Filters
	Coal Hand Fired	Oil Fired or Stoker Fired		‡C F M Required at 65° F	‡Approx Blower R P M at ¼″ S P	‡C F M Required at 65° F	‡Approx Blower R P M at ¼″ S P					
2280	103,000	117,000	1-12A	1112	401	1260	430	12″	1	¼	1725	2 (20″ x 20″)
2480	118 000	134,000	1-12A	1269	430	1439	459	12″	1	¼	1725	2 (20″ x 20″)
2780	155,000	176 000	1-15A	1666	367	1890	380	15″	1	⅓	1725	3 (16″ x 25″)
3080	182 000	207 000	1-18A	1962	295	2225	300	18″	1	½	1725	4 (16″ x 25″)
3480	222,000	252,000	1-21A	2385	245	2705	255	21″	1	¾	1725	4 (16″ x 25″)
4020	78 000	88,000	1-12A	838	373	951	390	12″	1	¼	1725	2 (20″ x 20″)
4420	95,000	108,000	1-12A	1020	401	1158	430	12″	1	¼	1725	2 (20″ x 20″)
4820	111 000	126,000	1-12A	1195	430	1356	459	12″	1	¼	1725	2 (20″ x 20″)
5220	133,000	150,000	1-12A	1426	459	1618	487	12″	1	¼	1725	3 (16″ x 25″)
5620	153,000	173 000	1-12A	1644	487	1865	516	12″	1	⅓	1725	3 (16″ x 25″)

‡Air heated from 65 to 165 F increases 19 per cent in volume Therefore warm air ducts should have capacity for 19 per cent greater C F M than listed above

Dimensions—Series No. 80

No	†▲Overall Width	Height Heating Compartment	Height Blower Compartment	▲Width of Blower Compartment	*Overall Depth	Air Discharge Opening	Air Intake Opening
2280	74″	60″	37¼″	28″	45⅝″	20″ x 20′	12″ x 40″
2480	76″	60″	37¼″	28″	48⅛″	22′ x 22′	12′ x 40′
2780	83½″	67″	42″	29″	53⅝″	25′ x 25″	12′ x 48½″
3080	90″	67″	43″	32½″	57⅛″	27″ x 27′	17″ x 54⅝′
3480	98½″	67″	48″	37″	61″	30′ x 30′	17″ x 58½″

*Smoke Pipe Tee extends out approximately 24 in additional in Nos 2280 and 2480, and 32 in in larger sizes †Includes width of angle iron base ▲Allow clearance at side equal to width of blower compartment for removal of filters, blower and motor

Dimensions—Series No. 20

No	*Overall Width	*Overall ▲Depth	Height Heating Compartment	▲Depth of Blower Compartment	Height Blower Compartment	Air Intake Opening	Air Discharge Opening (if Plenum Chamber is Used)
4020	42″	66⅜″	60″	28″	37¼″	12′ x 40′	18″ x 18″
4420	46⅛″	68⅜″	60″	28″	37¼″	12′ x 40′	20″ x 20″
4820	48⅞″	72⅜″	60″	28″	37¼″	12′ x 40′	22′ x 22″
5220	54″	77⅜″	64″	29″	42″	12″ x 48½″	23′ x 23″
5620	58″	80⅜″	64″	29″	42″	12′ x 48½″	24″ x 24″

*Includes width of angle iron base ▲Allow clearance in rear equal to depth of blower compartment for removal of filters, blower and motor.

THE FOX FURNACE COMPANY

SUNBEAM AIR CONDITIONING

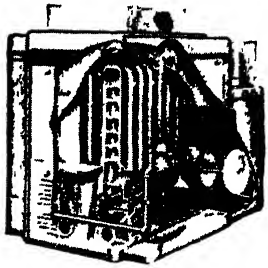
Sunbeam Gas Fired Air Conditioning Units Series D

This series, which performs the same functions as all other Sunbeam Air Conditioners, has the same attractive appearance as the Series 200 All bolts, screws, valves, pipes and wiring are located inside the cabinet out of sight, yet are readily accessible

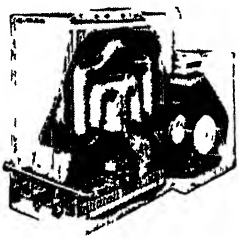
Exterior Cabinet—20 gauge furniture steel Modern in design and finished in two tones of green glossy enamel

Heating Elements—Cast iron Long-lived and leak-proof. Provide unusually high efficiency

Equipment—Blower, Motor, Filters, Humidifiers and Controls same as in Series 200



Interior view of Series D showing heating element, burner, thermostatic pilot, spray humidifier, blower, motor and filters



Interior view of Series M showing heating element, burner, valves, inner casing, blower, motor and filters

Series M

This series is designed and priced for small and average size homes Although modern in appearance and efficient in operation it is substantially lower in cost than the Series D

Exterior Cabinet—Finished in green crystalline enamel with dark green glossy enamel trim

Heating Elements—Made of 16 gauge steel, unique design provides ample heating surface and long fire travel.

Equipment—Blower, Motor, Filters, Humidifiers and Controls same as in Series 200

Rating—Blower-Motor-Filter Specifications—Series D and M

No	A G A Btu Input per Hour	Btu Capacity at Registers	Blower No	*Blower C F M at 65 F	*Approx Blower R P M at ¼″ S P	Diameter of Wheel	Motor H P	†Number Filters
D-2	140,000	107,100	1-12A	1016	401	12″	¼	2
D-3	210 000	160,650	1-15A	1526	344	15″	⅓	3
D-4	280 000	214 200	1-18A	2032	300	18″	½	4
D-5	350,000	267,750	2-15A	2540	344	15″	½	4
M-2	80,000	61,200	1-10A	580	550	10⅝″	¼	2
M-3	120,000	91,800	1-12A	870	380	12″	¼	2
M-4	160,000	122,400	1-12A	1160	475	12″	¼	3
M-5	200,000	153,000	1-15A	1450	340	15″	⅓	3

*Air heated from 65 to 165 F increases 19 per cent in volume Therefore, warm air ducts should have capacity for 19 per cent greater C F M than listed above †Size of filter, 16″ x 25″

Dimensions—Series D and M

No	Width of Heating Compartment	Overall Depth	Overall Height	Width of Blower Compartment	Height of Blower Compartment	Depth of Blower Compartment	Air Discharge Opening	Air Intake Opening
D-2	44½″	▲75⅜″	58″	39½″	41⅞″	28″	16″ x 22″	12⅜″ x 35¼″
D-3	56″	▲76⅜″	58″	48¼″	41⅞″	29″	16″ x 34″	12⅜″ x 44″
D-4	70½″	▲79⅝″	58″	65⅝″	41⅞″	32⅛″	16″ x 45″	12⅜″ x 59¼″
D-5	82″	▲76⅜″	58″	86″	41⅞″	29″	16″ x 56½″	12⅜″ x 70″

▲Allow clearance in rear equal to depth of blower compartment for removal of filters, blower and motor Allow 24 inches clearance in front for removal of baffles

No	Width of Heating Compartment	Overall Depth	Overall Height	Width of Blower Compartment	Height of Blower Compartment	Depth of Blower Compartment	Air Discharge Opening	Air Intake Opening
M-2	20¼″	*69⅛″	54″	35⅝″	39″	29⅛″	16″ x 15″	10″ x 20″
M-3	28¾″	*69⅛″	54″	38⅝″	39″	29⅛″	16″ x 22½″	10″ x 30″
M-4	37¼″	*69⅛″	54″	48⅝″	39″	29⅛″	16″ x 30″	10″ x 40″
M-5	45¾″	*69⅛″	54″	48⅝″	39″	29⅛″	16″ x 37½″	12″ x 42″

*Allow clearance in rear equal to depth of blower compartment for removal of filters, blower and motor

Delco-Frigidaire Conditioning Division

General Motors Sales Corporation

Dayton, Ohio

PRODUCTS OF GENERAL MOTORS

Automatic Heating and Air Conditioning Equipment for Residential and Commercial Applications

Delco-Frigidaire air conditioning equipment is backed by the research, engineering and production facilities of General Motors, whose products have earned their reputation for quality and value throughout the world

Each product carries the specialized experience of the manufacturer in building precision equipment Careful manufacturing methods, backed by actual testing under operating conditions with rigid test supervision, insure that each unit will meet the stringent requirements set for all Delco-Frigidaire equipment

In the field of summer air conditioning, basic research covering the requirements of equipment under operation, plus a wide experience in the refrigeration field and the years of experience of General Motors in building piston type equipment are all combined in the design and construction of Delco-Frigidaire products

Freon, the safe refrigerant, was developed by the combined research of General Motors, Frigidaire, and DuPont engineers in their search for a refrigerant which would fit the requirements of air conditioning It is now almost universally used, having answered many of the problems of the industry

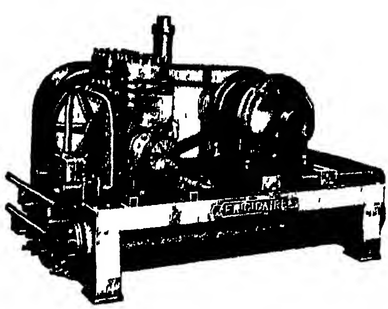
In the heating field, General Motors has the advantage of the years of study and research entered into as the outstanding organization in the combustion and application of liquid fuels

All Delco-Frigidaire equipment is built to be easy to install and operate, simple in construction and design, and quiet in operation, not only when new but after years of service It is backed by the stability and reputation of a responsible manufacturer, and has years of proven performance in installations of almost every type and under all conditions

If you have a special problem or require specific information on any of the Delco-Frigidaire products, see the classified section of your telephone directory for the name and address of your local distributor You will find it listed under "Air Conditioning—Delco-Frigidaire"

DELCO-FRIGIDAIRE CONDENSING UNITS

Delco-Frigidaire Condensing Units are designed for air conditioning application, and are matched to Delco-Frigidaire evaporators and unit air conditioners to give balanced operation of the air conditioning system as a whole



FW-85 10 HP Condensing unit

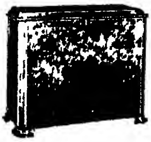
Their design and construction have been governed by the basic idea that dependability and economy are two of the most important requirements in this type of equipment—dependability because it must function day after day with a minimum of attention, and economy because the owning and operating costs of an air conditioning system over a period of years may represent a large investment as compared to the original cost of installation

A few of the typical features which give Delco-Frigidaire condensing units their added value are Specially treated compressor body castings, drop forged, case hardened alloy steel crankshaft; special machining of pistons for precision fit, self-oiling Durex bearings, and centrifugally cast babbitted connecting rod bearings

Condensing units are available in water-cooled models from 0 37 to 40 tons capacity, and in air-cooled models from 0 34 ton to 2 tons, rated at standard conditions

Call your local Delco-Frigidaire distributor for specifications covering this and other Delco-Frigidaire air conditioning equipment

DELCO-FRIGIDAIRE REMOTE AIR CONDITIONING UNITS



Typical floor type remote air conditioning unit

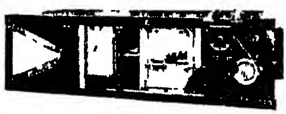
Delco-Frigidaire remote type air conditioning units are designed for use singly or in multiple with a Delco-Frigidaire condensing unit located in the basement, a nearby closet or storeroom, or other convenient location

Floor mounted types are available in ½ and 1 ton capacities for installation in smaller rooms or offices, or in multiple in larger areas Both may be equipped with heating coils for year around service, and have a convenient dial thermostat provided as standard equipment for control of room temperatures when cooling

Ceiling suspended units are available in 1½ and 3 ton sizes for installation in locations where floor space is limited The larger 3 ton size may be equipped with a heating coil for winter service, and a special model is available with a reheating coil for application in industry where dehumidification only is required

A special series of concealed suspended units, designed for installation in a closet space, is available in ½ and 1 ton capacity This type is particularly adapted to installations in hotel guest rooms, hospitals, apartments, etc

DELCO-FRIGIDAIRE CENTRAL PLANT AIR CONDITIONING EQUIPMENT



Delco - Frigidaire Central plant equipment for installations employing a duct distribution system includes a series of units factory designed and built for summer or year around air conditioning service

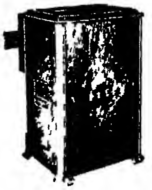
Through selection of coil and fan equipment, capacity may be varied to meet the requirements of a particular installation, and filtering, heating and humidification added to cooling and dehumidifying as required

These units are available in sizes of approximately 5, 10 and 20 tons capacity

Delco-Frigidaire evaporators for larger built-up central plant installations are available in a wide range of sizes and capacities They are designed to give a maximum of cooling capacity in a minimum of space Low air resistance and low pressure drop in the evaporator coils mean greater operating efficiency and lower cost

Delco-Frigidaire evaporative condensers are available for use on installations in communities where there is a scarcity of water, or where water for cooling the condenser is available only at high rates They are available in capacities of 3, 10, 20 and 40 tons

DELCO-FRIGIDAIRE SELF-CONTAINED AIR CONDITIONERS



Typical floor type self-contained unit conditioner, approximately ¾ ton capacity

The Delco-Frigidaire Model SC-80 self-contained air conditioner provides cooling, dehumidification and air circulation, to which may be added filtering and provision for ventilation air from outside if desired

As shown, the unit is enclosed in an attractive steel cabinet with a butt grained walnut finish, designed to harmonize in any setting The water-cooled condensing unit installed in the lower section of the cabinet is carefully mounted, balanced and insulated to assure quiet operation A convenient dial is provided for thermostatic control of room temperatures

Capacity of the SC-80 is approximately ¾ ton at standard rating conditions

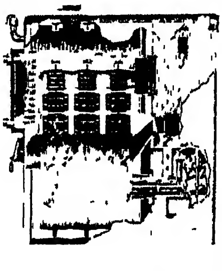
A large model self-contained unit, SC-301, is adapted to installations in small stores, offices, etc , and particularly where there is a peculiar lease or alteration problem It can readily be moved to another location by disconnecting power, water, and drain lines If desired, the unit may be installed behind a partition, with only the outlet grilles exposed

The condensing unit is 3 hp water-cooled, with a capacity of approximately 3 tons Thermostatic control of room temperatures is provided as standard equipment

DELCO-HEAT BOILER UNITS

In the Delco-Heat Boiler unit, the Delco-Heat Oil Burner or Gas Burner is coordinated with a boiler of special design for steam, hot water or vapor-vacuum heating systems. Provision for year around domestic hot water service is standard equipment on all DL models and is available as optional extra equipment on DH models.

The combustion chamber of the Delco-Heat Boiler unit is water backed on both sides and bottom. The volume of the chamber is ample to allow complete combustion of fuel before gases enter the upper passages.

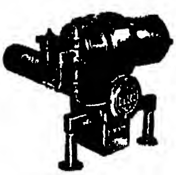


Section of Delco-Heat Boiler unit

The Delco-Heat Boiler employs the Im-Pak-Tor principle of heat transfer. Each section is honeycombed with a multitude of water-backed fins cast in staggered arrangement. When boiler sections are tightly fitted together, a series of passes is formed. Hot combustion gases must travel through at least 7 of these water-backed passes on their way to the chimney. This assures maximum heat transfer and low fuel cost.

Oil fired models are available in four sizes, from 150,000 to 414,000 Btu/hour and gas fired units in two sizes, 150,000 and 252,000 Btu/hour.

DELCO-HEAT OIL BURNERS



Delco-Heat Oil Burner

The Delco-Heat Oil Burner is of the mechanical atomizing type, featuring one moving part to eliminate wear and assure long lived, dependable performance.

The Delco-Heat "Thin-Mix" fuel control acts to prevent the flow of oil to the nozzle until the quantity of air being admitted is sufficient to assure proper combustion, and oil pressure is sufficient to atomize oil completely. At the end of the period of operation, or in case the oil flow is interrupted at any time, the control also acts to shut off the flow at once, preventing a dribble of oil at the nozzle which might cause carbon formation.

The Delco-Heat Oil Burner is available in a range of 4 sizes burning from 1.0 to 30.0 gal per hour.

DELCO-HEAT CONDITIONAIR

The Delco-Heat Conditionair is a completely harmonized automatic winter air conditioning system, and is available in either oil or gas fired models. With the addition of a condensing unit and evaporator coils the system is easily converted to a year around installation provided proper care has been given the selection of outlet locations and duct sizes to enable them to perform their dual functions.

The combustion chamber of the Conditionair is designed to give ample space to allow complete combustion of fuel before gases enter the upper passes of the unit. Long travel assures maximum opportunity for the gases to give up their heat to the air stream. The Multi-Path feature divides the air passing through the unit into 7 separate streams to provide even, rapid heating of the air as it passes over the heat transfer surfaces. Dust is removed by viscous type filters, and humidification is provided by a cascade type humidifier.

Section of Delco-Heat Conditionair showing air travel

Burner, operating controls, and blower motor are readily accessible inside the doors of the cabinet, but are outside the high temperature area of the unit.

The Conditionair is available in 3 sizes for oil firing and 2 sizes in gas fired models.

The Delco-Heat domestic hot water heater, also available in gas or oil fired models, may be used in connection with Delco-Heat Conditionair installations to provide year around domestic hot water service.

DELCO-HEAT GAS FIRED UNITS

In the Delco-Heat Conditionair or the Boiler unit, the Delco-Heat Gas Burner projects a long, sweeping luminous gas flame into the combustion chamber from specially designed refractory ports.

This adapts to residential heating the type of luminous gas flame long used in the steel and other heat treating industries, giving the advantages of the more efficient radiant heat transfer inherent in the luminous flame, around which both the Conditionair and Boiler units have been designed.

Gilbert & Barker Mfg. Company

Springfield, Massachusetts

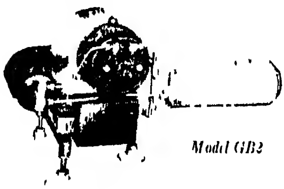
Branch Offices

New York City	Hempstead, N Y	Toronto	Vienna
Chicago, Ill	Detroit, Mich	London	Sydney
Rochester, N Y	Pittsfield, Mass	Paris	Buenos Aires

Gilbarco
OilHeating-AirConditioning

GILBARCO FLEXIBLE FLAME OIL BURNERS

"Gilbarco" Flexible Flame Oil Burners are of the pressure atomizing type, suitable for residential and commercial use in steam, hot water, vapor, or warm air heating systems Their operation is fully automatic with controls and safety devices integral with the system Due to the flexibility of the flame, the "Gilbarco" Burner is enabled to more completely fill the fire box with a heat-giving radiant flame



Model GB2

Each "Gilbarco" Burner combines the following definite advantages (1) Flexible Flame with "Econ-O-Flex" controlled combustion—assuring tailor-made application (2) Forced draft insuring complete combustion (3) Radiant type flame gives great heat output (4) Constant electric ignition (5) Separate air and oil controls insuring complete combustion (6) Oil filter in oil line assuring clean oil at all times (7) Radio interference eliminator (8) Burner installed outside boiler insures long life, easy inspection and service and general operating efficiency (9) Quiet operation no gears, belts or noisy mechanisms

SPECIFICATIONS OF AND CAPACITIES FOR 60 CYCLE MOTOR BURNERS

(Write to us for capacities of D C and odd cycle motor burners)

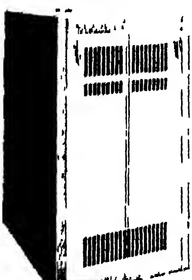
MODEL NUMBER	GB1	GB2	GB3	GB4	GB5
Max oil Capacity per hour U S Gallons	3 00	6 00	11 0	15 00	25 00
Total Steam Radiation (Radiation, piping, and pickup)	1150	2350	4400	6000	10 000
Total Hot Water Radiation (Radiation, piping, and pickup)	1840	3760	7040	9600	16 000
Ignition	Continuous electric single transformer	Continuous electric single transformer	Continuous electric single transformer	Continuous electric two transformers	Continuous electric two transformers
Motor (size)	1/12 H P	1/8 H P	1/6 H P	1/4 H P	1/3 H P
Controls	Stack mounted Protectorelay and Thermostat	Stack mounted Protectorelay and Thermostat	Protectorelay Protectostat and Thermostat	Protectorelay Protectostat and Thermostat	Protectorelay Protectostat and Thermostat
RPM	1750	1750	1750	1750	1750

GILBARCO HEAVY OIL BURNERS

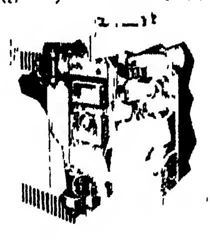
Complete Range for Industrial Heating and Power Application

GILBARCO SERIES "A6" BOILER BURNER UNITS

Three models for both steam and hot water in the "A6" series which range in capacity from 518 ft to 860 ft of steam radiation (gross) and 876 ft to 1376 ft of hot water (gross) The boiler, especially designed for oil fuel, is of cast iron, which can be molded into effective types of heat-absorbing surfaces "A6" series models are completely automatic and the high efficiencies which are attained in the boiler are due to the fact that the heat gases are compelled to be in direct contact with the heat-absorbing surfaces at all times All series "A6" Gilbarco Boiler Burner Units are designed to include Built-in hot water coil and Aquastat, Low-water Cut-off on steam systems, Pre-cast Refractory combustion chamber, Pressuretrol on steam systems, Surface Aquastat on hot water systems



Outside view of the "A6" Series Gilbarco Boiler Burner Unit Furnished in two-tone green with chrome trim

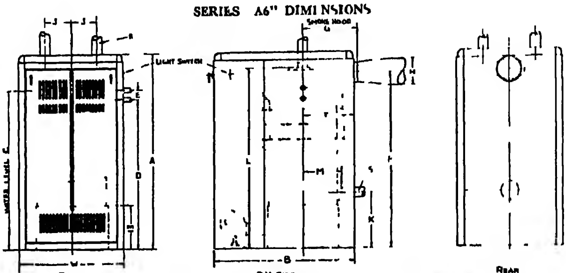


Cutaway view showing construction and gas travel Toggle switch on outside of cabinet controls lighting of burner compartment Gauges easily read through ventilator openings

* GILBARCO SERIES "A6" RATINGS

Unit Number	Square Feet Steam Radiation: Gross	Net: Without Domestic Water Coil	Net: With Standard Coil	Net: With Special Coil	Square Feet Hot Water Radiation: Gross	Net: Without Domestic Water Coil	Net: With Standard Coil	Net: With Special Coil	Domestic Water Coil Capacities, Gallons of Water in Three Hrs 100°F Temperature Rise — Steam Boilers at 180°F Average Temp: Standard Coil	Steam: Special Coil	Hot Water Boilers at 150°F Average Temp: Standard Coil	Hot Water: Special Coil
A6—19	548	350	321	291	876	560	539	514	10	80	20	40
A6—20	700	450	416	391	1120	720	692	671	45	80	22 5	10
A6—22	860	550	515	491	1376	880	853	834	50	80	25	10

SERIES "A6" DIMENSIONS



	A	B	C	D	E	F	G	H	J	K	L	M	W	R	S	T
A6—19	57½	39[illegible]	47½	44⅛	2⅝	51½	16[illegible]	8	7[illegible]	16⅞	54¼	11⅝	31½	2¼	1	11¼
A6—20	57½	41¾	47½	44⅛	2⅝	51[illegible]	17⅝	9	8[illegible]	16⅞	51¼	12¾	31	2[illegible]	4	11¼
A6—22	57½	44¼	47¼	44⅛	2⅝	51[illegible]	18[illegible]	9	9[illegible]	16⅞	54¼	13⅝	36	2[illegible]	4	15¼

GILBARCO SERIES "B" BOILER BURNER UNITS

Outside view of the "B" Series Gilbarco Boiler Burner Unit Furnished in two-tone green with chrome trim

There are five models for both steam and hot water systems in the "B" series which range in capacity from 780 to 2030 sq ft of steam (gross) and 1248 to 3246 sq ft of hot water (gross) Like the "A6" series units, the boilers of the "B" series are of cast iron construction insuring peak efficiency and great durability The specially designed boiler combines the following important efficiency factors (1) Extended fin type heating surfaces (2) Low draft loss (3) Large combustion chamber (4) Quick steaming (5) Water-backed combustion chamber (6) Ground joints between sections (7) Unique gas travel (8) Positive internal water circulation (9) Large steam liberating surface (10) Compact size

Cutaway view showing construction and gas travel Gages visible at all times through attractive escutcheon plates

* GILBARCO SERIES "B" RATINGS

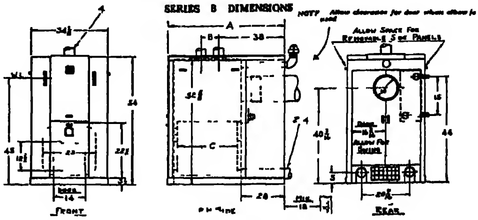
Unit Number	Square Feet Steam Radiation: Gross	Net: Without Domestic Water Coil	Net: With Standard Coil	Net: With Special Coil	Square Feet Hot Water Radiation: Gross	Net: Without Domestic Water Coil	Net: With Standard Coil	Net: With Special Coil	Domestic Water Coil Capacities, Gallons of Water in Three Hrs 100°F Temperature Rise — Steam Boilers at 180°F Average Temp: With Standard Water Coil	Steam: With Special Water Coil	Hot Water Boilers at 150°F Average Temp: With Standard Water Coil	Hot Water: With Special Water Coil
B—3	780	500	428	384	1248	800	740	706	100	160	50	80
B—4	1095	700	628	584	1743	1120	1060	1026	100	160	50	80
B—5	1405	900	828	784	2248	1440	1380	1346	100	160	50	80
B—6	1720	1100	1028	984	2752	1760	1700	1666	100	160	50	80
B—7	2030	1300	1228	1184	3246	2080	2020	1986	100	160	50	80

*NOTE —Gross rating is the total output in square feet of radiation at the boiler nozzle Net ratings represent the allowable standing cast iron radiation which can be connected to each boiler when the domestic hot water load is as shown at the top of each column No deduction for the hot water coil, regardless of storage tank size, should be made from the net rating, since the maximum load imposed by each coil has already been deducted Efficiency—Series "A6" Boilers show a test efficiency of 77 per cent, Series "B"

80 per cent, plus over-all at rated capacity for continuous firing

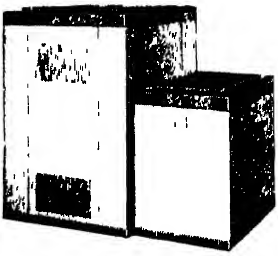
NOTE—When ordering be sure to specify whether the unit is for a steam or hot water system

Number of Boiler	B3	B4	B5	B6	B7
Number of Double Sections	3	4	5	6	7
A	51"	59	67"	75	83
B	0"	8"	16"	24"	32
C	15¾	23¾	31¾"	39¾	47¾



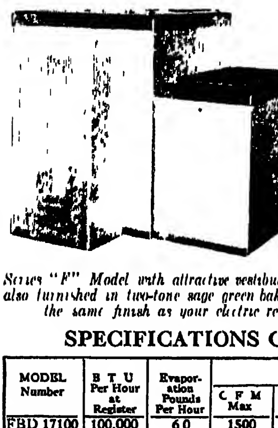
GILBARCO SERIES "F" AIR CONDITIONING UNITS

HEAT—CLEAN - HUMIDIFY—CIRCULATE

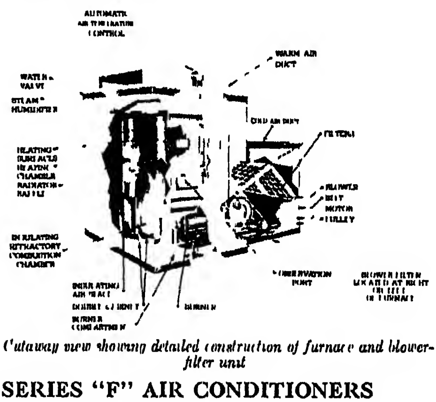


Deluxe Model, compact, self-contained unit Furnished in two-tone sage green baked enamel Burner and controls completely enclosed

Gilbarco Series "F" Air Conditioning Systems are efficient, dependable and unusually economical in operation They are specially designed for oil fuel and develop a high degree of heating efficiency The oversized radiator within the furnace, with its long, retarded gas travel insures a great amount of heat with economical fuel consumption The filtering and humidifying equipment is engineered according to advanced principles of this new science and the fan and motor are of sufficient capacity and durability to insure many years of efficient service



Series "F" Model with attractive vestibule housing also furnished in two-tone sage green baked enamel the same finish as your electric refrigerator



Cutaway view showing detailed construction of furnace and blower-filter unit

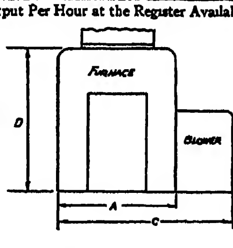
SPECIFICATIONS GILBARCO SERIES "F" AIR CONDITIONERS

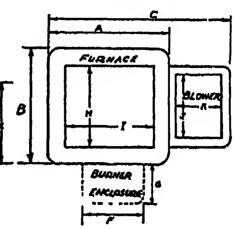
MODEL Number	B T U Per Hour at Register	Evaporation Pounds Per Hour	BLOWER				DIMENSIONS (Inches)										
			C F M Max	Fan Diam	Motor H P	Filters No — Size	A	B	C	D	E	F	G	H	I	J	K
FBD 17100	100,000	60	1500	12	⅙	4 -20 x 20	36	45	64	61	40	—	—	37	28	35	20
FBD 22125	125,000	80	2000	12	¼	4 -20 x 20	43	51	71	61	40	—	—	43	35	35	20
FBD 27150	150,000	120	2200	16	¼	6 -16 x 25	54	63	82	61	43	—	—	55	46	43	20
FBD 27200	200,000	140	2500	16	⅓	6 -16 x 25	54	63	82	61	43	—	—	55	46	43	20
FBS 32300	300,000	160	4000	18	¾	9 -20 x 20	61	62	97	63	56	32	27	54	53	52	28
FBS 35400	400,000	180	5000	21	¾	12 -16 x 20	61	67	97	72	59	32	27	59	53	57	28
FBS 35500	500,000	200	6000	21	1	12 -16 x 20	61	67	97	72	59	32	27	59	53	57	28

Special Models for 750,000 and 1,000,000 B T U Output Per Hour at the Register Available Consult the Factory

F indicates Series
B " Blower
D " Deluxe Cabinet Enclosing Burner
S " Standard Cabinet, Burner Not Enclosed

Burner Enclosure can be Furnished on Standard Models at an extra Cost





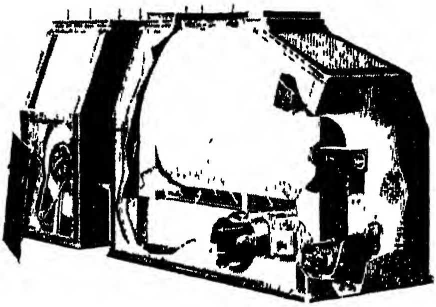
Gar Wood Industries, Inc.

AIR CONDITIONING DIVISION

7924 Riopelle Street *Gar Wood* **Detroit, Mich.**

Licensed Distributors in All Principal Cities

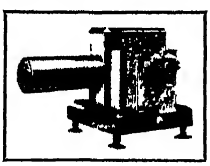
TEMPERED-AIRE UNIT



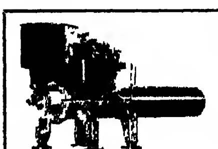
Tempered-Aire heating and air conditioning equipment includes filters, blower, humidifier, furnace (with "Economizer") and integral oil burner of pressure atomizing type Cloth filters can be easily laundered All units can be equipped with a water heating coil for use in winter The installation of an auxiliary duct for intake of outside and basement air in summer is desirable Air can then be drawn from the basement during the day and from the outside at night to provide ventilation and cooling during warm weather Change from basement air to night air may be manually or automatically controlled

Ratings and Dimensions	No 102-A	No 102	No 103	No 104	No 105	No 105 C
Btu 1 Hour at Bonnet	120,000	120,000	165,000	225,000	300 000	400,000
Btu 1 Hour at Grilles	100,000	100,000	135 000	185,000	245,000	
Air Delivery, CFM	1000 to 1575	1000 to 1575	1375 to 2150	1875 to 2950	2500 to 3925	3000 to 4500
Heating Surface, Firebox, Sq Ft	30	30	33	44	55	55
Heating Surface, Economizer, Sq Ft	60	60	99	132	165	165
Total Heating Surface	90	90	132	176	220	220
Filter Area Sq Ft	24	34	43	60	77	77
Motor Horse Power—Burner	1/6	1/6	1/6	1/6	1/4	1/4
Motor Horse Power—Blower	1/4	1/4	1/3	1/2	3/4	1
Current Consumption—Burner	250W	250W	250W	250W	325W	325W
Current Consumption—Blower	250W	250W	330W	500W	750W	1000W
Overall Length, Inches	92¼	96 9/16	117 13/16	140¼	156¼	156¼
Overall Width, Inches	38⅛	40	40	40	40	40

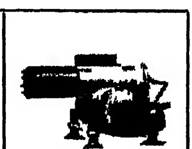
CONVERSION OIL BURNERS—Provide automatic oil burning equipment for coal-fired heating plants Pressure atomizing type, handle heavy, low grade fuel oil. Sturdy, noiseless, easily accessible. Fit any furnace or boiler regardless of shape



Model "B"
412–2500 sq ft net steam radiation



Model "K"
412–2400 sq ft net steam radiation



Model "H"
Up to 625 sq ft net steam radiation

MODEL "Q" AUTOMATIC OIL-FIRED WATER HEATER Provides quick, convenient, inexpensive and unlimited supply of domestic hot water for industrial and commercial buildings Complete, quiet Burns low priced No 3 fuel oil

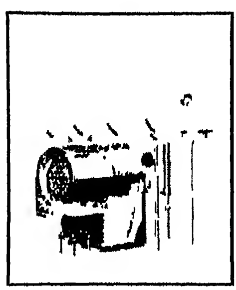
Capacities and Dimensions

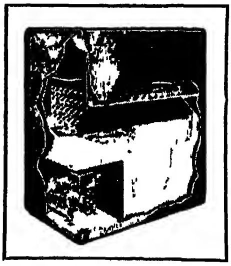
Two sizes, 200 and 300 gal capacity 100° F rise per hour

Overall height 48 in.

Overall width 28¼ in.

Length 49 in and 61 in. respectively





MODEL "R" BOILER-BURNER UNIT

A compact, fire-tubular steam or hot water heating boiler, with an integral oil burner

Boiler built of heavy rust-resisting boiler plate, electrically welded The combustion chamber walls are carried clear to the crown sheet The result is a considerably higher temperature of fire than is found in the conventional boiler, and better combustion of No 3 oil

Since there are no water legs, the bulk of the heating surface is concentrated in the firetubes, or secondary heating surface, resulting in maximum extraction of heat from the hot gas which leaves the firebox and consequent low stack temperature, high efficiency and operating economy

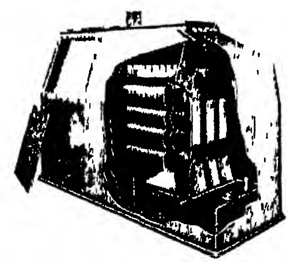
Ratings and Dimensions	R475	R750	R1000	R1400	R1800
Maximum Net Steam Load Sq Ft	475	750	1000	1400	1800
Maximum Net Hot Water Load Sq Ft	760	1200	1600	2240	2880
Maximum Gross Steam Load Sq Ft	712	1125	1500	2100	2700
Maximum Gross Hot Water Load Sq Ft	1140	1800	2400	3360	4320
Heating Surface, Sq Ft	52	68	84	118	154
Overall Width Inches	30 3/16	30 3/16	37 11/16	37 11/16	37 11/16
Overall Length Inches	53	65	59 3/4	66	80

INDIRECT AIR CONDITIONING CABINET

Combined with "R" boilers, it provides heating, humidifying, filtering and circulation of air

The air drawn by blower is first cleaned by dry cloth filters, then forced through the humidifying chamber, where proper amount of healthful moisture is added by passing through a warm vapor mist, regulated by a room humidistat. The air is then warmed to correct temperature by passage through the copper blast heater Standard units, in capacities from—

111,000—497,000 Btu Hour at bonnet
89,000—397,000 Btu Hour at grilles



RATINGS AND DIMENSIONS

	No 90	No 120
AGA Input, Btu per hour	135,000	180,000
AGA Output, Btu per hour	101,250	135,000
Grille Delivery Btu per hour	90,000	120,000
Air Delivery CFM	900–1425	1200–1900
Filter Surface Sq Ft	34	34
Blower Motor Size	1/4 hp	1/4 hp
Blower Motor Current Consump	250 Watts	250 Watts
Overall Length, Inches	96 9/16	96 9/16
Overall Width, Inches	40	40

GAS-FIRED TEMPERED-AIRE

Built in a single compact unit it accomplishes the various functions of air conditioning in an efficient manner Filtering, ventilating, humidifying and cooling in summer functions are similar to the oil-fired Tempered-Aire

Heating is automatic, clean and odorless The gas valve is controlled by a thermostat The furnace is made of heavy gage copper-bearing steel welded into a single unit Corrugations and ribbings give the heating surface more than double the area of a flat surface of equal outside dimensions

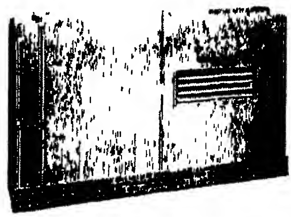
See also Pages 1026-1027

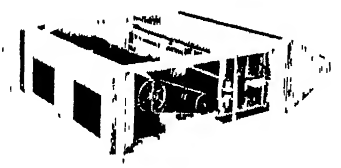
GENERAL ELECTRIC COMPANY

AIR CONDITIONING PRODUCTS

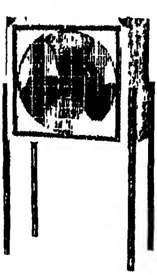
Air Conditioning Department, Bloomfield, N. J.

G-E Unit Air Conditioners and Room Coolers—Available in a wide range of sizes and types for a variety of air conditioning applications Type FR-1 Unit Room Air Conditioner includes condensing unit and complete year 'round air conditioning unit, Type AD Room Air Conditioners include complete year 'round air conditioning unit, Type AG Room Coolers include floor, wall and suspended cooling units



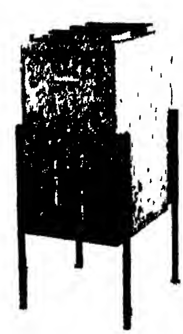


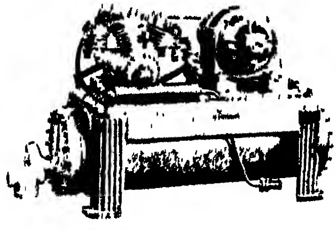
G-E Central Plant Air Conditioners—A complete line of factory designed and assembled air conditioners for summer, winter or year 'round applications Type HD-100, HD-200 and HD-300 of the suspended type include aphonic radial flow fan, filters, humidifiers, cooling coils and heating coils in combinations to meet a wide range of air conditioning functions and required capacities Larger sizes Type HD-400, 500, 600 and 700 include filters, humidifiers, cooling coils and heating coils to meet a variety of functions and capacities



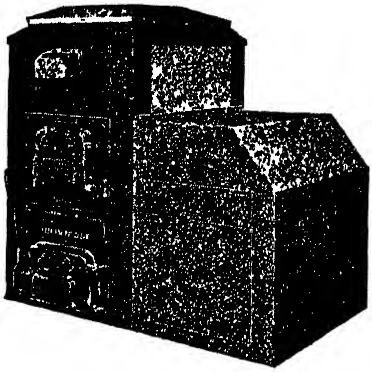
G-E Air Circulator Type HV-1 for attic ventilation, air circulation and exhaust applications

G-E Air Conditioner for Winter—Type HW-1 designed for winter air conditioning of radiator heated homes Includes filters, humidifier, tempering coil and radial flow aphonic fan





G-E Condensing Units—Available in sizes from 1 hp thru 40 hp Several air cooled models in small sizes, water cooled models with shell-and-coil and truly cleanable shell-and-tube condensers Efficient design provides high cooling capacities Designed especially for air-conditioning application as part of complete G-E air conditioning systems



Moncrief "Aristocrat" Air Conditioning System —for coal, hand or stoker fired

The Moncrief Aristocrat Air Conditioning Unit is made in two Series, "C" for cast iron and "S" for steel, riveted and welded. The Series "S" unit up to and including the No 2456 is $\frac{3}{16}$ in boiler plate construction. All larger steel units are made of $\frac{1}{4}$ in boiler plate.

The equipment consists of a quiet operating blower, a patented windbox (exclusively Moncrief) which distributes the air evenly and at a high velocity over the heating surface, Minneapolis-Honeywell Thermostat, damper motor, blower and high limit control, automatic humidifier, and air filters.

The casing is well designed and is furnished in red and black crackled enamel finish.

Dimensions, Capacities and Data of Moncrief Aristocrat Coal Fired Air Conditioning Units

C SERIES-CAST

Furnace Number	Size of Casing Without Blower: Width	Length	Height	Size of Casing With Blower on Rear: Width	Length	Height	Blower on Side Width	Blower Size	Motor HP	Heating Surface Sq Ins	Grate Area Sq Ins	Casing Free Area Sq In	C F M Required	Baffle Marvell Therms	[illegible]	Forced Air Register Btu Coal Hand Fired	Forced Air Register Btu Oil or Stoker Fired	Gravity Register Btu	Gravity Pipe Area Sq Ins	Ratio
4020	40	40	61¾	40	73	61¾	76	110	¼	5021	214	398	967	350	2	73800	85,200	55 [illegible]00	401	2346
4422	42	36¼	61¼	42	72½	61¼	78	112	¼	5609	269	476	1157	350	3	88 300	101,800	64000	478	2085
4824	44	40	61¼	44	76	61¼	80	112	¼	6010	326	547	1330	350	3	101 [illegible]00	117300	75800	5[illegible]	1845
5226	46	44	62¾	46	80	62¾	82	112	¼	6753	397	650	1580	350	4	120 300	1 [illegible]9000	90000	651	1700
5628	48	48	62¾	48	84½	62¾	84	114	⅓	7579	472	757	1840	350	5	140 300	16[illegible]000	104800	762	1606
6033	54	57	73	54	99	73	96	214	[illegible] ¾	10920	705	1115	2710	350	6	206600	2[illegible]8400	1[illegible]4500	1108	1550

S SERIES-STEEL

Furnace Number	Size of Casing Without Blower: Width	Length	Height	Size of Casing With Blower on Rear: Width	Length	Height	Blower on Side Width	Blower Size	Motor HP	Heating Surface Sq Ins	Grate Area Sq Ins	Casing Free Area Sq In	C F M Required	Baffle Marvell Therms	[illegible]	Forced Air Register Btu Coal Hand Fired	Forced Air Register Btu Oil or Stoker Fired	Gravity Register Btu	Gravity Pipe Area Sq Ins	Ratio
2054	40	40	61¾	40	90	61¼	76	110	¼	6226	214	440	1070	350	2	81600	94,100	60900	442	291
2254	40	43	61¾	40	97	61¾	76	112	⅓	6499	269	506	1232	350	3	94000	108400	70,200	508	241
2456	40	46	61¾	40	100	61¾	76	112	⅓	6960	330	585	1424	350	4	108,700	125,300	81,200	589	211
2758	40	51	62¾	40	106	62¾	76	114	⅓	8086	434	747	1815	350	5	138600	160000	103500	717	1865
3060	48	54	68½	48	109	68½	84	212	½	10041	552	923	2240	350	5	171,000	197300	127,700	927	182
3060J	52	54	68½	52	109	68½	88	212	½	11480	552	971	2362	350	5	180 300	208000	134600	980	208
3462	48	60	68½	48	120	68	90	214	¾	11245	730	1152	2800	350	6	213800	246700	159,600	1162	154
3462J	52	60	68½	52	120	68	94	214	¾	12186	730	1190	2895	350	6	2[illegible]1200	255200	165,200	1201	1697

With blower on side use dimensions length and height under size of casing without blower

C F M is based on 70 deg room temperature and 135 deg register temperature

For heating capacity 2 per cent was added for each square foot of heating surface above ratio of 20 to 1 and 2 per cent deducted for each square foot below ratio of 20 to 1

Heating capacities are based on a 7 lb combustion rate using 12,000 Btu coal and efficiencies of 65 per cent at bonnet, 85 per cent register for fan coal hand fired 75 per cent at bonnet, 85 per cent at register for oil or stoker fired (oil—140,000 Btu and 0 55 gal per sq ft grate per hour) 55 per cent bonnet, 75 per cent at register for gravity hand coal fired

Gravity pipe area equals 1 75 × grate area [1 00 + 02 (R − 20)]

If 6 lb combustion rate is desired for prolonged firing period deduct approximately 15 per cent from forced air Btu ratings

Maximum velocity through filters, 200 feet per minute

The Anthracite Winter Air Conditioner consists of a MONCRIEF Unit and a Miles Combustion Modulator

The outside temperature is correlated with the warm air temperature to affect a Graduated Room Temperature, 68 deg in mild weather and 78 deg in zero weather

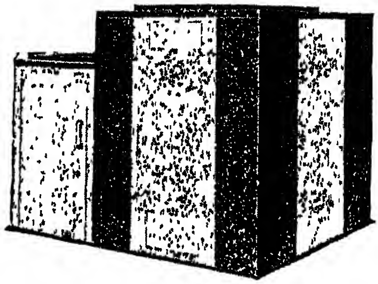
Constant Modulated Heat Supply eliminates COLD 70 and complicated control equipment.

Write for particulars

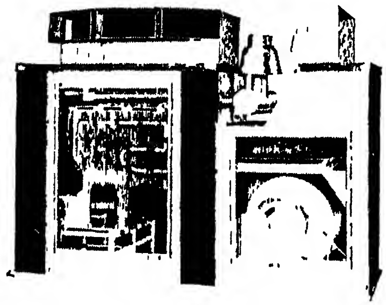
NOTE This unit can be made with either the Moncrief Cast or Steel Aristocrat Air Conditioner.

Dimensions and Capacities of Moncrief "Special" Oil Fired Air Conditioners

Unit No	75	100
Input Gallons per hour	¾	1
Size of casing		
Width	28″	28″
Depth	70¾″	70¾″
Height	60″	60″
Blower size	110	112
Motor H P	1 6	¼
Heating surface—sq in	6120	6120
C F M	800	1100
No filters	2	2
Btu Del at Register	73 000	100 000



Moncrief "Aristocrat" Oil-Fired Air Conditioning System



Moncrief "CBF" Gas Air Conditioning System

The Moncrief Gas Air Conditioning Unit is a cast iron heater, designed to burn efficiently, mixed, artificial or natural gas It is equipped with a quiet operating blower, air filters, thermostat, blower and high limit control, automatic humidifier, motor valve, pressure regulator, shut-off cock and automatic pilots It is furnished in a well designed casing with manifolds enclosed, and finished in green crackled enamel

Controls are Minneapolis-Honeywell

The Moncrief Special Oil Fired Air Conditioner, designed for small homes—¾ and 1 Gal Per Hour Input and Moncrief Aristocrat Oil Fired Air Conditioning units designed for larger homes with correspondingly larger oil input Both welded steel, 3/16 in boiler plate

These units are compact, finished in Red and Black Crackled Enamel, with enclosure for burner, supplied with automatic humidifier, air filters, quiet operating blower and adjustable blower control They are made for use with standard types of oil burners and are supplied less burners and burner controls

The Aristocrat model is equipped with special patented windbox supplying a directional flow of air over all heating surfaces

Moncreif Air Conditioning Unit for 1937 is being improved in style and these new styles will be shown in literature which may be had on request

Dimensions, Capacities and Data of Moncrief Gas Furnaces and Gas Air Conditioning Units

Furnace Number	Size of Casings in Inches: Type B and Type C Width	Type B Less Blower Length	Type C Less Blower Length	Type C With Blower Length	Type B Height	Type C Height	Number of Burners	Dia Flue Pipe	Size Gas Line	Heating Surface	Casing Free Area	Velocity Thru Furnace	BTU Input	Forced Air BTU at Register	CFM	Blower Size	Motor HP	Number of Filters	Velocity Thru Filters	Gravity BTU at Register	Gravity Pipe Area
50	12	44			48		1	3	⅜	2160	233		30000							19125	170
100	19	44	51½	72	51¼	54	2	4	1	4320	376	215	60000	42750	560	110	⅓	2	105	38250	281
150	26	44	51½	72	51¼	54	3	5	1	6480	438	277	90000	64125	840	110	¼	2	158	57375	422
200	34	44	51½	72	51¼	54	4	6	1	8640	584	277	120000	85500	1120	112	⅓	3	140	76500	562
250	41	44	51½	72	51¼	54	5	7	1¼	10800	730	277	150000	106875	1400	112	¼	4	131	95625	703
300	49	44	51½	74	51¼	54	6	8	1¼	12960	976	248	180000	128250	1680	114	⅓	4	158	114750	844
350	56	44	51½	74	51¼	54	7	8	1¼	15120	1146	246	210000	149625	1960	114	½	5	147	133875	984
400	64	44	51½	72	51¼	54	8	9	1¼	17280	1168	277	240000	171000	2240	212	½	5	168	153000	1124
450	71	44	51½	72	51¼	54	9	10	1¼	19440	1314	276	270000	192250	2520	212	½	5	183	172000	1265
500	78¾	44	51½	74	51¼	54	10	10	1½	21600	1376	293	300000	213750	2800	214	¾	6	175	191250	1406
600	94½	44	51½	74	51¼	54	12	10	1½	25920	1678	288	360000	256500	3360	214	¾	6	210	229500	1688

Type B furnace casing has manifold exposed and is designed for gravity use Blower and filters can be added as separate unit

Type C furnace casing has manifold enclosed and is designed for blower-filter use Can be used for gravity circulation if desired

Forced air Btu ratings are based on efficiencies of 75 per cent at bonnet and 95 per cent at register

Gravity Btu ratings are based on efficiencies of 75 per cent at bonnet and 85 per cent at register

C F M capacities are based on 70 deg room temperature and 135 deg register temperature

Add 11 in to overall height for 14 in pitched top used with Type B casing

Lennox Furnace Co., Inc.

Aire-Flo Air-Conditioning Units—Riveted Steel Furnaces

Syracuse, New York **Marshalltown, Iowa**

Dealers in Principal Cities

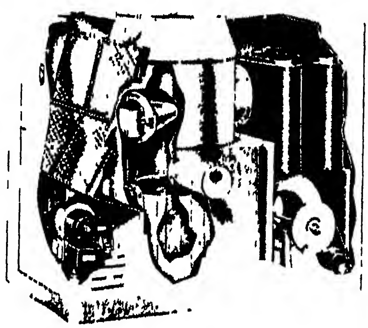
Lennox offers a complete line of direct-fired winter air conditioners, backed by a long experience in the field of domestic air conditioning

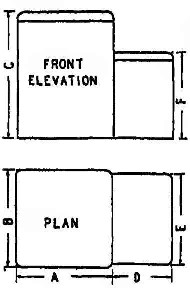
Lennox is the world's largest manufacturer of steel heating furnaces All heaters for gravity and forced-air systems are of the famous Lennox riveted-steel construction—permanently gas- and dust-tight In the line are units designed especially for every kind of fuel and every size home Lennox Aire-Flo air-conditioning units are unusually quiet in operation and are highly efficient Every unit delivers warm air at uniform temperature and velocity to every room

Basic specifications of the more popular air-conditioning models are given in the accompanying tables Larger blowers are used in well-insulated buildings where ventilation requirements are high in proportion to heat losses Complete specifications on gravity furnaces, air conditioners, blowers and oil burners on request

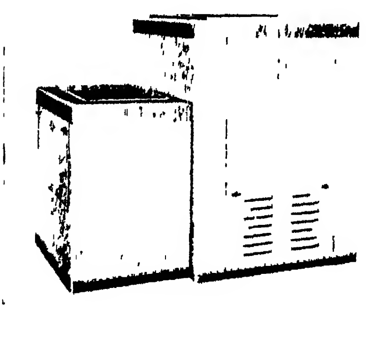
LENNOX OIL AIRE-FLO

Highly efficient unit designed especially for oil Cold air enters at the front of the cabinet, goes through the filters, circles the inner casing on both sides, is baffled up over the large double doughnut radiators, down through the blower, into the heating chamber and up into the plenum chamber Heat loss through cabinet is less than ½ of one per cent Low stack losses Inserted steam type humidifier Firepot lined with insulating type brick which heats fast, results in clean, more efficient fire Blower and oil burner unusually quiet Heater, filters, blower, humidifier and oil burner in one attractive cabinet





Unit Number	Btu at Registers	Normal Cfm	No and Size of Filters	Dimensions A	Dimensions B	Dimensions C
F2-125-1	80,000	1,050	4-16 x 20	40	73 [illegible]	59
F2-125-2	100,000	1,310	4-16 x 20	40	73 [illegible]	59
F2-125-3	125,000	1,640	4-16 x 20	40	73 [illegible]	59
F2-200-1	150,000	1,950	6-16 x 20	48	83	59
F2-200-2	175,000	2,300	6-16 x 20	48	83	59
F2-200-3	200,000	2,620	6-16 x 20	48	83	59
F2-300-1	250,000	3,300	9-16 x 20	60	104	63
F2-300-2	300,000	3,920	9-16 x 20	60	104	63



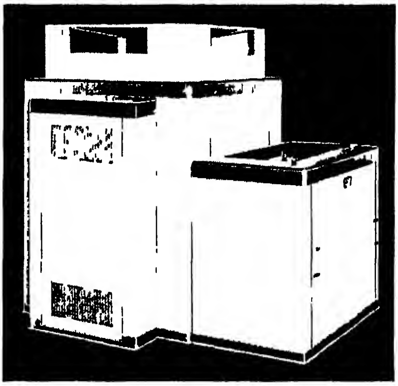
LENNOX GAS AIRE-FLO

For Natural, Manufactured or Mixed Gas

Designed exclusively for gas, with the large amount of heating surface required for this fuel Highly efficient A.G.A approved The K1-7 series has a large circular radiator completely encircling the combustion chamber The larger units have two circular radiators All control valves except main shut-off are enclosed Each unit equipped with the new Lennox gas burner, external pilot lighter and automatic safety controls Blower cabinet can be placed on either side

Unit Number	Btu at Register	Heating Surface Sq In	Normal Cfm Range	No and Size of Filters	Dimensions					
					A	B	C	D	E	F
K1-701	65,000	7,760	500-1000	2-16 x 20	40	46	58	26	33	30
K1-721	90,000	7,760	800-1400	4-20 x 20	40	46	58	30	40	36
K1-1522	125,000	12,814	1300-2000	4-20 x 20	45	48½	58	30	40	36
K1-2062	160,000	18,623	2200-2600	6-20 x 20	53½	57	63	34	48	37
K1-2063	200,000	18,623	2500-3000	6-20 x 20	53½	57	63	34	48	37

LENNOX C8 AIRE-FLO FOR ALL FUELS

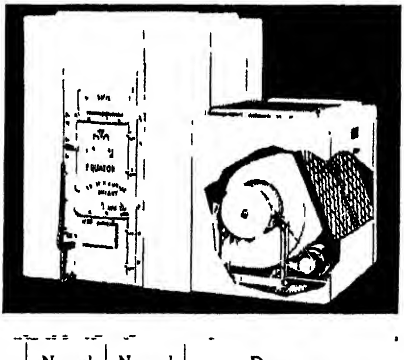


These units are designed especially for coal, but are also efficient with other fuels Heaters are of riveted-steel, permanently tight against dust and fuel gas Inserted steam-type humidifier, with adjustable float valve to control rate of evaporation Large heating surfaces Cabinets are air-cooled Front vestibule illustrated not standard equipment New front (not shown) is a complete departure from old-fashioned design, combining well-proportioned flat surfaces and round corners and edges Entire unit efficient, easy to operate, quiet and moderately priced

Unit Number	Btu Soft Coal	At Hard Coal	Registers Oil-Gas-Stoker	Grate Area Sq In	Heating Surface Sq In	Normal Cfm Range	No and Size of Filters	Dimensions					
								A	B	C	D	E	F
C8-2422	110,000	100,000	120,000	318	7,605	1300-2000	4-20 x 20	40	47½	59	30	40	36
C8-2722	120,000	110,000	135,000	415	8,774	1300-2000	4-20 x 20	44	53	61	30	40	36
C8-2761	125,000	115,000	145,000	415	8,774	1800-2300	6-20 x 20	44	53	61	34	40	37
C8-3262	150,000	140,000	170,000	605	10,866	2200-2500	6-20 x 20	51	63¼	63	34	40	37
C8-3263	175,000	160,000	200,000	605	10,866	2500-3000	6-20 x 20	51	63¼	63	34	40	37
C8-35182	200,000	180,000	300,000	719	15,333	3300-4000	9-20 x 20	61	67	71	36	60	56
C8-35211	275,000	245,000	350,000	719	15,333	4000-5000	12-16 x 20	61	67	71	36	65	59
SS-800	800,000	720,000	1,000,000	1037	28,723	Special		62*	96*	74½			

*Casing Obround

LENNOX EQUATOR AIRE-FLO



These three units are designed for the smaller homes and are extremely low in price All fuels The heaters are built of leak-proof riveted steel Open-pan type of evaporator Fuel efficiency only slightly lower than that of the regular C8 Aire-Flo units Blower is efficient and unusually quiet This unit is ideal for the architect who is designing a small, moderately priced home

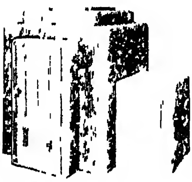
Unit Number	Btu at Registers			Grate Area Sq In	Heating Surface Sq In	Normal Cfm Range	No and Size of Filters	Dimensions					
	Soft Coal	Hard Coal	Oil-Gas Stoker					A	B	C	D	E	F
QBI-220	70,000	60,000	75,000	262	5,363	500-1000	2-16 x 25	37	38⅜	56	26	33	30
QBI-242	95,000	85,000	105,000	318	6,448	800-1400	4-20 x 20	41	42¾	56	30	40	36
QBI-272	115,000	105,000	130,000	415	7,170	1300-2000	4-20 x 20	44	46¼	58	30	40	36

THE MEYER FURNACE COMPANY

PEORIA, ILLINOIS

Manufacturers of Domestic Heating and Air Conditioning Units for Coal, Gas and Oil Burning

Branches and Distributors
KANSAS CITY, MO
OMAHA, NEB
GREEN BAY, WIS
PITTSBURGH, PA
NEW ORLEANS, LA
DETROIT, MICH
ST LOUIS, MO
SAN FRANCISCO, CALIF
DES MOINES, IOWA
MINNEAPOLIS, MINN

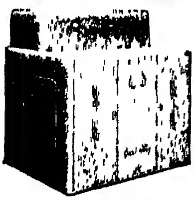


WEIR Conditioned Air Unit

The WEIR Conditioned-Air Unit for coal burning, built around the famous WEIR Steel Furnace, is a complete unit for conditioning the air in the home during the heating season Equipment includes automatic humidifier, renewable filter, centrifugal blower and automatic damper and blower controls

WEIR Gravity Heater

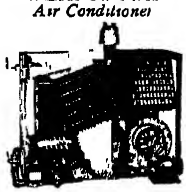
No	Grate Surface (Sq Ft)	Ratio Htg to Grate Surface	Smoke Outlet Diam (In)	Gravity Circulation				Fan Circulation		
				Casing Dimen		Rated Output				
				Round (In)	Rect'lar (In)	At Reg (Btu Hour)	Pipe Area (Sq In)	Casing Dimen (In)	Air Delivery (CFM)	Rated Output at register (Btu Hour)
621	1 26	41 2	9	48		54,400	400			
624	1 78	33 9	10	52	47x50	73,600	541	47x90	1200	92,000
628	2 32	29 2	10	54	50x52	94,100	692	50x99	1600	118,000
630	3 08	26 4	10	58	54x56	119,000	875	54x103	2000	148,000
633	3 82	22 7	10	65	56x64	138 000	1015	56x110	2300	172,000
636	4 74	19 4	10	67	56x66	160,000	1180	56x118	2700	200,000
540	6 25	19 3	12					60x106	4000	264,000
544	7 60	18 5	12					64x114	5000	316,000



WEIR Oil Fired Air Conditioner

The WEIR Oil-Fired Air Conditioner does a complete job of winter air conditioning Designed for oil fuel and forced circulation Features include quiet operation, dependability, safety, long life, modern appearance and high efficiency

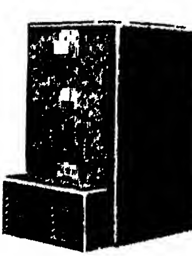
The MEYER Gas-Fired Air Conditioner provides complete winter air conditioning Modern in appearance, compact and efficient with heavy gauge, welded steel, gas tight heating section Equipment includes automatic humidifier, renewable filters, centrifugal blower and fully automatic controls.



MEYER Gas Fired Air Conditioner

No	Input at Burner (Btu/Hour)	Output at Bonnet (Btu/Hour)	Vent Diam (In)	Dimensions			Air Delivery ¼ In S P (CFM)	Motor Size (HP)
				W (In)	L (In)	H (In)		
WEIR Oil-Fired Air Conditioner								
125-A	200,000	150,000	7	48	65	48	1700	1/4
175-A	275,000	205,000	7	56	65	48	2300	1/3
225-A	350,000	265,000	7	63	65	48	3000	1/2
MEYER Gas-Fired Air Conditioner								
B-1	90,000	67,500	4	20	53	42	1000	1/6
B-1½	135,000	101,250	5	27	60	44	1500	1/4
B-2	180,000	135,000	7	40	53	42	2000	1/3
B-3	270,000	202,500	8	60	53	42	3000	1/2
B-4	360,000	270,000	9	80	53	42	4000	3/4
B-5	450,000	337,500	10	100	53	42	5000	3/4
MEYER Gravity Gas Furnace								
C-100	100,000	75,000	5	38	38	60	Complete descriptive literature, including data on Summer cooling, upon request	
C-120	120,000	90,000	6	42	42	67		
C-150	150,000	112,500	6	42	42	67		

The MEYER Gravity Gas Furnace —Efficient Economical —All steel, welded heating section—large heating surface A G A approval



MEYER Gravity Gas Furnace

Kelvinator

Division of Nash-Kelvinator Corporation

SUMMER AND WINTER AIR CONDITIONING

Factories in Detroit, Michigan, and London, Ontario

Distributors in all Principal Cities

Air Conditioning
Household Refrigeration
Automatic Heating
Water Cooling
Beverage Cooling
Milk Cooling

Ice Cream Cabinets
Truck Refrigeration
Commercial Refrigeration For Every Need of Merchants, Manufacturers and Institutions

Selecting the Most Suitable Type of Air Conditioning Installation

Where a year-'round central system installation is out of the question due to space limitations, remodeling expense, time allotment or other factors, the following general suggestions will be helpful Whenever possible, we urge consultation with your local Kelvinator Air Conditioning distributor He offers you every advantage of Kelvinator's broad experience in air-conditioning all types of buildings—his services are freely available without charge or obligation

Single rooms in residential or commercial buildings: Water-cooled or air cooled room coolers, or type C suspended units

Large rooms, such as sales rooms, general offices, or manufacturing quarters: One or more CA Suspended Units, as required for capacity and air distribution, all served by a single condensing unit if practical Or a type CS Unit with duct distribution throughout the quarters to be air conditioned Condensing unit may be remotely located most convenient to plumbing and electrical connections

Suites, or a separate floor of a building: CA units exposed, or concealed behind closet or corridor walls, served in multiple by a single condensing unit installation in an out-of-the-way location where plumbing and electrical connections can be arranged economically Or a CS unit with duct distribution to each office

New construction of all types: Complete year-'round air conditioning employing CS unit and duct distribution, or individually-designed system Consult your Kelvinator distributor

Capacity Requirements: Summer-load capacity requirements depend on the type of construction of the building, exposure, amount of occupancy and traffic, activity of occupants, heat generating equipment within the enclosure, local weather averages and other factors Consult your local Kelvinator distributor for latest experience data on capacity requirements in your locality

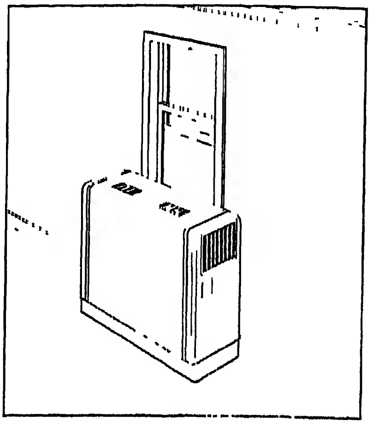
Costs: Your Kelvinator distributor is qualified to aid you in determining economical methods of air conditioning new or existing construction, from the standpoints of operating expense as well as initial cost

All Kelvinator Air Conditioning Units Are Provided with Contact Type Spun Glass Filters

For Complete Specifications, Features and Functions of the Latest Types of Equipment Consult Your Local Kelvinator Air Conditioning Distributor

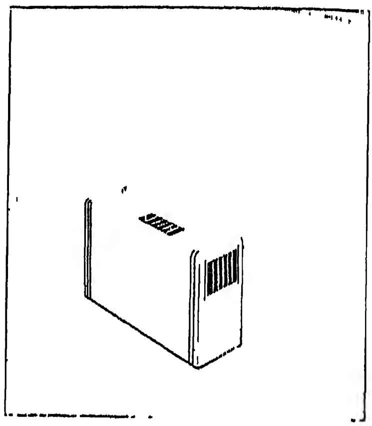
See also Page 930

Room Coolers Air Cooled: For summer air conditioning of a single room or private office. Especially desirable for installations where water connections are not economically available, and for tenants holding short-term leases. Installation involves only an electrical connection and vent to outside air as in window placement shown at right. Can be provided with steam coils for winter heating where removal of radiator is desirable. ½ hp and 1 hp sizes.



Air Cooled Room Cooler

Room Coolers Water Cooled: A compact and economical self-contained floor unit for cooling individual rooms or offices. Can be installed at any wall where water and drain connections can be made, as in inside wall installation shown at right. When disconnected from plumbing, can be moved like other household or office equipment. Attractive all-metal cabinets in walnut grain finish. Sizes ½ to 1½ hp.

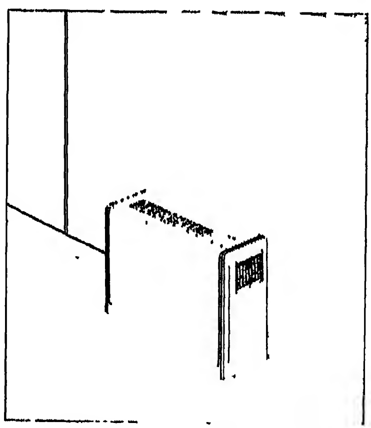


Water Cooled Room Cooler

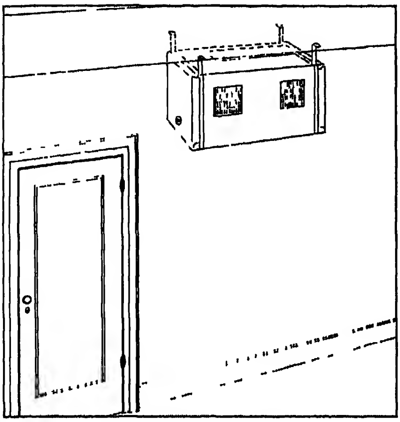
Floor Type Air Conditioning Unit: For summer or year-'round application where floor space is not at a premium. Attractive walnut grained, all metal cabinet, containing cooling and dehumidifying evaporator, filter, blower and grills. Heating coil and humidifier can be included, if desirable. Wide range of sizes for individual or multiple use with remote condensing unit.

All Kelvinator air conditioning units are provided with contact type spun glass filters.

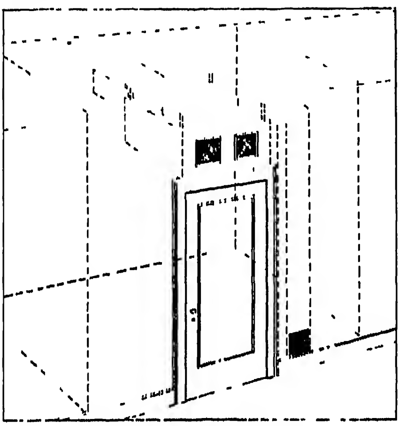
For complete specifications, features and functions of the latest types of equipment consult your local Kelvinator Air Conditioning Distributor.



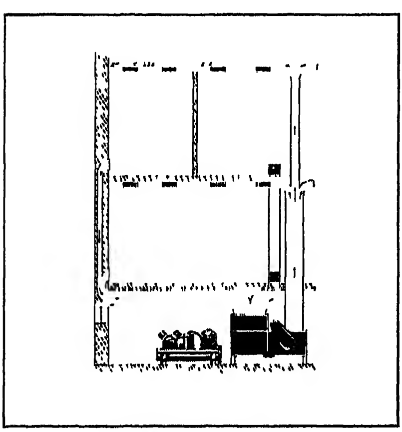
Floor Type Air Conditioning Unit



Type C Suspended Unit



Type CA Suspended Unit



Type CS Central System

Suspended Air Conditioning Unit, type C: For location just below the ceiling in existing individual rooms that are to be air-conditioned. Requires only flexible copper refrigerant lines drain, and electrical connections. Four sizes, for flexibility —0.7 tons to 5.0 tons I.M.E. per day.

Suspended Year-'round Air Conditioning Unit type CA: For individual or multiple connection to remote condensing unit.

Cabinet contains cooling and dehumidifying evaporator, filter, blower and directional air grilles. Heating coil and humidifier can be included, for winter air conditioning. Requires only light-load electrical connections for blower, copper refrigerant lines for connection to remote condenser and drain. Can be installed in the open, in unproductive overhead space as shown, or behind an inside wall as in a closet or storeroom. Capacities 5.0 tons to 10 tons I.M.E. per day.

Central System CS Units: For new and existing construction where an entire establishment or suite of rooms is to be air conditioned. CS units include cooling and dehumidifying evaporators, filters and blower, and heating coils and humidifiers if required. Extremely flexible in arrangement to meet space limitations of installation. Capacities 10 to 40 tons, I.M.E. per day.

Kelvinator DS Evaporators: For locally-designed central system summer or year-'round air-conditioning installation. Exact Selection assures exactly the correct type and size for each need, for economy in first cost and assurance of adequate performance.

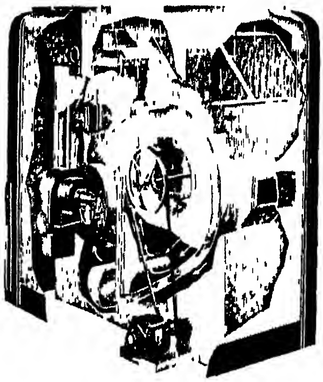
Kelvinator Heating Coils: Provided in complete ranges of sizes, and capacities for all individual needs.

Kelvinator Condensing Units: Air Cooled models 1/6 to 5 hp. Water-cooled models 1/3 to 20 hp. Larger requirements handled by multiple installations providing important advantages in first cost, operating economy, stand-by performance, and savings in upkeep.

Branches		**Branches**
St Louis Memphis Omaha Minneapolis Salt Lake City Dallas	**L. J. Mueller Furnace Co.** Established 1857 2009 W. Oklahoma Ave , **Milwaukee, Wis.**	Los Angeles Kansas City Baltimore Philadelphia Pittsburgh Chicago

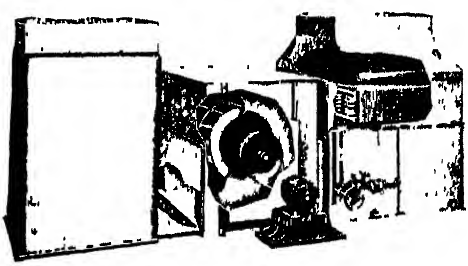
SERIES "O" OIL-FIRED AIR CONDITIONING UNIT

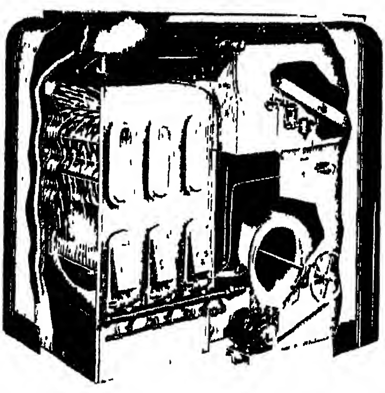
Heats, filters, humidifies and circulates the air, completely controlling indoor "climatic" conditions during the winter, and if desired, cooling and dehumidifying Operation of the unit is automatic Oil burner is pressure atomizing type, available in capacities from 0 75 gal per hour and up The compact simplicity of design, reducing thermal capacity, secures instant heat and eliminates parasitic losses, slow response and temperature override The inner construction and method of operation are shown in the cut-away view Available in three sizes with capacities to handle practically any residential requirement



MUELLER CLIMATOR AIR CONDITIONING SYSTEMS

The assembly shown is designed to handle all functions of both winter and summer air conditioning. Any type Mueller furnace, coal, oil or gas-fired may be used Where winter conditioning only is desired, dehumidification and refrigeration equipment is omitted Climator units are available in a range of sizes and capacities to provide complete or partial conditioning for any size residence





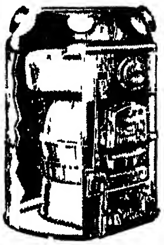
GAS ERA AIR CONDITIONING FURNACE

Enclosed in the compact, ultra-modern cabinet are the furnace, blower, humidifier and filters, together with controls for automatic operation. Heat is supplied by Gas Era steel furnace sections The powerful Mueller fan provides positive circulation of conditioned air to all rooms In the cut away view is shown the fire travel through the end heating section, also the complete assembly of fan, motor, humidifier, filters and controls Unit is available in five sizes, providing a range of capacities capable of handling the requirements of practically any residence

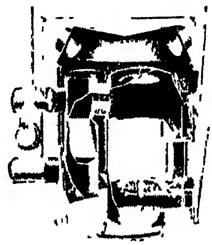
Complete literature on above units upon request

Mueller Heaters For All Fuels

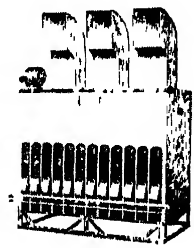
A Complete Line for All Purposes



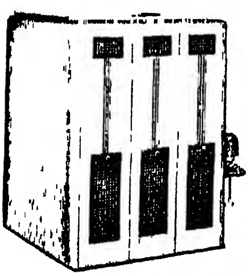
Return Flue all-cast Furnace 18 in to 30 in fire-pots, single and double fire-door styles Available in round, galvanized or square, lacquered casings



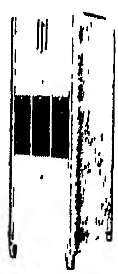
Mueller Steel Furnace Riveted and welded Extra heavy construction Burns any fuel Seven sizes, 20 in to 31 in drums Square or round casing



Gas Era Unit Heater Floor-mounted or suspended type Available in sizes from six to twenty sections, with A G A output ratings from 216,000 to 720,000 Btu per hour



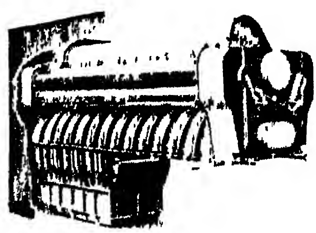
Gas Era cast iron Furnace Sectional construction *A G A* input rating, 65,000 Btu per hour per section. Insulated, lacquered casing for fan or gravity



Muelleraire unit Gas-fired furnace, with blower, filters, and humidifier Available in four sizes with A G A output ratings from 72,000 to 180,000 Btu per hour

Series "A" Gas Era Boiler, for steam, hot water or vapor heating, or hot water storage Nine sizes, A G A ratings of 180 to 1,260 sq ft steam, 290 to 2,015 sq ft hot water

Series 20 Oil-fired Air Conditioning furnace Furnished with or without burner Practically any burner may be used Three sizes, from 125,000 to 200,000 Btu per hour at registers



Nos 93, 94 and 95 cast iron Horizontal Tubular Heaters are especially designed for schools, churches and similar large buildings Capacities from 1,188,000 to 1,390,000 Btu per hour

Catalog on Mueller gravity and air conditioning registers and grilles available upon request

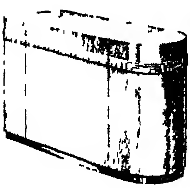
National Air Conditioning Incorporated

***Division of* NATIONAL RADIATOR CORPORATION, Johnstown, Pa.**

General Sales Office, 101 Park Avenue

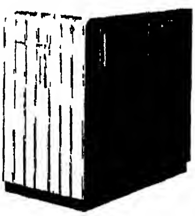
New York, N. Y.

Packaged Heat



National Steel Oil Heating Unit

National Steel Oil Heating Unit—A complete unit incorporating a specially designed National Radiator Corporation steel boiler and a Williams Oil-O-Matic burner An outstanding red jacket design by Lurelle Guild Shipped complete, ready for connection



National Cast Iron Oil Heating Unit (018)

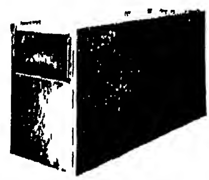
National Cast Iron Oil Heating Unit—The new NATIONAL-WILLIAMS, NATIONAL-NU-WAY and NATIONAL-TORIDHEET Cast Iron Units are especially engineered for these burners respectively Highly efficient, four-pass National cast iron oil heating boilers can be used by burner manufacturers to form a unit with any burner, rotary or gun-type

Model 250 Portable Air Conditioner

National Portable Room Coolers—Model 250 Provide dehumidification, filtration, cooling, circulation and ventilation for offices, stores and rooms in summer, circulation, filtration and ventilation all year 'round Two types—A, air cooled, W, water cooled They are self contained, quiet, and plug in Furnished in richly grained walnut or with a prime coat Write for Form 240 for full information

National Air Conditioning Units

Model 450 Air Conditioning Unit



Model 850 Air Conditioning Unit

Model 1600 Air Conditioning Unit

National Air Conditioning Units are made in three sizes, Models 450, 850 and 1600, for a variety of heating capacities, suitable for small to moderately large homes Additional capacity can be provided by adding more units They furnish all the functions for true summer or winter air conditioning and may be used in humidifying, split or all-air-conditioned systems. All three sizes are available in standard packages with heating or tempering coils Send for Form 247 for complete details on National units, and their equipment, operation, specifications and instructions for installation

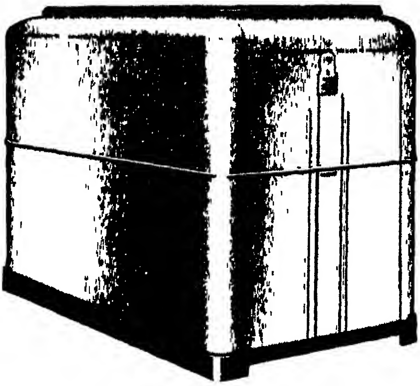
See also Pages 906-908

Norge Division

Borg-Warner Corporation

606-670 E. Woodbridge Street, **Detroit, Michigan**

Norge Fine-Air Conditioning Furnace Unit



One complete, practical, scientific unit that warms, humidifies, filters, circulates, and purifies the air throughout the house, provides plenty of hot water during winter, and has a simple adjustment to prevent steam In summer, it may easily be converted to cool and dehumidify the air Being a scientifically balanced, matched unit, with exceptionally high fuel efficiency, the Norge Fine-Air Conditioning Unit operates at low cost and the low first cost makes it an ideal installation for a new home, or to replace present equipment

Fully automatic with oil or gas as fuel

1500-2200 cfm adjustable air delivery

Effective radiation surface—101 sq ft

Hot gas travel—36 lineal feet

Bonnet output up to 200,000 Btu per hour

Fuel efficiencies over 80 per cent

Air filter automatically controlled, removes 95 per cent of floating impurities in the air

Correct humidification—vaporizes 2 to 16 gal a day

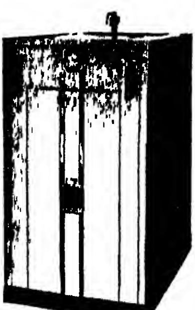
Adjustable, high-output, automatic domestic water heater

Oversize deep-blade fan with reserve power simplifies air duct arrangement

Height	62 in
Width	40 in
Length	68 in

The Norge Whirlator Oil Burner is standard equipment with the Fine-Air Furnace manufactured by Norge Heating and Conditioning Division of Borg-Warner Corporation, Detroit, Michigan

Norge Ideal Matched Boiler Burner Units



Made in two types and six sizes covering heating requirements from 600 to 1350 sq ft of steam radiation. Also supplied with controls for hot water or vapor heat

NC-12 series Boiler is made in four sizes

NC-121, 705 sq ft of steam radiation, NC-125, 912 sq ft, NC-126, 1125 sq ft, NC-127, 1350 sq ft

Norge Fin-Type Boilers, NC-8F5 for 500 sq ft steam radiation, NC-8F7 for 700 sq ft

Model N-8 Norge Whirlator Oil Burner included with all boilers, except NC-127, which requires model N-18 burner

Boilers completely jacketed, beautifully finished Complete controls and complete fire brick included Taco Hot Water Supply Heater optional

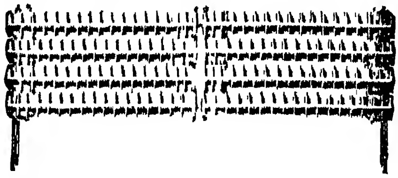
See also Page 932

UTICA RADIATOR CORPORATION

UTICA, N.Y.

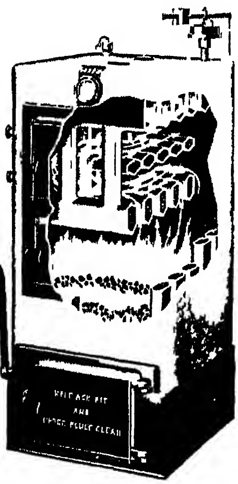
THE UTICA CAST-IRON CONVECTOR

The Utica Convector is designed for installation within enclosures and to heat effectively by convection The cored section is ample to assure good circulation and rapid air elimination The widely spaced fins, integrally cast, will not clog and offer minimum resistance to air flow Utica Convectors will perform equally well on steam, vapor, vacuum and hot water systems They are simple in construction, durable and efficient



The Utica Convector

Made in 3⅝ in, 5⅝ in, 7⅜ in, and 9⅜ in widths and in any length above 18 in in multiples of 5 in



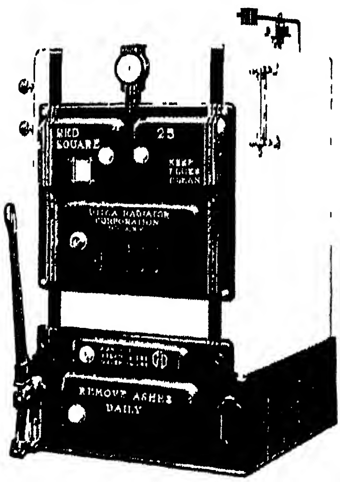
17 Series Redsquare Boiler

UTICA RED-SQUARE BOILERS are stream line in appearance and scientifically designed for efficiency and fuel economy

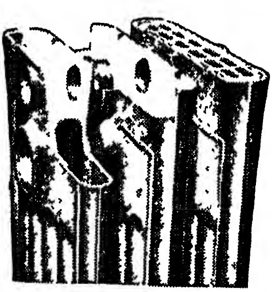
The 17 Series will carry from 200 to 740 sq ft steam radiation and 330 to 1230 sq ft of water radiation

The 25 Series will carry from 620 to 1240 sq ft of steam radiation and 990 to 1990 sq ft of water radiation

Also available in oil burning types with extended jacket



25 Series Redsquare Boiler



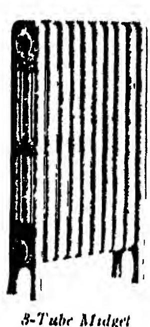
Cut-away view of Humid-Heet Radiator showing porcelain enameled water wells

Humid-Heet Radiator

The HUMID-HEET Radiator furnishes the required amount of radiator heat—and replenishes the air with moisture Made in a wide range of sizes

Midget Radiators

Midget Radiators Due to their narrow width (2-tube, 3¼ in, 3-tube, 4⅝ in.) and sections 1½ in center to center, occupy one-third less space than standard tube radiators

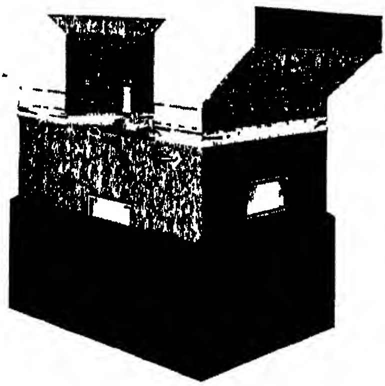


3-Tube Midget 4⅝ in wide

Also Manufacturers of Standard Tube and Wall Radiators

UTICA RADIATOR CORPORATION
UTICA, N.Y.

THE UTICA AIR CONDITIONER
For Homes, Small Buildings and Stores



Utica Air Conditioner Hydro-Air System

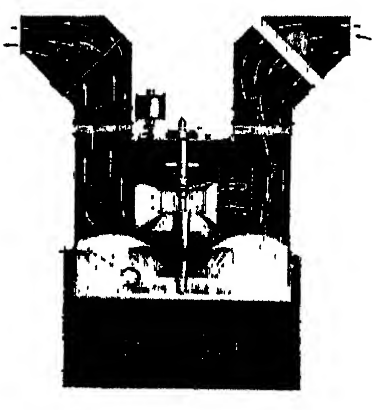
The Utica Air Conditioner was designed by engineers prominently identified with air conditioning since its inception. It is constructed on the Hydro-Air System, which provides operating characteristics and performance similar to the commercial installations of fine theatres, auditoriums and restaurants.

The Hydro-Air System provides accurate control of the conditioned air as it leaves the unit. This permits automatic control of the temperature and humidity in the structure.

A line of standard package units for various sized jobs giving true air conditioning in its fullest sense for any type of residence, small buildings or shops, stores, laboratories and offices. A single moving part simplifies the construction and practically eliminates service.

The Utica Air Conditioner is extremely flexible in its adaptation to operate independent of or in conjunction with any kind of heating system in old or new buildings. Any Utica unit may be installed first as a winter air conditioner and then easily converted into a summer cooling unit, or installed as a year-around conditioner providing heat through ducts or a split system.

A true air conditioner at a reasonable price, in package form, requiring only simple, inexpensive installation. The Utica Air Conditioner is proven in performance by dozens of installations all over the country in all types of jobs over a period of three years.



The Hydro-Air System under which the Utica Air Conditioner is built, provides thorough washing of the air at all times, thus removing not only dust and soot but pollen, odors and smoke

UTICA'S HYDRO-AIR SYSTEM WASHES THE AIR.

Williams Oil-O-Matic Heating Corporation

Manufacturers of Air Conditioning Equipment

Bloomington, Illinois

Service to Architects and Builders

CHICAGO, ILL, 641 N Michigan Avenue — NEW YORK, N Y, 1231 Graybar Building

For Williams Oil-O-Matic Oil Burner Equipment and Ice-O-Matic Refrigeration Equipment, see File Index

Air-O-Matic Combines Heating and Cooling in One Air-Conditioning System

WILLIAMS AIR-O-MATIC CONDITIONING

The Williams Oil-O-Matic Heating Corporation has developed an outstanding year 'round air conditioning system known as "Air-O-Matic" Low pressure steam, which is usually provided by an Oil-O-Matic operated boiler unit, is supplied directly to a copper finned heating coil within the central air distributing unit for heating service, which can be supplemented by direct radiation if desired. Proper provision for the addition of moisture is provided for winter heating service

This same low pressure steam, through an especially developed absorption refrigeration unit, provides the proper degree of temperature and humidity reduction for summer comfort A change from winter to summer operation can be effected almost instantaneously by means of a master control located in a suitable, convenient place

The Williams Low Pressure Steam Absorption Type Unit, the outstanding feature of Air-O-Matic, has been especially developed to meet the particular requirements of air conditioning through years of research in the Williams' laboratories It affords adequate comfort cooling facilities with unusual advantages of economical operation, mechanical simplicity, compactness and freedom from fire and toxicity hazards Both the solvent and refrigerant are newly developed chemicals and are essentially non-toxic, non-inflammable, non-corrosive to the common metals and chemically stable under all operating conditions

The steam requirements for one ton of refrigeration are 21 lb (20,000 Btu) per hour at a pressure of 10 to 12 lb guage The electrical power requirements for one ton of refrigeration are 75 to 100 watt hours per hour per ton

Low Steam and Power Requirements

The low power requirements make possible the use of single phase current in the smaller size installation, thereby saving the expense of providing 3-phase current

For all sizes of machines any low pressure steam boiler of the proper capacity may serve to generate the required steam

> NOTE - Any low pressure steam boiler of proper capacity may be used to generate the required steam

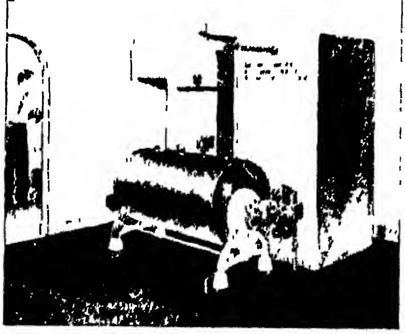
The only metal to metal contact of moving parts within the refrigeration unit is between the seal face and seal seat of the solution circulating pump This seal face operates in a bath of oil, thereby assuring proper lubrication

In normal operation the pressure on the low side of the unit is between 8 in and 2 in of vacuum and the pressure on the high side is between 22 and 28 lb gauge

These new absorption refrigeration units are remarkably compact For example, the 20-ton unit is 50 in long, 46 in. wide, and 70 in high.

There are now available 5 different sized units ranging in capacity from 3 to 20 tons

The low side and water cooling equipment is of conventional design



Williams Oil-O-Matic Heating Corporation

Bloomington, Illinois

Manufacturers of Automatic and Manually Controlled Fuel Oil Burners

Service to Architects and Builders

CHICAGO, ILL , 641 N Michigan Avenue — NEW YORK, N Y , 1231 Graybar Building

For Williams Air Conditioning Equipment and Ice-O-Matic Refrigeration Equipment, see File Index

A Complete Line

Williams Oil - O - Matic offers a complete line of oil heating equipment, six Oil-O-Matic burner models—a genuine Williams Oil-O-Matic of ideal capacity for every size and type of house, for every apartment, public building or commercial structure

New complete boiler-burner units, a new and revolutionary type of oil-burning automatic water heater, and a new oil-burning range burner for heavy duty ranges are available

Heating Capacities of the Various Williams Oil-O-Matic Burners

Model	Domestic Shipping Weight Lb	Length	Width	Height	HP	RPM	Gals Fuel Oil for Operating Hour Min	Max
K-150	115	30″	14½″	15½″	1/10	1800	½	1½
K-3	145	31½″	15½″	17″	1/10	1800	1	3
K-4, 5	170	32½″	20⅝″	18″	1/5	1800	2	4½
K-7	175	33″	22⅛″	18½″	1/5	1800	3	7
J-1800	255	45″	23″	25″	1/2	1800	7	15
JJ	295	50″	33″	22″	1	1800	15	25

See *The Architect's Handbok of Williams Oil-O-Matic Heating* for additional data

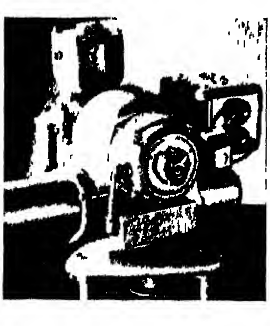
Water Heaters

WHA*	600	58″	23″	28″	1/10	1800		½
WHB†	885	75″	23″	28″	1/10	1800	½	1 0
WHC‡	1380	88″	29″	34″	1/10	1800	1	1¾

*Output 90 F rise, 60 gal per hour †Output 90 F rise, 120 gal per hour ‡Output 90 F rise, 210 gal per hour

Oil-O-Matic Hot Water Supply

A new efficient hot water supply unit made in 3 sizes covering ordinary home use, large home or commercial use, and heavy duty requirements The entire apparatus - including Oil-O-Matic oil burner, combustion chamber, water reservoir, and all automatic controls—is combined in one compact unit Tank being horizontal instead of vertical, permits the use of a unique triple flame travel, the hot fire traversing the length of the tank three times before escaping, transfers all possible heat to the water



The unit illustrated, WHA, will deliver 60 gal of hot water per hour, raised 90 F in temperature, which is equivalent to a constant flow capacity of 1 gal of hot water per minute, using ½ gal of fuel oil per operating hour WHB has twice, and WHC has three and one-half times, the capacity of WHA The WHC is also available as a boiler-burner unit—the WHC-S for steam, the WHC-W for hot water

Underwriters' Listing

Oil-O-Matic burners are listed as standard by the National Board of Fire Underwriters to burn oils conforming to commercial, standard specifications for Nos 1, 2 and 3 fuel oil Each burner carries the Underwriters' label Also approved by important codes and governing bodies

Regulations governing oil burner installations are covered thoroughly in Regulations of the National Board of Fire Underwriters for the Installation of Oil Burning Equipment and for the Storage and Use of Fuel Oils in Connection Therewith

Range Burners

New type Oil-O-Matic range burner burns low-cost fuel oil Economical and efficient Designed for heavy duty ranges, bringing oil heat economy to restaurant, hotel, hospital, steamship, dining car, resort and club The starting and stopping of the motor, the ignition, the varying of the amount of oil burned and of air delivered, are all controlled with one single lever

Engineering Service

Available to architects See A I A File No 30 G-1

The Lamson Company

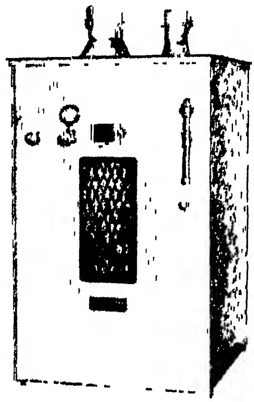
INCORPORATED

Syracuse **Offices in Principal Cities** **New York**

THE LAMSON

DEHUMIDIFIER

for Controlled Humidity



DRY AIR
IN ANY WEATHER
IN ANY QUANTITY
AT LOW COST

The Lamson Dehumidifier is designed to meet the many demands of industry where the control of humidity is a vital part of processing The Dehumidifier provides low humidity at ordinary temperatures A solution of Lithium Chloride is the moisture absorbing agent of the Lamson Dehumidifier applicable wherever controlled, reduced humidity is required such as low temperature industrial drying processing rooms, storage warehouses, in processing hygroscopic materials and many other manufacturing requirements Write for more complete information

Lamson Features

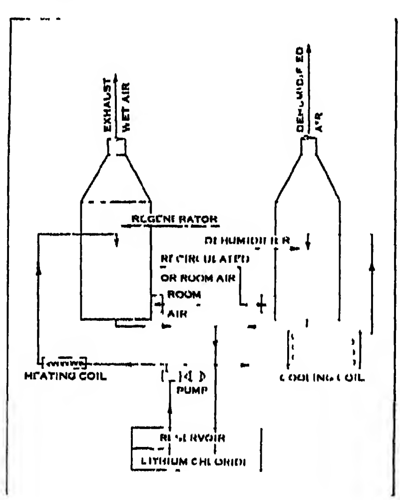
1 Low Dewpoints at ordinary temperatures
2 Uniform Humidity regardless of weather—fully automatic
3. Flexibility of operation — variable capacity.
4 Central Regenerating Unit — Pipe Solution to point of use.
5 Kills bacteria, molds, etc
6 Prevents accumulation of odors
7 Less Heat required low pressure steam
8 Uses only ordinary available water
9 No replacement of Lithium Chloride
10 Low cost of operation

Table I—Equilibrium Dew Points of Lithium Chloride Solutions

Specific Gravity at 75 F	Temperature of Solution and Air at Equilibrium				
	60 F	65 F	70 F	75 F	80 F
1 22	24	29	34	38	42
1 23	22	26	30	34	38
1 24	19	23	27	32	36
1 25	17	21	25	29	32
1 26	15	18	22	26	30
1 27	13	16	20	24	28

Table II—Dehumidifying Effects of Lamson Dehumidifier

Dewpoint of Humid Entering Air	Dewpoint of Dehumidified Air	Water Vapor Removed per 1000 Cu Ft of Air per Minute
75 F	57 F	38 lb per hour
70 F	55 F	28 lb per hour
65 F	52 F	21 lb per hour
60 F	48 F	17 lb per hour
55 F	45 F	12 lb per hour
50 F	42 F	8 lb per hour
45 F	40 F	5 lb per hour



Pittsburgh Lectrodryer Corporation

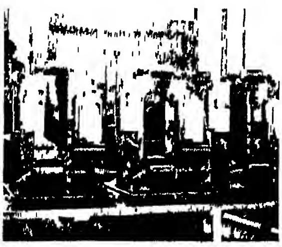
Foot 32nd Street, **Pittsburgh, Pa.**

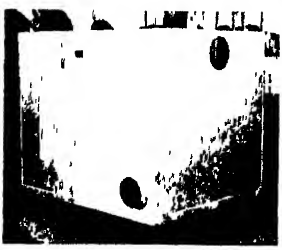
LECTRODRYER

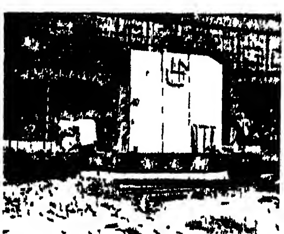
"ACTIVATED ALUMINA SYSTEM"

Trade Mark Reg U S Pat Off

Rugged Industrial Units for Direct Dehumidification in Air Conditioning, Control of Humidity in Chemical Processes, Storage Warehouses, Etc.







What Is It

The air conditioning type of **Lectrodryer** is a ruggedly built piece of industrial equipment designed specifically for partially or completely dehumidifying air These units, when combined with auxiliary apparatus for cooling, filtering and circulation, provide an economical means for complete summer air conditioning, for industrial processing rooms, pharmaceutical houses, theaters, restaurants, office buildings, stores, etc In control of humidities for industrial and chemical processing, the **Lectrodryer** may be used with or without cooling as needed

How It Operates

In the **Lectrodryer** moisture is removed from the air by the physical phenomenon of adsorption using "the Activated Alumina System" The adsorbing agent is activated alumina, a product especially prepared for this purpose by the Aluminum Company of America The moisture is removed from the air as it passes through the activated alumina without physical or chemical change in the activated alumina The activated alumina is reactivated automatically in place The correct amount of the adsorbent is supplied with the machine and no further additions are required

Where It Is Used

The air conditioning type of **Lectrodryer** is designed to operate on a continuous day in and day out basis, if required, to maintain the required low relative humidities **Lectrodryer** equipment enables the maintenance of lower than normal relative humidities in storage warehouses, in pharmaceutical and chemical plants where hygroscopic and effervescent salts are handled, in drying cabinets and similar equipment, for drying cupola blast and controlling moisture content in combustion furnaces, for dehumidifying in connection with comfort installations and similar problems

Advantages

No corrosive chemicals are employed in connection with **Lectrodryer** equipment thus assuring non-contamination of the air stream

Units built for gas, steam or electric reactivation, which means that in many instances waste heat from other processes can be used as a source of reactivating heat

Low maintenance costs are assured as no regular over hauling or replacement of parts is required

Equipment is fool-proof and sturdy

The Lectrodryer Dries with or without Cooling

Sizes

Standard sizes available with rated flows of from 200 to 12,000 cfm and with maximum moisture removal capacities of from 8 lb to 500 lb per hour

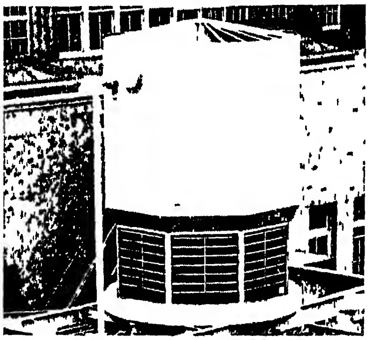
Research Corporation

405 Lexington Avenue, **New York City**

"Water Saved is Money Earned"

COEY MULTI-STAGE COOLING TOWER

(PATENTED)



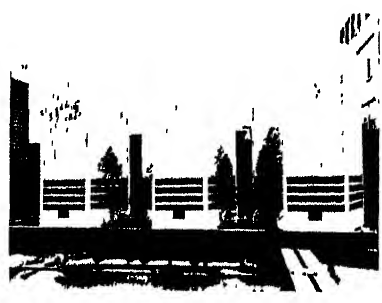
Gives Controlled Cooling.
Minimum Noise Level.
Sprayless Operation.
Architectural Harmony.

Because of its compactness, light weight, spray free and quiet operation, the Coey Multi-Stage Cooling Tower is especially well-suited to roof and basement installations in congested areas, moreover, the highly efficient cooling provided by the multi-stage principle makes this unit desirable for installation anywhere

The Coey Multi-Stage Cooling Tower is constructed of a copper bearing steel shell, nickel iron castings, Red Gulf Cypress wood baffles, a non-overloading reverse blade centrifugal fan rotor, and the Coey Spray Eliminator (Patented) of copper bearing steel

Air Movement—non-overloading centrifugal rotor, high efficiency, and variable speed **Principle of Operation** —Three-pass, counter-current flow No ice formation on fan rotor during Winter operations

No	GPM	Sides	Short Diameter	Height Above Beams	Net Weight Tower	Est Weight of Water	Total Oper Weight
4	40	6	3'-2"	6'-4"	1000	500	1500
10	100	6	5'-0"	9'-0"	3000	1300	4300
20	250	6	9'-0"	14'-9"	10000	3500	13500
35	375	6	10'-0"	14'-9"	13000	4500	17500
50	550	6	11'-9"	17'-8"	16500	6500	23000
85	850	8	14'-6"	18'-6"	24000	11000	35000
110	1100	8	16'-9"	21'-0"	35000	14000	49000
150	1400	8	19'-0"	25'-0"	44000	19000	63000



Research Air Conditioning System

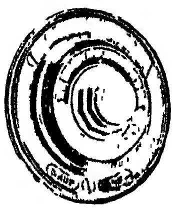
Removes excess moisture by chemical absorption, with an inexpensive compound known as "Caloride" The system has two distinct functions, (1) the control of humidity and air purity for industrial applications, including the drying and purification of various gases, and (2) the control of effective temperature and air purity for human comfort and health

Other Equipment: Cottrell Electrical Precipitation Systems Multiclone Dust Collectors— Impact Separator- Cottrell Royster Deodorizer Royster Stove for High Temperature Heat Exchange.

Young Regulator Company

Department G

4500 Euclid Avenue, **Cleveland, Ohio, U. S. A.**



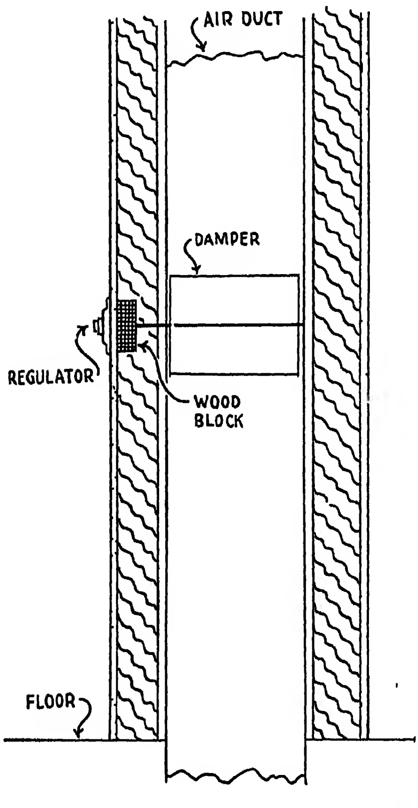
The Young Regulator is simple to install, with direct connection to the damper It is made of rust-resisting metal and presents a neat appearance Finished in plain, nickel and bronze Any other finish if desired

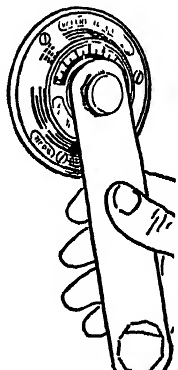
YOUNG REGULATOR

A Convenient Control of Air Conditioning and Ventilation

The Young Regulator controls the volume of air flow thru a duct by opening and closing a volume damper with which it is connected by a $^3/_8$ in square rod Contrary to all past practice the damper can now be placed near the grille outlet, and the Young Regulator installed on the wall surface, where it is both decorative and easily accessible

In air conditioning installations, it is essential that all dampers be securely locked in position, after proper air flow has been adjusted, otherwise the system cannot function properly—The Young Regulator permits accurate setting of the damper and positive locking, so that no unauthorized person can tamper with the damper and throw the entire system out of balance





For service in schools, hospitals, residences, offices, banks, factories and public buildings, the operator in charge of air conditioning and ventilating finds it easy with the device, to establish and maintain the proper volume of air, free from interference or meddling

For Locking a Volume Damper in a Permanent Position

With the handy wrench the damper is readily set to admit the desired amount of tempered air, and locked in that position making it tamper proof The device for adjusting is the outermost nut, or head, with 8 sides, turned by wrench Behind the adjusting nut is a set nut with 10 sides, of larger diameter which, with a turn of the wrench, will lock the regulator at any desired position or release it This nut is difficult of access by any ordinary wrench

The Air-Maze Corporation

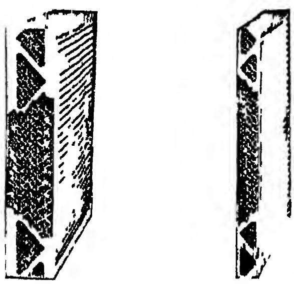
813 Huron Road, Cleveland, Ohio

ENGINEERS AND MANUFACTURERS OF AIR FILTERS EXCLUSIVELY

DIRECT FACTORY REPRESENTATIVES IN NEW YORK, PHILADELPHIA, DETROIT, CLEVELAND, CHICAGO TULSA AND SANTA BARBARA

DISTRIBUTORS IN PRINCIPAL CITIES AND TOWNS THROUGHOUT THE UNITED STATES

During more than a decade devoted exclusively to air filter engineering and manufacturing, a great deal about the control and elimination of dust, dirt, pollens and grit has been learned by Air-Maze engineers Their design and development of a unique type of filter element construction, embodying distinctive advantages, has been considered a worthy contribution to the air filtering science and has resulted in wide acceptance of Air-Maze air filters in all fields of application



4 in Thick Panel *2 in Thick Panel*
Cut-away views showing open end edge of Air-Maze Permanent Cleanable Panel Filters

Note Advantages Made Possible by Air-Maze Scientific Construction:

Costs Little to Clean –The separating layers and exact spacing of baffles permit free washing action between and around all baffles Cleaning and oil charging is done at the same time quickly, simply and economically

Great Dust Capacity– On the face are coarse baffles exactly spaced, which evenly impinge bulky particles Progressively inward are finer meshes Instead of one or two progressions in density there are many; consequently, great dust capacity is possible enabling long operating periods before servicing is required

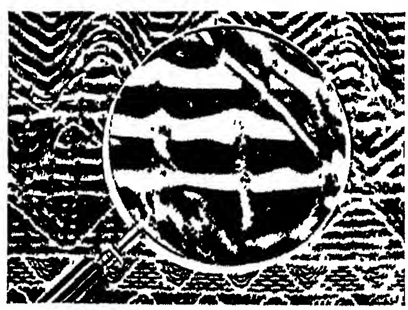
Vibration Proof—Vibrations in service cannot shake filter media out of position the uniform density remains *permanently* perfect, **no replacements are necessary!**

No Special Oil Required-- In Air-Maze each square inch of the panel handles the same amount of air Properly drained, as directed, *no oil will be carried through*, since the air velocity is held evenly low and uniform in every square inch Therefore, especially adhesive oil is unnecessary

Low Restriction The relatively "open fabrication" of Air-Maze is only possible because of built-in *exact* density Such construction enables low pressure drop which remains proportionately low during long operating periods

No Clogging - Because Air-Maze panel filters are easy to *completely* clean and since the exact density enables uniform deposit of dust, no clogging can occur

Adaptability In addition to air conditioning and power equipment installations Air-Maze panel filters are effectively used in humidifiers, water eliminator units, paint spray-booths, oil separators, and other applications where specific problems and unusual requirements are easily handled by adaptations of the panels Air-Maze panels will be made to fit frames of existing installations and can be furnished with locking handles and latches, or with flanged edges and lift handles



Magnified Section of "Loaded" Air-Maze Air Filter Element Note that dirt has been quite evenly impinged on only one side of the wires No obstructed spaces can be seen This feature accounts for the Low Pressure Drop and Non-clogging characteristics of Air-Maze

TECHNICAL INFORMATION

Sizes Stock sizes are two and four-inches thick, 10 x 20, 20 x 20 and 16 x 21 in All other sizes are made to order at no additional fabrication costs

Capacity - The recommended air capacity is 1½ to 2½ cfm per square inch for normal industrial installations Thus the recommended capacity of a 20 x 20 in Air-Maze Panel is 600 to 1000 cfm

The Air-Maze Corporation

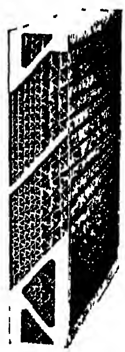
813 Huron Road, **Cleveland, Ohio**

Resistance—For 2 in thick panels the resistance varies from 0 10 in to 0 16 in H_2O when handling 2 cfm per square inch of filter area, and for 4 in thick filters the resistance varies from 0 126 in to 0 146 in H_2O, the variation being in accordance with the different types of filter media construction available To obtain specific restriction data write for graph RE-1

Construction—Air-Maze filters are of patented construction consisting of a maze of alternately placed and exactly spaced flat and crimped galvanized wire screens of selected meshes, these are arranged with precision so as to create graduated and progressive density, and to positively embody the baffle impingement principle The filter element is enclosed in a 16 gage metalescent enameled steel frame having a "Z" bar open end to simplify servicing

AS EASY TO CLEAN AS A. B. C.

Wash out filtered matter in a pan of light oil The exact density makes it easy to do this



Cut-away View

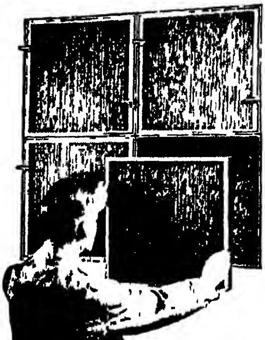
Set the panel on one edge, with open end down to drain for about 20 min This drains off excess oil

From a flat surface raise one end and let it drop sharply several times This completes drainage

Cleaning Instructions—When cleaned as illustrated a film of oil is left on the filter media and no further dipping or charging is required Filters may be cleaned in gasoline or kerosene if more convenient, in which case they should be allowed to dry and then dipped into thin oil before placing in service.

Kind of Oil to Use—A very thin oil is always used in order that cleaning and drainage will be facilitated Expensive, especially processed oil is not necessary with Air-Maze Inexpensive oils of 75 or less, SUV viscosity, at 100 F, obtainable from oil companies at a nominal cost, should be used

AIR-MAZE INSTALLATION FRAMES



Air-Maze Panel holding frames assure efficient attractive installations

Air-Maze Panel Holding Frames are constructed of metalescent enameled 16 gage channel-formed steel having rolled front edge and ¾ in flanged back edge A thick felt lining on inside of flange insures against air leakage when panels are in place Each frame is a complete unit and may be used alone in single panel installations, or many units may be bolted together, with a felt seal between edges to make a large bank of filters Every unit is fitted with latches for the locking handles supplied on panels

In determining frame sizes, ⅝ in. is allowed over both width and length dimensions of the panel filters This ⅝ in. includes frame edge, clearance and felt seals between edges

Specify Air-Maze—for all air filter installations and you will be assured of *efficient, economical* PERFORMANCE

Engineering Service Available—The Air-Maze Engineering Department will gladly offer installation suggestions for special air filter applications

Other Air-Maze Products—In addition to the panel types, Air-Maze Corporation also manufactures a complete line of circular shaped air filters for use in various Railroad, Industrial and Automotive applications

Literature Available—Bulletins PAN-36 and BPAN-86· Covering complete panel air filter details Graphs G-11 and G-12. Showing results of tests made with A S H V E. Code Test Apparatus Brochure RR-95 Describing Railroad Air Filters Catalog 5-M-86 Covering complete line of Air-Maze

AMERICAN AIR FILTER COMPANY INC.

INCORPORATED

1st Street and Central Avenue, **Louisville, Ky.**

Representatives in Principal Cities

Dust Engineering—Dust Engineering is that branch of applied science which deals with the origin, nature and characteristics of the small solid air-borne particles called "dust," and the development of methods, processes and apparatus for its control or elimination

The American Air Filter Company, Inc, has had an important part in advancing the science of Dust Engineering The efforts of its Research and Engineering Staff for the past twelve years have been devoted exclusively to the study of dust problems and the development of a complete line of air cleaning equipment for modern air conditioning, building ventilation and the control of process dust in industry

American Air Filter products, therefore, not only embody the knowledge accumulated from years of constant research and the experience gained from designing, building and applying thousands of air filters, but are backed by ample technical and financial resources to insure their outstanding position in the Dust Engineering field

Products—American Air Filters are available for every condition, with operating characteristics and efficiencies to suit specific problems In general, there are two distinct types based upon the "viscous film" and "dry mat" principles Each type is made in several styles which differ in method of operation, servicing, space required and initial cost to meet the various conditions encountered in air cleaning problems A discussion of various filter types will be found in the Technical Data Section under "Air Cleaners"

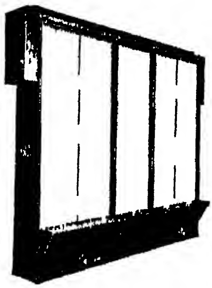
Air filters are generally used for the removal of dust, dirt, bacteria and other foreign matter from the air and are applied to general ventilation, modern air conditioning, process dust control, for air compressors and Diesel Engines, mill motors, turbo-generators and other electrical applications, and for air or gas under pressure to remove entrained oil, moisture and dirt

Air Filters In Air Conditioning Filtered air is today recognized as essential in modern air conditioning There are other important factors which contribute to our comfort such as temperature, air movement and humidity, but science today emphasizes the prime necessity of pure air for health and efficiency

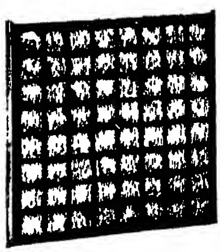
Air cleaners have, of course, always been considered an integral part of large central systems These are usually of the fully automatic type such as the Multi Panel filter, illustrated in the accompanying photograph

There are now available to manufacturers of unit air conditioners moderate priced unit filters such as the Renu filter, the Throway filter, and other types of filters illustrated on this page.

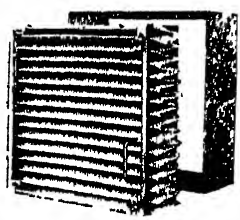
The Renu filter is an entirely new departure in air filter construction It consists of a permanent metal frame provided with a removable cover and renewable filter pad The cover



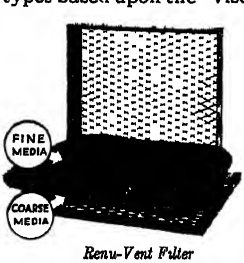
Automatic Multi-Panel Filter



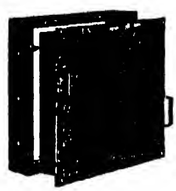
Throway Air Filter



Airmat Type PL-2½ Filter



Renu-Vent Filter



Standard Viscous Unit Filter

is easily removed without the use of tools, and filter pad can be lifted out and replaced with a new one at very small expense in less than a minute's time

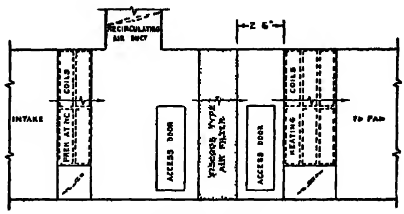
The Throway filter, as the name implies, is an inexpensively constructed unit designed to be discarded after it has served its maximum period of usefulness and replaced with a new filter unit The Filter pad is enclosed in a perforated cardboard container which makes it possible to dispose of the dirty filter by burning it in a furnace or incinerator

There is probably no single item which costs as little and may mean as much in the design of an air conditioner as air filtration These units are furnished in any dimensions or shapes desired They are usually built in units handling 400 cfm and from 2 in to 4 in thick They are usually made in the following sizes - 20 x 20 in, 16 x 25 in and 16 x 20 in High cleaning efficiencies can be secured, with a resistance to air flow ranging from $^1/_{16}$ in to $^3/_8$ in water gauge

Automatic Self-Cleaning Air Filters The American line of automatic air filters is complete, the most popular types being Multi-Panel, Horizontal and Phoenix Filters

All types are furnished for either continuous or intermittent service and are available in sizes and set-ups suitable to any desired capacity or space condition

Standard Viscous Unit The American Unit Air Filter incorporates the time tested unit principle of construction Each unit consists of a standard steel frame and interchangeable cell equipped with automatic latches to facilitate removal for cleaning and recharging

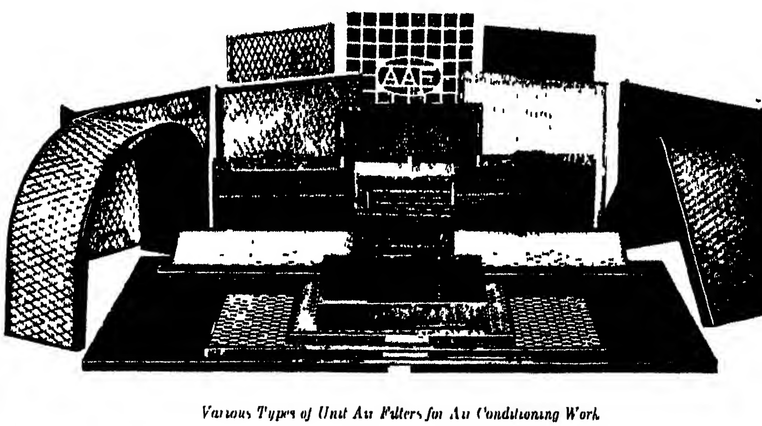


Typical Installation

Airmat Filter Dry Type The filtering media in this type is the Airmat sheet, a dry filter mat composed of thin sheets of gauzy, cellulose tissue The Airmat sheets are supported in screen pockets mounted in a unit frame of box-like construction These unit frames can be set up to meet any capacity requirement or space condition The Airmat sheets are renewable - their life depending on the dust condition and hours of daily service

Airmat filters are used both for air conditioning and industrial air conditioning In the latter field they are particularly well adapted for the recovery of valuable dusts and for abating the dust nuisance which confronts so many industrial plants Airmat filters are available in two types, the PL-21 as illustrated and the Well Pocket type unit

Our standard data books and catalogues are to be found in most engineering files or libraries We will be glad to furnish full data to engineers or manufacturers interested in this subject



Various Types of Unit Air Filters for Air Conditioning Work

Coppus Engineering Corporation

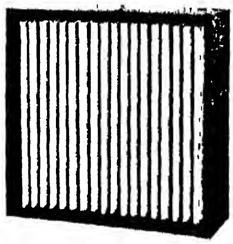
339 Park Avenue, **Worcester, Mass.**

MANUFACTURERS OF AIR FILTERS, STEAM TURBINES, FORCED DRAFT BLOWERS, COOLING FANS

"COPPUS AIR FILTERS PASS CLEAN AIR"

Coppus Unit Filter

Coppus Unit Filter—The Coppus Unit Filter is of the dry type using a removable filter glove of all-wool felt and supported by a rigid, welded distender frame which is adjustable so that after assembly the filter glove may be stretched and held tautly inside of the filter casing, giving the pockets a tapered shape so essential for an even air flow All metallic parts are rust-proofed (Bonderized, Cadmium Plated), and Duco painted



Air Intake Side of Stationary Unit Filter

Specifications

Normal Rating 800 cfm
Resistance when clean 2 in W G
Dust Arrestance (cleaning efficiency) 99 61 per cent (Tested in accordance with A S H V E Standard Code for Testing and Rating Air Cleaning Devices Used in General Ventilation Work)
Dimensions 20 in by 20 in by 6½ in
Weight per unit 25 lb

Cleaning Filter Elements with Portable Vacuum Cleaner

Outstanding Advantages

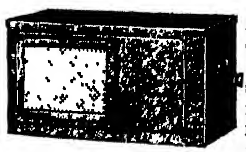
1. It has an exceptionally high dust arrestance
2. It maintains a high dust arrestance even under diverse conditions of neglect
3. Its operation is not impaired by atmospheric conditions
4. It is a Medium Air Resistance Type (Class C) according to the A S H V E Code for Air Cleaning Devices
5. It is easily and quickly cleaned without removing the filter element
6. Its cost of upkeep is very low because the *permanent* filter element is reconditioned periodically with a vacuum cleaner
7. It combines scientific knowledge and practical engineering methods with highest quality of material and workmanship

Coppus Dry-Matic Filter

Automatic Self-Cleaning Dry-Type Air Filter—It uses a specially woven cotton textile as filter medium arranged in the shape of an endless belt in zig-zag fashion over rolls The horizontal pockets have a tapered shape thus securing an even rate of air flow over the whole effective filter area. At predetermined intervals the filter curtain is automatically moved by a small geared motor over the rolls, thus passing over a suction nozzle. For certain types of dust the filter is also equipped with a beater which, at predetermined intervals, automatically reconditions the filter curtain The cleaning and reconditioning operation takes place while the ventilating system is in operation. Built in capacities from 2000 cfm up.

Coppus Window Air Filter

It supplies a continuous flow of filtered air, is extremely quiet in operation, keeps out street noises, eliminates dirt and dust including invisible particles, and last, but not least, is very effective against rag weed pollen in concentrations commonly found in the hay fever season Its use is recommended for offices, homes and hospitals



Air Filters for Compressors and Internal Combustion Engines.

Steam Turbines, Horizontal and Vertical, 1 to 150 Hp.

Forced Draft Blowers.

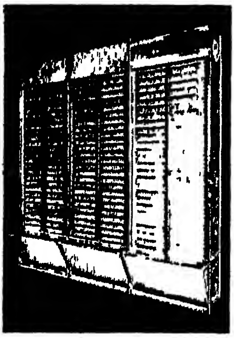
Portable and Cooling Ventilating Fans.

Write for Complete Bulletin on Any Filter

Independent Air Filter Co.

215 West Ohio Street, **Chicago**

Representatives in All Principal Cities



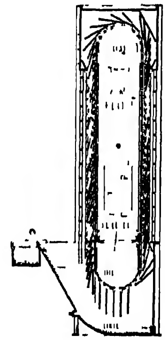
Three Section Group, 9 ft 3 in high by 9 ft 0 in wide, capacity 30,000 cfm "Double - Duty" filters are built in capacities from 3,000 to 600,000 cfm

DOUBLE-DUTY

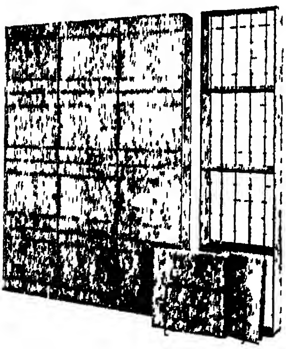
Self-cleaning Non-clogging

Maintenance Free

Designed for large installation requirements, where automatic operation and unvarying air flow are important factors The filtering principle is that of true impingement on viscous coated metal surfaces Endless filter curtain composed of die stamped steel louvre plates, heavily indented and disposed in overlapping form Acute deflections of individual fins, constantly coated with heavy oil, result in high cleaning efficiency Entirely self-cleaning, regardless of dust content of air Self-draining design of curtain plates eliminates danger of oil entrainment



Cross-section through filter Sludge is removed at intervals of two to six months Oil consumption negligible - never changed, merely replenished



For group systems, sectional frames are furnished as illustrated Frames are fabricated in the form of a rigid vertical section of the required units in height

Servicing operation unloading, refilling and replacing - consumes at the most 3 min per unit Useful life of filter medium varies from three to six months depending on individual conditions

Simple method of detaching unit from frame will be noted above Improved style locking device clamps felt seal firmly against main frame

PERMO PAD

Washable Waterproof

Acidproof

For general air filtration purposes A highly resilient filter expanding tightly against its frame providing a perfect seal against air leakage PERMO PAD is made entirely self-supporting by an expanded metal grill incorporated in the rear side by a process that makes it an integral part of the unit Frame size (overall dimensions) 20 x 20 x 6 in Rated capacity per frame unit—800 cfm Initial resistance (tandem arrangement) 0 18 W.G

KOMPAK

A Dry Fabric, Low Velocity Air Filter

With Renewable Medium

For general ventilation and industrial air cleaning Filtering principle that of straining air through fabric material at low velocity Standard filter medium high grade, fluffy cotton material with high cleaning efficiency and remarkable dust holding capacity Each unit contains 27 sq ft of filter medium, which, at normal rating of 600 cfm results in a velocity of 22½ fpm through medium Initial resistance 0 15 in W G.

Engineering Information Available on Request

INDUSTRIAL MATERIALS DIVISION

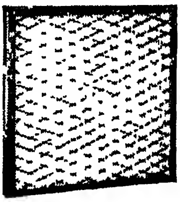
Owens-Illinois Glass Company

Toledo, Ohio

AIR FILTERS FOR APPLICATION TO RESIDENTIAL, COMMERCIAL *and* INDUSTRIAL HEATING, VENTILATING *and* AIR-CONDITIONING SYSTEMS

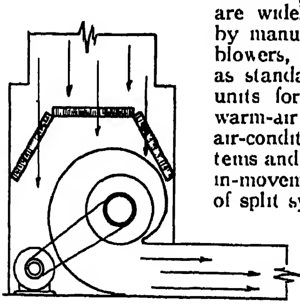
DUSTOP

"FIBERGLAS" MEDIUM - ADHESIVE-COATED - REPLACEMENT TYPE



The Dust-Stop Air Filter consists of a series of mats of Fiberglas, progressively packed - -coarse glass fibers of lesser density at the intake face and fine glass fibers of greater density at the discharge face These mats are coated with a non-evaporating fireproof adhesive having extraordinary wetting power, retains its viscosity under operating temperatures ranging from 15°F below to 300°F above zero, will not flow off or charge the air with molecules Dust-Stop Air Filters are engineered to provide high efficiency at low cost of installation and maintenance

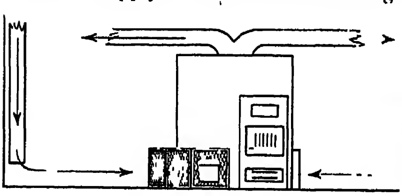
BLOWER APPLICATIONS



Dust-Stop Air Filters are widely accepted by manufacturers of blowers, engineered as standard in their units for residential warm-air heating and air-conditioning systems and for the air-in-movement phase of split systems

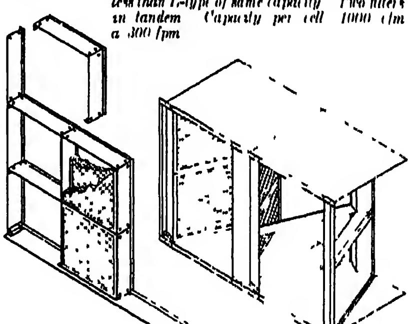
GRAVITY WARM-AIR FURNACES

Furnace manufacturers have developed a filter boot application of standard 20 in x 20 in Dust-Stop low-resistance Air Filters which encircle the furnace, with openings at the base of the casing introducing a uniform supply of cold air to the castings



Dust Stop Air Filter frame assemblies are standard in the units of many manufacturers and are also installed by engineers of commercial and industrial heating, ventilating and air conditioning Frame members of cold rolled steel are assembled vertically or horizontally in combinations to provide for any cfm and space requirement Dust Stop Filter Frame assemblies are complete with nuts, bolts, gaskets and filters Efficiency 97 per cent (tested according to A S H &V E Standard Code for Testing and Rating Air Cleaning Devices Used in General Ventilation Work)

(LEFT) *"L" member assembly Cell size 20 in x 20 in Two filters in tandem Capacity per cell 800 cfm at 300 fpm* (BELOW) *"V" member assembly Frontal area 40 per cent less than L-type of same capacity Two filters in tandem Capacity per cell 1000 cfm a 300 fpm*



SPECIAL APPLICATIONS

Dust-Stop Air Filters are standard equipment in individual room air conditioners, window ventilators and circulating units for the removal of pollen and dust carried bacteria Enclosed units for fresh air intake in commercial and industrial ventilating and air-conditioning systems Fiberglas Eliminator Mats are used alone or in conjunction with regular Dust Stop Filters in cleaning air and removing particles of moisture from the air in systems which employ water spray for cooling and dehumidification

H. J. Somers, Inc.

6063-69 Wabash Avenue, **Detroit, Mich.**

Fig 1—"V" Type—soldered construction

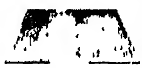


Fig 2—Flat type with flat frame

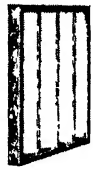


Fig 3—"W V" Type—all-welded construction Filter Pack removable

The Somers Filter Mat requires no adhesive material to catch the dust particles of the air stream in passing between the fibers Its construction is such that it forms an effective dust barrier that is highly efficient, the range depending on the type of pack, air stream velocity and the character of the dust Cleaning the filter is easy and it is a most desirable construction for use where operating velocities or temperatures are relatively high or where a high degree of air purity must be secured

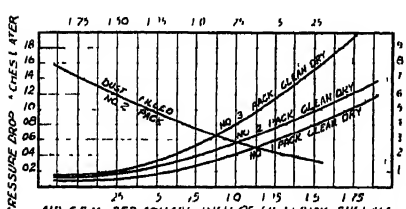
"Hair Glass" in this form reacts with practically none of the chemical elements carried in the air stream It is easy to maintain in proper functioning condition as "Hair Glass" neither rots nor disintegrates in service "Hair Glass" is pliable and tough, it is both odorless and non-absorptive Consequently, the user has wide latitude in his methods of handling and cleaning the unit

The Somers Filter does not have to be replaced as soon as it fills up with dust The entire pack may be washed with hot water, cold water, or chemical solvents and as often as necessary, without in the least impairing its operating efficiency

Standard frames and mesh are galvanized or alloy although other materials can be furnished if required

The Somers filter may be assembled in different thicknesses

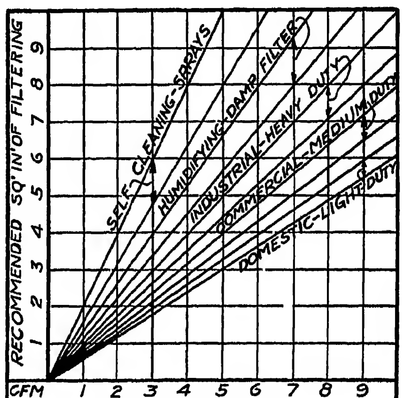
No 1 Pack—The thinnest, is used where the resistance must be kept low It is usually employed for domestic applications, recirculating systems and gravity flow



No 2 Pack—Is generally used for all commercial and industrial applications It will arrest liquid particles in the air stream

Filter units can be made to exact dimensions

Bank holding frames can be supplied Give outside dimensions



—PROPORTIONAL—
FILTER SELECTION CHART

CODE LETTER	FRAME TYPE	FILTER FRAME DIMENSIONS HEIGHT	NOTE	WIDTH	SOLDERED DEPTH	WELDED VEE DEPTH	AREA IN INCHES PROJECTED	RATIO VEE TO ONE	FILTERING MEDIUM	FILTER ONLY WEIGHT IN POUNDS	FILTER TYPE	FIGURE NO
A	FLAT	20	XY—AS NEEDED, TO FIT OPENING	20	1/4		400	1 1	400	3 25	FLAT	2
B		16		25	1/4		400	1 1	400	3 25		2
FF		X		Y	1/4		XY	1 1	XY	008XY		2
J	DEEP	20		20	1⅞	2	400	1 2 1	480	5 25	VEE	1
K		16		25	1⅞	2	400	1 2 1	480	5 75		1
Q		X		Y	2½	2⅝	XY	1 5 1	15XY	016XY		1
R		X		Y	2¾	2⅞	XY	2 1	2XY	020XY		1
S		X		Y	2 15/16	3 1/16	XY	2 5 1	25XY	023XY		1
T		X		Y	3¼	3⅜	XY	3 1	3XY	.025XY		1
U		X		Y	3 1/16	3¼	XY	3 5 1	35XY	027XY		1
V		X		Y	3 3/16	3 5/16	XY	4 1	4XY	029XY		1
W		X		Y	3 5/16	3 7/16	XY	4 5 1	45XY	030XY		1

STATE 'X' AND 'Y' DIMENSIONS IN ORDER GIVEN VEE RIDGES RUN VERTICALLY, —PARALLEL TO 'X' DIMENSION

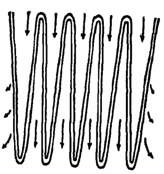
Staynew Filter Corporation

Air Filters for Every Purpose

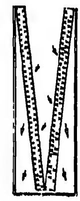
6 Leighton Ave. **Rochester, N. Y.**

PROTECTOMOTOR DRY TYPE FILTERS

(For removing foreign matter from air at atmospheric or other pressures, with various types for building ventilation, dust recovery, oxygen chamber and all air cleaning purposes.)



Cross Section of Panel Insert



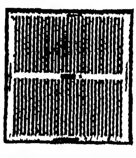
Cross Section of Multi-V-Type Insert

One basic principle—the fin or V-Type construction is employed in a slightly different manner for every Protectomotor dry filter This basic principle permits (1) a large area of filtering medium to occupy the smallest possible space, and (2) the intake currents to move parallel to the filtering surface at low velocity Protectomotor Dry Filters require no oil or other adhesive material to catch dust clean, dry, odorless air is assured Authorities agree that the positive dry type filter is most efficient in stopping the smaller air borne particles Protectomotor dry filters actually prevent the passage of pollen, bacteria and soot

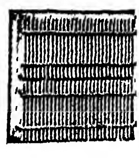
Cleaning is easily and quickly effected during operation by use of a special vacuum cleaner nozzle Protectomotors operate from 3 months to a year without attention under average conditions Panel Units average slightly longer wear than Multi V-Type Units several years at least without replacement

Resistance to the flow of air is less than 0 25 in water gauge for Panel Units or Multi-V-Type Units when operated at rated capacity

Panel Units: These consist of a panel insert and frame The panel insert is composed of two rows of 60 hollow loops or fins 6 in in depth These fins are formed of special rust resisting embossed wire mesh, supported by a retaining grate of steel or aluminum and a similar spacing grate Each row of fins is covered with a single piece of Feltex filtering medium. This medium is a felt-like material, especially prepared for the application Specifications below



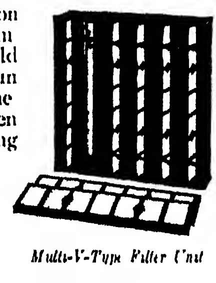
Front new Panel Unit showing spacing grate and frame



Rear new Panel Unit showing Fin Construction

Size of filter and frame	20 x 20 x 6⅝ in
Size of filter panel	19½ x 19½ x 6 in
Capacity of filter unit (normal air)	800 c f m
Surface of filtering medium	42 sq. ft.
Linear velocity of air through filter medium	18 f p m
Resistance of clean filter to flow of air	¼ in water gauge
Weight of filter panel .	18 lb
Weight of filter panel and frame	28 lb

Multi-V-Type Filter Unit: Similar in fundamental construction to the Panel Unit, but employing a slightly different filtering medium The Multi-V-Type medium is of closely pressed cotton fibres held between two sheets of cotton gauze The medium is arranged in patented V shaped pockets in a fibre board and pressed metal frame These patented cells can be quickly and inexpensively replaced when worn out Their arrangement makes possible an active filtering surface of 27 times face area—the highest ratio of filtering surface to face area of any Protectomotor, making the Multi-V-Type ideal where face area is limited



Multi-V-Type Filter Unit

In certain installations the Multi-V-Type is more practical than the Panel Unit because its construction fits the available space better, or because it is lighter in weight per square foot of filtering area

Space is lacking to give complete specifications, but these will be mailed promptly on request

Staynew Filter Corporation

Air Filters for Every Purpose

6 Leighton Ave. **Rochester, N. Y.**

PROTECTOMOTOR AIR FILTERS

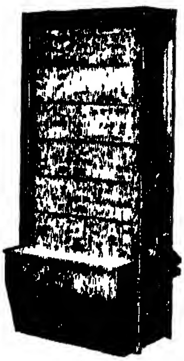
PROTECTOVENT

A Window Ventilator unit that supplies clean air at all times to home or office. Motor driven fan, filter and silencer combined in a practical device with many obvious applications. No draft—no outside noises! Helpful to hay fever sufferers, since the Multi-V-Type filter medium removes 99.9 per cent of all pollen and dust as well as 98 per cent of bacteria.

Protectovent Window Ventilator

PROTECTOMOTOR IMPINGEMENT AUTOMATIC AIR FILTER

(A filter for efficiently and economically handling large volumes of air for all ventilating purposes.)



Automatic Filter

In this model foreign matter is removed from the air by impingement on *two* endless revolving curtains. Most filters of this type possess one curtain only. The first curtain passes through an oil bath, where foreign matter drops off and sinks to the bottom. This oil bath moistens the curtain thoroughly with oil.

The second curtain does not pass through the oil bath. Its purpose is to arrest the slight amount of oil entrained in the air stream from the first curtain and any finer particles of dust that might remain in the air stream. The second curtain makes clean, dry air a certainty.

Two additional exclusive features are to be noted. One is the compressed air cleaner system (see diagram) consisting of copper tubes drilled with holes, placed at the lower edge of each curtain and connected with an air line. These cleaners provide a positive method of cleaning the curtains from time to time and what is even more important—prevent oil entrainment. Such a method is desirable under conditions of heavy dust concentration.

Another exclusive feature relates to the direction of curtain travel. Ordinarily in filters of this type, the direction of rotation is such that dust collected on the intake side is carried over to the outlet side before passing through the oil bath. In this Protectomotor model, dust collected on the intake side is removed by passing through the oil bath before the curtain which had been carrying the dust returns to the outlet side.

Curtain travel is intermittent, rotating 4¾ in. during the operation of the mechanism which is approximately 20 seconds each half hour. The mechanism is actuated by a 1/6 h.p. motor controlled by a telechron motor for A.C. circuits or electrically wound clock for D.C. A.C. timing devices are adjustable for various periods of operation.

The oil bath is charged with Pingene, a highly viscous, odorless, germicidal liquid, with high fire and flash points and low pour test. Average operating resistance is 0.35 in. water gauge.

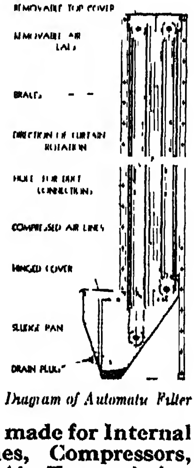


Diagram of Automatic Filter

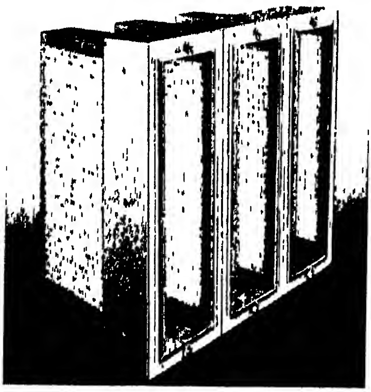
WRITE FOR CATALOG MENTIONING SPECIAL INTERESTS

Protectomotors also made for Internal Combustion Engines, Compressors, Turbo-Generators, Air Transmission Lines, etc.

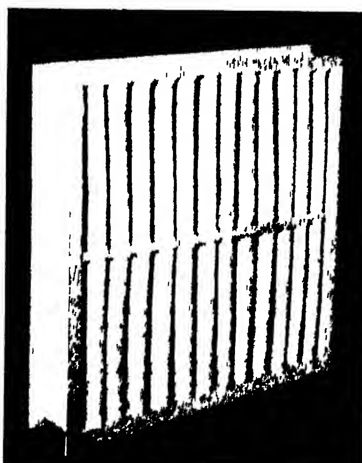
Universal Air Filter Corp.

332 West Michigan Street

Duluth, Minn.



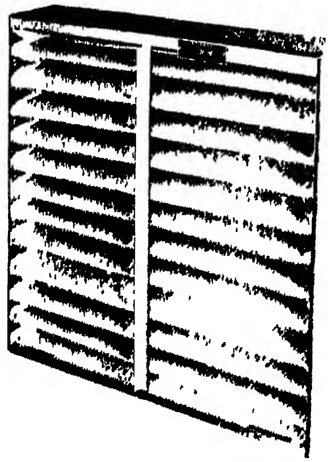
Universal Type C-2 Basket Construction



Universal Type E Replacement Filter

EFFICIENT PRACTICAL

Filtering media is a cotton like sheet designed to arrest passage of dust, dirt and foreign material from the incoming air quietly and efficiently No odor added Filters easily disengaged, media quickly removed



Universal Type C Sectional or V Type Frame

REFILL FILTERS

Universal Type C-2

Basket type construction of 3 to 5 baskets (optional) supporting cotton like sheet of filtering media between screens Baskets are supported in horizontal position by steel frame formed in sections 25 in x 25 in Capacity 2000 cfm Resistance 0 10 wg when clean

Universal Types C-3 and C-4

Combine low first cost and low upkeep in one filter By means of wire screens supporting cotton like sheet in zig zag shape, filtering area is greatly increased Refilled by removing screens and media from steel frames, inserting clean media between screens and replacing assembly in frame

Type C 3 Capacity 1000 cfm

Type C-4 Capacity 750 cfm Resistance clean, 15 wg

REPLACEMENT FILTER

Universal Type E

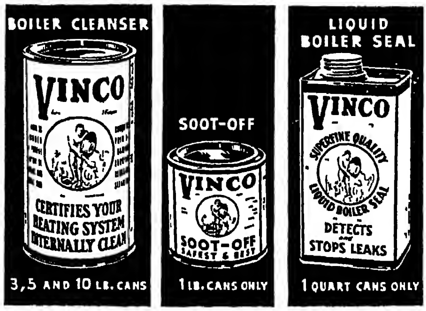
Cotton sheet media with paperboard case fitted to 20 x 20 sectional steel frame Light, efficient, easily replaced, and completely disposable Adapted to fit various types of frames now in use Capacity 600 cfm Resistance 0 15 wg when clean.

The Vinco Company, Inc.

305 East 45th Street — New York, N. Y.

VINCO BOILER CLEANSER

A positively harmless insoluble powder cleaner for new, remodeled and old heating systems. A unique, scientifically processed compound on a special formula not to be confused with other powder boiler cleaners.



Vinco for Old Systems

Annual cleaning of the old heating system adds years of life to the boiler, prevents rust deterioration and saves much fuel and fire attendance Only half quantities given in specification table

Our Three-Fold Guarantee

1 VINCO contains no potash, lye, soda of any kind, oil, acid or other harmful ingredients *Leading boiler manufacturers have placed tags on their boilers advising against use of acids or alkalies*
2 Purchase price is refunded if results are not as claimed when VINCO has been used according to directions
3 Your time, money and comfort are further safeguarded by

Our Free Laboratory Service

Saves thousands of dollars by analyzing the boiler water *before* making needless mechanical changes If desired, the boiler water is again examined *after* the VINCO treatment is completed, and "certified chemically correct" for boiler operation (Write for details)

Adopted By

*American Radiator Co
*Burnham Boiler Corp
Hoffman Specialty Co
International Heater Co
*National Radiator Corp
Sarco Co
Titusville Iron Works Co
*United States Radiator Corp
and many others

*Vinco Distributors

What Vinco Does

VINCO permanently removes oil, grease, scale, rust and dirt from the internal surfaces and from the boiler water *without the labor of blowing boilers over the top*

By this thorough cleansing Vinco stops foaming, priming, surging, and slow steaming

How Vinco Works

Each minute grain of VINCO powder absorbs several times its own weight of oil, rust and dirt These larger grains of absorbed impurities then settle and are blown through the bottom, according to directions on each can

Vinco Specifications for New and Remodeled Steam, Vapor and Hot Water Heating Systems

Cleaning the System — Upon completion of the installation, the contractor shall clean the system by the VINCO method to remove oil, grease, rust and dirt from the boiler using † lb of VINCO in exact accordance with manufacturer's directions (Boilers should not be blown down under pressure, nor should acids or alkalies be used)

This compound must remain in the boiler for at least 10 days of normal operation—not longer than 60 days At the end of this period, boiler must be thoroughly drained and flushed before refilling with clean water

†In writing specification, insert in this space number of pounds of VINCO to be used in accordance with the following schedule For systems having actual installed radiation of

Up to 350 sq ft	—	3 lb
351 " 600 " "		5 "
601 " 1100 " "		8 "
1101 " 1400 " "		10 "
1401 " 1800 " "		13 "
1801 " 2100 " "	—	15 "
2101 " 2700 " "	-	18 "
2701 " 3100 " "	-	20 "
3101 " 3700 " "	—	23 "
3701 " 4200 " "		26 "
4201 " 4600 " "	—	28 "
4601 " 5000 " "		30 "

Above 5000 sq ft use an additional pound of VINCO for each additional 300 sq ft of installed radiation

VINCO SUPERFINE LIQUID BOILER SEAL

A different liquid leak seal Unique in that it does not induce priming and foaming It has no unpleasant smell Makes speedy and permanent repairs of boiler and heating system leaks Fine to tighten up new jobs Directions simple

Quantities

Steam and Vapor Systems--Use 1 quart VINCO Liquid Boiler Seal to each 6 sq ft *grate* area

Hot Water Systems—Use 2 quarts VINCO Liquid Boiler Seal to each 6 sq ft *grate* area

VINCO SOOT-OFF

Destroys soot from coal, oil or gas burning heating equipment Easier, cleaner and quicker than brushing. Makes short work of jobs too hard for brush or scraper Cleans fire pot, flues and chimney in one simple operation Non-explosive, thoroughly safe

McDONNELL & MILLER

Manufacturers of McDONNELL *Boiler Water Level* CONTROLS

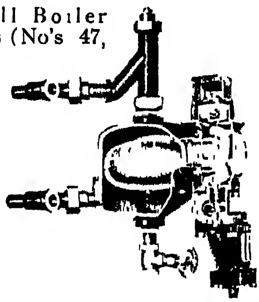
General Offices: Wrigley Building, **Chicago, Ill.**

"Doing one thing well"

PRODUCTS:

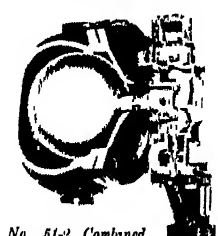
Boiler Water Feeders; Combined Boiler Water Feeders and Low Water Cut-offs; Low Water Cut-offs; Combined Low Water Cut-offs and Pressure Controls; Humidifier Water Valves.

McDonnell Boiler Water Feeders (No's 47, 51 and 53) automatically supply water to heating boilers whenever the boiler water level falls to the danger point. They not only end the low water hazard, they also improve boiler efficiency by feeding water only when it is needed and in exact proportion to the needs



No 47-2 Combined Feeder and Low Water Cut-off with "Quick-Hook-Up" for boilers below 5,000 sq ft capacity Also furnished without low water cut-off switch for hand-fired boilers (No 47) Steam pressures up to 25 lb

For automatically fired boilers they are combined with low water cut-off switches (No's 47-2, 51-2 and 53-2) In these combined units the feeder takes care of all normal operation If an emergency should arise such as extreme priming, foaming or failure of water supply, the switch stops the burner or stoker, but only until the emergency is passed--completely automatic control that "makes the boiler water level as automatic as the firing"



No 51-2 Combined Feeder and Low Water Cut-off for larger boilers Also furnished without switch for hand fired boilers (No 51) Steam pressures up to 25 lb

McDonnell Low Water Cut-offs (No's 66, 66-A, 66 B and 34-HP) are for automatically fired jobs where the owner is not willing to pay the slightly higher cost of an entirely automatic combined feeder and cut off They offer a dependable means of stopping the burner in the event of a low water condition Low water cut-offs are also furnished with a built in pressure control (No's 62-A and 62-B)

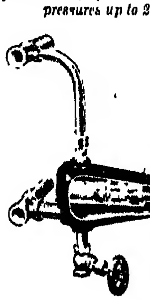
MECHANICAL ADVANTAGES

The proved dependability of McDonnell Boiler Feeders is founded on the complete isolation of the feed valve and all vital working parts from the heat of the float chamber. As a result the water at the feed valve does not reach the critical temperature at which lime and scale is precipitated This eliminates "liming" and sticking of the feed valve the cause of most feeder failures This feature the "cool feed valve" is covered by Patent No 1,931,186 All McDonnell Feeders have stainless steel valves, double sylphon (packless) construction, straight thrust valve action, multiplied closing power, large, easily cleaned, integral strainer

McDonnell Feeders, Cut offs and Feeder Cut off Combinations are available with the McDonnell "Quick Hook Up" feature which permits quick and easy installation in gauge glass tappings, automatically determines precise cut off point, and assures ideal reproduction of water level in float chamber difficult to obtain in small boilers by any other means This feature (found in No 47, No 47 2, No 66, No 66-B and No 62-B) is covered by Patent No 1,997,785

OTHER EQUIPMENT

Other McDonnell equipment includes No 87 Water Level Control for humidifier pans of warm air furnaces, high and low liquid level switches, gas pilot valves (interchangeable with low water cut off switches) and a wide range of special controls Catalog 36 contains complete descriptions, capacity chart, service recommendations, installation instructions, electrical ratings, wiring diagrams, dimensional drawings, and typical specifications.



At the left is the No 66 simple, compact, trim Low Water Cut-off with "Quick-Hook-Up" feature for boilers of any size Steam Pressures up to 15 lb

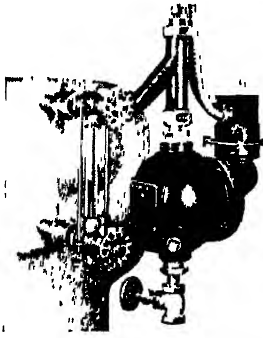
SERVICE RECOMMENDATIONS

Boiler Size	Steam Pressure	McDonnell Product to Use
FOR HAND FIRED BOILERS		
Up to 5,000 sq ft	Under 25 lb	No 47 Water Feeder
Above 5 000 sq ft	Under 25 lb	No 51 Water Feeder
Any Size	25 to 75 lb	No 53 Water Feeder
FOR AUTOMATICALLY FIRED JOBS Water Feeder— Low Water Cut-off Combinations		
Up to 5,000 sq ft	Under 25 lb	No 47-2 Feeder Cut-off Combination
Above 5 000 sq ft	Under 25 lb	No 51-2 Feeder Cut-off Combination
Any Size	25 to 75 lb	No 53-2 Feeder Cut-off Combination
Low Water Cut-offs		
Any Size	Under 15 lb	No 66 Low Water Cut-off
Any Size	15 to 25 lb	No 60-A or 60-B Low Water Cut-off
Any Size	25 to 100 lb	No 34-HP Low Water Cut-off
Low Water Cut-off— Pressure Control Combinations		
Any Size	Under 12 lb	No 62-A or 62-B Combined Cut-off and Pressure Control

TYPICAL SPECIFICATIONS FOR LOW WATER CUT-OFF

Furnish and install on each boiler, in accordance with the manufacturer's instructions, a McDonnell & Miller No (insert †) Low Water Cut-off, to be float operated and to have packless construction Control wiring to be of flexible armored cable All wiring to meet the requirements of the City Electrical Inspection Department and the *National Board of Fire Underwriters*.

†Insert at this point "No 66," "No 60-A," "No 60-B" or "No 34-HP," as indicated by service conditions in the Table above (If No 62-A or No 62-B is indicated specifications should cover the pressure control feature as outlined on Page 27 of the McDonnell Catalog 36)



No 60-B "Quick-Hook-Up" Low Water Cut-off with McDonnell wiping contact (No 2) Switch for boilers of any size Steam pressures up to 25 lb No 62-B Combined Cut-off and Pressure Control is similar in appearance and construction, maximum steam pressure 12 lb

TYPICAL SPECIFICATIONS FOR BOILER WATER FEEDER OR COMBINED FEEDER AND CUT-OFF

Furnish and install complete in every essential detail for each boiler unit Automatic Boiler Water Feeder —and Low Water Cut-off (if automatically fired)—equipment as manufactured by McDonnell & Miller, Wrigley Bldg , Chicago

Boiler Water Feeder to be (insert*) of the manufacturer's most improved design, with all working parts isolated from the steam and hot water zone by packless sylphon construction Float power to be multiplied through a leverage mechanism which contains a self-centering roller directly above the valve stem All bearing points, pivots, and levers to be outside of the steam and water zone Valve cone and seat to be stainless steel, removable and renewable A strainer to be incorporated in the design and to have a solderless basket mounted on a flange for easy removal Feeder to be installed complete with all piping, valves, fittings, and specialties, as indicated by descriptive diagram in the manufacturer's construction bulletin.

Upon completion of the installation, the contractor to place this automatic equipment in successful operation, subject to acceptance and approval of the architect's . ... engineer

When combined units No 47-2, No 51-2 or No 53-2 are specified, add the following paragraph to specifications· "Make electrical connections for low water switch to control fuel supply equipment, so that the fuel is automatically shut off by the boiler water control. Control wiring to be of flexible armored cable All wiring to meet the requirements of the City Electrical Inspection Department and the *National Board of Fire Underwriters* "

*If job is hand fired, insert· "No 47," "No 51" or "No 53," as indicated by service conditions in the Table above

*If job is automatically fired, insert "No 47-2," "No 51-2" or "No 53-2," as indicated by service conditions in the Table above

Aquatic Chemical & Metallurgical Engineers

114 East 32nd Street — LAND A-C-M-E WATER SERVICE — **New York, N. Y.**

Water Treatments for Correcting **Rust, Scale, Foaming and Related Water Troubles**

An Organization of Water Treatment Experts Announces Two New A. C. M. E. Developments to Eliminate Metal Deterioration in

1. STREET STEAM SYSTEMS

Having successfully treated corrosion in marine and stationary high pressure boiler plants, **A. C. M. E.** undertook the correction of a serious problem of corrosion existing in the street steam heating system in one of the largest buildings in the world. Nearly three years of intensified chemical and engineering research resulted in the perfection of a unique and wholly new **A. C. M. E. Service.**

This consists of an exact proportioning mechanism and chemicals (patent applied for) which protect the entire heating system by reaching *every* point in accurate proportions. These **A. C. M. E.** Treatments contain no heat retarding ingredients.

2. AIR CONDITIONING SYSTEMS

The presence of a very *large* amount of air, with its many impurities, dissolved in the water used in air conditioning plants accelerates the corrosion many fold. This results in a very real and serious problem. **A. C. M. E.** Air Conditioning Treatment protects *all* parts which come in contact with water by using regulated amounts of proper chemicals. The percentages of each and the relation of one to another are maintained by our *automatic* feeding device. These Treatments are not poisonous *regardless* of the amounts used and therefore can be handled and used safely.

Red brown particles in water indicate c o r r o s i o n. Our analysis will determine accurate conditions. Bottles sent on request. (No charge)

Results:
1. Eliminates corrosion and erosion in the *entire* heating system. Tests show **A. C. M. E.** treated metals last at least 8 to 10 times longer than untreated metals. See our Street Steam Reports.
2. Saves at *least* 7 per cent on steam by keeping traps *clean*.
3. Saves electricity. Clean traps do not permit "blowing through" thus shortening vacuum pump operation.
4. Pays for itself in longer equipment life and smoother operation of system by stopping costly renewal of pipe and reconstruction work.

A. C. M. E. Treatments for Boiler Heating Systems

Instead of depending on a single preparation for all boilers and all troubles, three specially prepared **A. C. M. E.** products are now available. See picture below.

The Action of A. C. M. E. Boiler Heating Treatments is to mix with the steam in the form of small bubbles. This has a *washing* effect on the entire system (air valves, traps, hot water heaters and auxiliaries), bringing all loosened oil, rust and dirt particles back into boiler. These particles are kept in a chemical and physical colloidal suspension until blown out, according to directions in our No. 1 L. P. Bulletin.

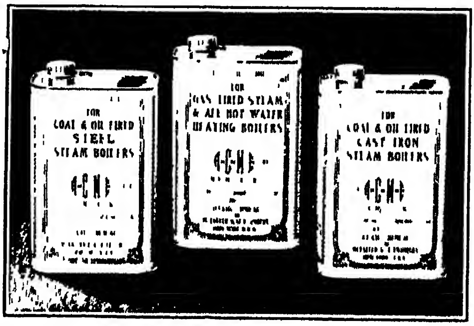
A. C. M. E. ingredients do not settle out in the boiler, and cause clogging.

RECOMMENDED AND USED BY LEADING HEATING AND VENTILATING ENGINEERS

A Clean Heating System

No Priming

Fuel Savings Up to 25%



More Steam at Lower Boiler Pressures

Quicker Steam

Stops Rust and Prevents Cracking.

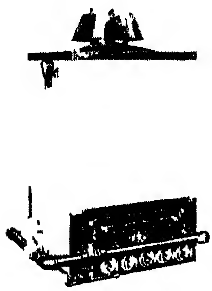
A. C. M. E. ***Users are Impressed by Positive Results***

American Gas Products Corporation

Division of American Radiator & Standard Sanitary Corporation

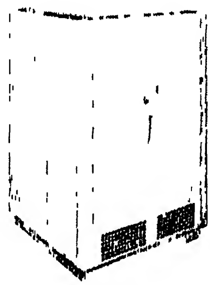
40 West 40th Street, **New York, N. Y.**

Gas Boilers for Hot Water . . . Steam and Vapor Heating . . . Automatic Hot Water Storage Heaters . . . Air Conditioners . . Gas Fired Steam Radiators



Standard Ideal

Approved by
A G A Laboratory
for Steam, Vapor
or Hot Water
Heating Systems
Ratings Below



Empire Ideal

Specifications for Standard Ideal and Empire Ideal Gas Boilers

STEAM BOILERS					WATER BOILERS				
Steam Boiler Number	A G A Steam Rating Sq Ft	Supplies Sq Ft Direct C I Radiation	No and Size of Tappings		Water Boiler Number	A G A Water Rating Sq Ft	Supplies Sq Ft Direct C I Radiation	No and Size of Tappings	
			Supply In	Return In				Supply In	Return In
0-GS-4	270	173	2 2½	1 1½	1-GA-4	210	135	2 1½	2 1½
0-GS-5	360	230	2 2½	1-1½	1-GA-5	280	180	2 1½	2 1½
0-GS-6	450	289	2 2½	1 1½	1-GA-6	350	225	2 1½	2-1½
0-GS-7	540	346	2 2½	1 1½	1-GA-7	420	270	2 1½	2 1½
0-GS-4-L	255	163	1 3	1 1½	2-GA-4	420	270	1 2½	1-2½
0-GS-5-L	340	218	1 3	1 1½	2-GA-5	560	359	1-2½	1 2½
0-GS-6-L	425	272	1 3	1 1½	2-GA-6	700	449	1 2½	1-2½
0-GS-7-L	510	327	2 3	2 1½	2-GA-7	840	539	1 2½	1-2½
0-GS-9-L	680	436	2-3	2 1½	4-GA-9	1120	725	2 2½	2 2½
0-GS-11-L	850	545	2 3	2-1½	4-GA-11	1400	915	2 2½	2 2½
1-GS-4	610	391	2 4	2 4	4-GA-13	1680	1109	2 2½	2 2½
1-GS-5	775	497	2 4	2 4	1-GW-4	980	631	2 4	2 4
1-GS-6	940	606	2 4	2 4	1-GW-5	1240	807	2 4	2 4
1-GS-7	1105	715	2 4	2 4	1-GW-6	1500	980	2 4	2 4
1-GS-8	1270	825	2 4	2 4	1-GW-7	1770	1168	2 4	2-4
1-GS-9	1435	938	2 4	2 4	1-GW-8	2030	1353	2 4	2-4
1-GS-10	1600	1050	2 4	2 4	1-GW-9	2300	1598	2 4	2 4
1-GS-11	1765	1165	2 4	2-4	1-GW-10	2560	1739	2 4	2-4
4-GS-6	2000	1332	2 6	2-5	1-GW-11	2820	1935	2-4	2 4
4-GS-7	2400	1624	2 6	2 5	4-GW-6	3200	2214	2 6	2 5
4-GS-8	2800	1922	2 6	2-5	4-GW-7	3840	2676	2 6	2 5
4-GS-9	3200	2214	2 6	2 5	4-GW-8	4480	3151	2 6	2-5
4-GS-10	3600	2505	2 6	2 5	4-GW-9	5120	3634	2-6	2-5
4-GS-11	4000	2793	2 6	2 5	4-GW-10	5760	4114	2-6	2-5
4-GS-13	4800	3387	4 6	3 5	4-GW-11	6400	4571	2 6	2 5
4-GS-15	5600	4000	4 6	3 5	4-GW-13	7680	5486	4 6	3 5
4-GS-17	6400	4570	4 6	3-5	4-GW-15	8960	6400	4 6	3 5
4-GS-19	7200	5143	4 6	3-5	4-GW-17	10240	7314	4 6	3 5
4-GS-21	8000	5714	4 6	3 5	4-GW-19	11520	8230	4-6	3 5
4-GS-22	8400	6000	6-6	4-5	4-GW-21	12800	9143	4-6	3 5
4-GS-25	9600	6857	6 6	4 5	4-GW-22	13440	9600	6-6	4 5
4-GS-28	10800	7714	6-6	4 5	4-GW-25	15360	10970	6 6	4 5
4-GS-31	12000	8570	6-6	4-5	4-GW-28	17280	12343	6-6	4-5
4-GS-33	12800	9145	8-6	5-5	4-GW-31	19200	13714	6-6	4-5
4-GS-37	14400	10285	8 6	5-5	4-GW-33	20480	14625	8-6	5-5
4-GS-41	16000	11430	8-6	5-5	4-GW-37	23040	16457	8-6	5 5
					4-GW-41	25600	18286	8-6	5 5

All Boilers except type 4-G are available in either Standard Ideal or Empire Ideal models

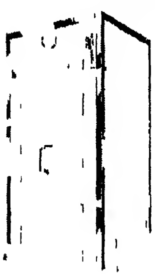
AMERICAN RADIATOR COMPANY

DIVISION OF AMERICAN RADIATOR & STANDARD SANITARY CORPORATION

40 West 40th Street, New York, N. Y.

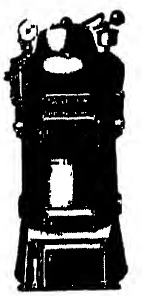
PRODUCTS FOR EVERY HEATING REQUIREMENT

AUTOMATIC COAL-FIRED BOILER No. 21



Designed especially for use with mechanical stokers Recommended by leading stoker manufacturers Can be used with either built in or external water heater The hot gases travel four times the boiler's length during which time all heat is absorbed Available in four sizes **Ratings:*** Steam 510-960 sq ft, Water 815-1535 sq ft

IDEAL ARCO ROUND BOILER

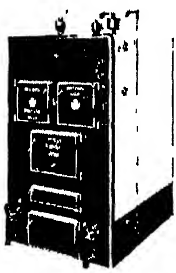


A low priced boiler Uses hard or soft coal or coke May be easily converted to automatic firing or oil or gas fuel Machine ground surfaces of doors and sections prevent waste of heat **Ratings:*** Steam 200-800 sq ft, Water 320-1280 sq ft. Available in six sizes

NEW IDEAL ARCO ROUND BOILER

A deluxe edition of the old Arco Round Attractive red enamel jacket New Arco Circulator for quicker pick-up Precision ground Arco Relief Valve Sensitive all-metal Arco Automatic Regulator Asbestocel insulation **Ratings:*** Steam 300-800 sq ft, Water 480-1280 sq ft

IDEAL REDFLASH BOILERS



Regular

Regular A practical boiler for the average home Red enamel jacket Shipped in sections so that it can be easily installed Multi-ply Asbestocel insulation Available in five different sizes Automatic Damper Regulator **Ratings:*** Steam 275-9100 sq ft, Water 440-14,560 sq ft Special Boiler for Burning soft coal

Deluxe

Deluxe Attractive red enamel jacket completely conceals the regular boiler and provides space for installation of indirect water heater without decreasing efficiency Cool external surface by complete insulation Easily accessible boiler Same ratings as regular model

ARCOLA

Hot Water Boiler for small homes up to six rooms, stores, garages or other small buildings Needs no basement. Available in three styles unjacketed, with insulated jacket, and with circulating jacket Radiators can be attached to it.

***Recommended Load** Represents attached load in square feet E D R including piping which may be placed on Boiler in accordance with accepted installation standards for economical operation

See also Pages 843 and 974

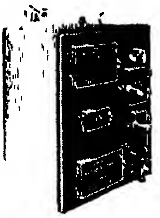
AMERICAN RADIATOR COMPANY

DIVISION OF AMERICAN RADIATOR & STANDARD SANITARY CORPORATION

40 West 40th Street, New York, N. Y.

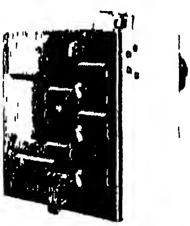
PRODUCTS FOR EVERY HEATING REQUIREMENT

MAGAZINE BOILER No. 15



Works on a gravity feed principle The fuel is stored at the top and moves down by its own weight to replace fuel consumed Requires less attention than ordinary boilers Can be fueled with many kinds of coal or coke **Ratings:*** Steam 400-1000 sq ft, Water 640-1600 sq ft

MAGAZINE BOILER No. 25

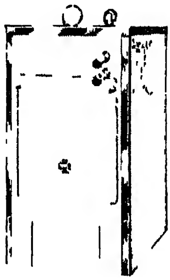


Larger size than No 15 Has special device to separate steam from water insuring dry steam, Arco Automatic Regulator for combustion control Unique by-pass directly to chimney provides quick starting **Ratings:*** Steam 750-2100 sq ft, Water 1200-3360 sq ft.

OIL BURNING BOILER No. 92

Large size automatic oil burning boiler for large homes, apartments, stores, etc **Ratings:** Steam 1360-2460 sq ft, Water 2180-3940 sq ft at transmission rate 4500 Btu/square foot Any type burner can be used with it

OIL BURNING BOILER No. 12



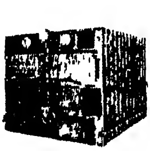
Medium size automatic oil burning boiler Also equipped with built in hot water system and low water control Asbestocel insulation of the lining to conserve the heat Can be used with rotary or gun type burner Available in four sizes **Ratings:** Steam 510-960 sq ft, Water 815-1535 sq ft at transmission rate 4500 Btu/square foot

OIL BURNING BOILER No. 11

Small size automatic oil burning boiler with built in Taco Water Heater Equipped with Detroit Boiler Protector which stops the burner if the water level drops Available in six sizes **Ratings:** Steam 355-755 sq ft, Water 570-1205 sq ft at transmission rate 4500 Btu/square feet

***Recommended Load** -Represents attached load in square feet E D R -including piping- which may be placed on Boiler in accordance with accepted installation standards for economical operation

BOILERS FOR LARGER INSTALLATIONS



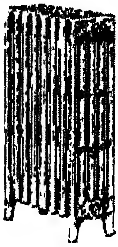
"Ideal" Water Tube Sectional Boilers—For large buildings and commercial installations Sectional constructions permit easy installation. Available in five sizes from 23 in to 79 in **Ratings:*** Steam 450-15,000 sq ft, Water 720-24,000 sq ft

AMERICAN RADIATOR COMPANY

DIVISION OF AMERICAN RADIATOR & STANDARD SANITARY CORPORATION

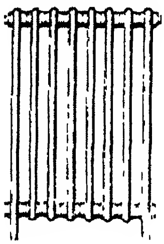
40 West 40th Street, New York, N. Y.

CAST IRON RADIATORS AND ENCLOSURES



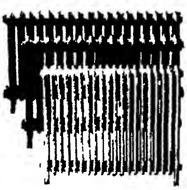
CORTO RADIATOR

A slender tube radiator which is quick-heating and responsive to varying demands of comfort Available in 7 heights and 5 widths Factory tested for 125 lb hydraulic pressure



CORTO HOSPITAL RADIATOR

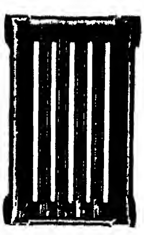
Extremely wide spacing between sections allows easy and thorough cleaning The tubes are smooth with no dust catching edges, no tie rods to collect dirt



ARCO RADIATORS

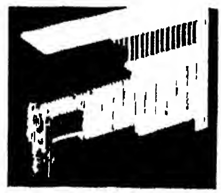
Compact radiators with an output equal to old types of larger radiators Available in two widths and three heights with 2½ or 4½ in legs or for suspension by wall brackets.

PEERLESS WALL RADIATOR



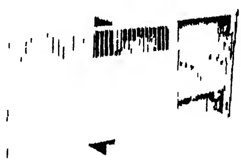
Especially desirable for commercial installations Fits into restricted spaces of practically any size or shape, under windows or between them, on walls, ceilings or in skylights Available in many sizes

ARCO RADIANT CONVECTOR



Combines the functions of radiator and convector Concealed behind an attractive enclosure Five styles of enclosures in any length up to 60 in Designed for steam, hot water or vapor systems

ARCO CONVECTOR

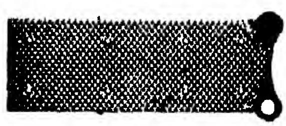


Concealed behind an attractive enclosure Long, specially designed fins heat the air thoroughly Available in four different widths 3¼-9½ in and the length may be varied by the number of sections Enclosures to fit every size of convector

VENTO CAST IRON RADIATOR

For blast heating and conditioning systems Rust proof cast iron construction makes them ideal for use with washed or humidified air Streamlined studs give maximum air wipe

PERFECTION PIN



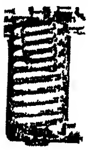
For gravity indirect heating with steam or hot water Threaded nipple or flange and bolt connections

AMERICAN RADIATOR COMPANY

DIVISION OF AMERICAN RADIATOR & STANDARD SANITARY CORPORATION

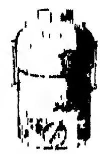
40 West 40th Street, New York, N. Y.

PRODUCTS FOR EVERY HEATING REQUIREMENT



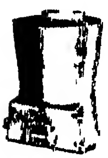
EXCELSO WATER HEATER

Provides hot water indirectly from steam or hot water boilers Available with Single, Double, Triple and Dual Coils Capacities up to 3750 gal



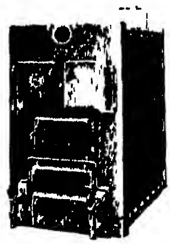
IDEAL DOME TYPE WATER HEATER

A new cast iron coal burning heater Made in seven sizes with capacities of 65-250 gal Plant tested to 315 lb pressure and guaranteed for working pressure of 125 lb



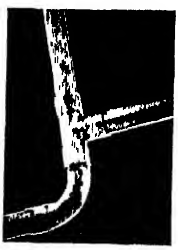
SCUTTLE-A-DAY

A coal burning water heater for small homes, stores, etc In three sizes with capacities of 75-175 gal Also **The Arco Incinerator** which burns the garbage with the same fire that heats the water Made in 75-110 gal capacities



ARCO HIGH TEST TANK HEATER

A water heater for large buildings Adapted to complete automatic control with hard or soft coal, oil or gas Hand-fired, will burn without refueling for eight hours or longer Capacities Coal 1500-3000 gal, Oil 1520-2520 gal



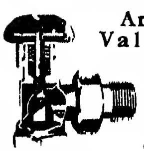
ARCO WROUGHT COPPER PIPE AND FITTINGS

Pipe and fittings expand and contract equally Ideal for heating installations Proof against leaks by vibration Corrosion resisting, non-porous Complete line ¼ to 4 in Also Arco Cast Iron Pipe 1½ to 12 in

ARCOLOY RANGE BOILERS AND TANKS

Made of the new patented metal Arcoloy with appearance and corrosion resistance of copper and the strength of mild steel Guaranteed for twenty years under normal water conditions Available in seven sizes 25-100 gal

CONTROLS AND ACCESSORIES

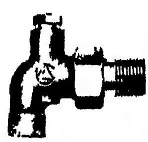


Arco Packless Valves Never leak or need repacking For steam, hot water, vapor or vacuum heating systems Five sizes from ½ to 1½ in.

Ideal Vari-port Valve - For steam and vacuum systems Easily adjustable port with wide range of venting rates permits balanced heating

Arco Hurivent Designed to vent the mains with utmost speed so that the steam can reach the radiators with a minimum of frictional loss

Arco No. 201 Thermostat — Balances radiator output against room temperature to provide an even and continuous warmth throughout the house New in conception, unequalled in performance



Arco Balancing Elbows Will balance hot water systems by restricting the flow with adjustable baffle With threaded or sweat connection in three sizes From ⅜ to ¾ in

Burnham Boiler Corporation

Manufacturers of Cast Iron and Welded Steel Boilers, Cast Iron Radiators and Heating Accessories

Irvington-on-Hudson, N. Y.

Zanesville, Ohio

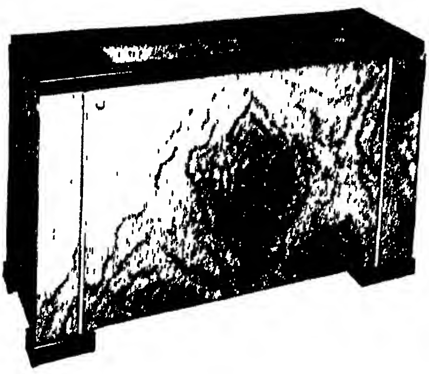
Branch Offices

Boston, Philadelphia, Chicago, Queens Village, L I, Long Island City, N Y, Baltimore, Springfield, Lancaster, Pittsburgh, Zanesville, Elizabeth, Geneva, N Y

Plants

Elizabeth, N J, Lancaster, Pa
Zanesville, Ohio, Geneva, N Y

Burnham Air Conditioning Unit and Radiator Air Conditioning Systems



The Burnham Air Conditioner is a complete unit for Winter conditioning taking the place of a radiator, and in Summer it can be operated to circulate and cleanse the air of dust and pollen

It consists of a substantial frame of cast iron and sheet steel completely enclosed within a good-looking cabinet, finished in burl Walnut which will harmonize with any type of room decoration or furniture

The cabinet also completely encloses all piping and connections to the Unit

The heating element is of a capacity up to 80 sq ft E D R and can be used on any type of Steam, Vapor or Hot Water Heating System

Has a 1/100th hp Motor, quiet in operation, and a fan having a capacity up to 350 cu ft of air per minute

A rheostat to control speed of motor is conveniently located on side of Unit

A large spun-glass type of filter, of the throw away kind, is supplied Coarse spun-glass, oil impregnated on under side and fine spun-glass on top, completely filters all air passing through the Unit

A large copper humidifier over which all heated air passes automatically controls amount of humidity taken up by the air As much as 12 gal of water per day are evaporated if the room air requires that amount Copper connecting elbow for tubing to waste line is furnished and a needle type feed water valve is supplied Float valve can be furnished when desired

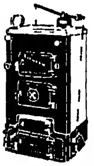
A Thermodisc and switch can be supplied, which will prevent operation of motor unless temperature of heating element is 120 deg or higher. Switch will by-pass Thermodisc and allow operation of Unit in Summer months or at any time that it is desired to operate motor This Unit requires no duct work whatever, it fits right in with a radiator heating system

Full detailed particulars given in printed matter.

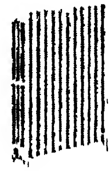
Capacity and Size

Three conditioners are sufficient for a 7 or 8 room house One each in living room, dining room and master bedroom is the usual procedure Additional Units can be added at any time by simply replacing a radiator

Size—23¾ x 39¼ x 13¾



Burnham Boiler Corporation



Irvington-on-Hudson, New York

Zanesville, Ohio

There's a Burnham for Every Purpose

Catalogs Sent on Request

1—Welded Steel Boilers—Also Three Purpose Welded Steel Boilers.

For heating, hot water supply and incineration Coal or oil Completely welded for 15 lb working pressure Multiple shaking grates Sizes for commercial or domestic uses Special folder sent on request

2—Water Tube Boilers for Steam and Hot Water Heating.

17, 21, 27, and 36 in Double shaking grates and long fire travel Rating to 9,050 sq ft for steam and 15,085 sq ft for water

3—Water Tube Boilers Jacketed in Color.

17, 21, and 27 in Steel Jacket and 4 ply air cell asbestos insulation Enameled rich red and black. Jacket goes on after all other set up work Rating to 4,225 sq ft for steam and 6,800 sq ft for water

4—Burnham Oil-Burning Boilers.

A specific-sized boiler for each specific heat job for use with any standard oil burner Round Sectional Burnhams in 6 series and 24 sizes. Square Burnhams in 5 series and 39 sizes For steam, vapor, or water

5—Big Twin Sectional Boilers.

50 in. Grate, divided for easy shaking. Twin sections, divided down the middle Ratings to 19,450 sq ft for steam, 31,800 sq ft for water.

6—Tube Type Smokeless Boilers.

For burning soft coal efficiently and without smoke Meet smoke ordinances everywhere Similar to (2) above with addition of smokeless feature.

7—Round Sectional Boilers.

This boiler made the long fire travel famous Handled easily. Very large steam dome Ratings up to 1,550 sq ft for steam, 2,560 for water

8—High Pressure Hot Water Supply Boilers.

Sectional construction Guaranteed to 120 lb working pressure Supplies up to 14,000 gal

9—Junior Hot Water Supply Boilers.

Will keep 175 to 700 gal tank always full of hot water Guaranteed to 120 lb working pressure

10—Burnham-Taco Tanks.

Combining water heater and storage tank in one unit for summer-winter use. Removable copper heating element Tanks may be galvanized, Everdur or copper.

11—Burnham Slenderized Radiators.

Cast iron radiators that occupy 40 per cent less space than ordinary type of same rating Shorter Lower Narrower 3-tube type $3\frac{1}{4}$ in wide 4-tube type $4\frac{7}{16}$ in wide 5-tube type $5\frac{11}{16}$ in wide 6-tube type $6\frac{15}{16}$ in wide Can be recessed

12—Fero Tube Radiators.

All heights—3, 4, 5, 6, and 7 tubes

13 Burnham Air Conditioning Units.

Do double duty of both heating and winter air conditioning Units placed in the room Have no basement equipment Take up no more room space than usual grille-enclosed radiator Entirely automatically controlled

14—Burnham Air and Vacuum Valves.

Full line for radiators, risers and mains.

15—Burnham Radiator Valves.

Complete line of heating accessories Including steel tanks of all kinds

16—Burnham Flexible Headers.

17—Federal Unit Heaters.

Complete line in modern designs.

Crane Co.

Manufacturers of Valves, Fittings, Fabricated Piping, Steam Specialties, Plumbing and Heating Materials

836 S. Michigan Avenue, Chicago, Ill.

Branches in All Principal Cities

Write for Catalogs and full information on any materials

COMPLETE LINE OF RESIDENTIAL AND COMMERCIAL BUILDING BOILERS AND HEATING MATERIALS

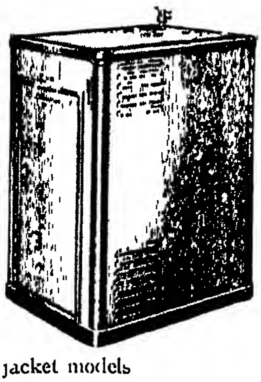
Crane boilers are made in sectional or round cast-iron styles for hot water, steam, vacuum, and vapor heating with coal, coke, gas, and oil

Crane furnishes air conditioning units and split system equipment for all types of residential installations

Crane Radiation includes direct (legless, bathroom, hospital, wall), concealed, and shielded types of high efficiency and graceful design

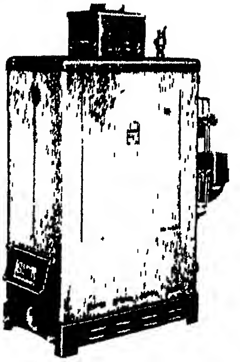
Unit heaters, humidifying radiators, and a complete assortment of valves, fittings and specialties complete the Crane heating line for residential and commercial use

Crane Deluxe D-1 Oil Burning Boiler

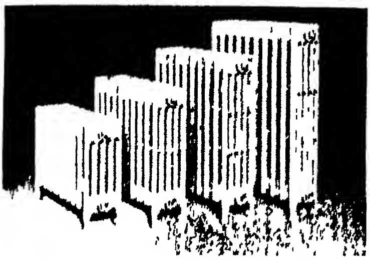


The "Sustained Heat" principle of design incorporated in this boiler assures high combustion efficiency and low standby-loss Available in regular and deluxe jacket models

Basmor Gas Fired Boiler



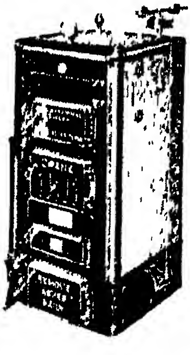
With the famous Butterfly Bunsen-type burner, this boiler develops high efficiency with negligible stand-by-loss All types of gas Available in both regular and deluxe jacket models



Crane Directed Radiation

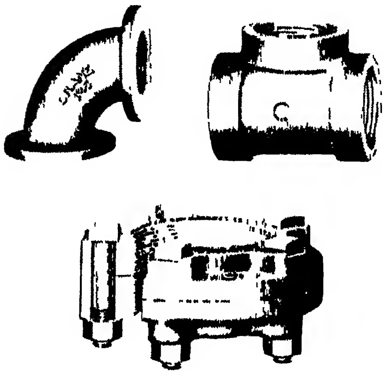
Invisible low-cost shields throw heat outward into room, equalizing temperatures, protecting walls and drapes.

Crane Coal Fired Boilers

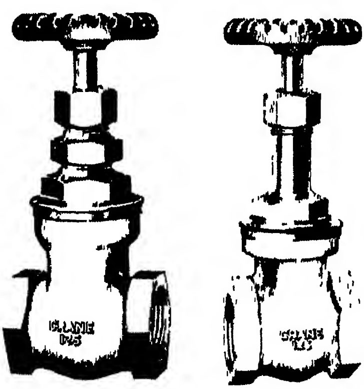


50 per cent more "ceiling" surface, controlled water travel, and a score of operating refinements place this boiler on a plane of highest efficiency

EVERY TYPE OF VALVE AND FITTING FOR HEATING INSTALLATIONS

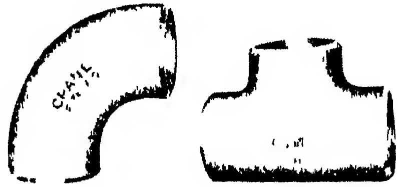


Crane Standard Cast Iron Fittings – Of dense, uniform metal, with faces accurately aligned and at exact right angle to flow Smooth-cut threads, smoothly faced flanges

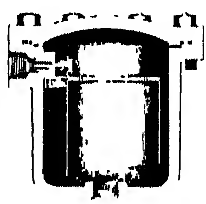


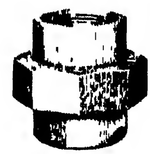
Crane Wedge Gate Valve (Left)—A brass valve with non-rising stem, stuffing box gland, and disc guides that prevent wear of seating surfaces Can be packed while wide open under pressure For steam working pressures up to 125 lb

Crane Double Disc Gate Valve (Right)—A brass valve with rising stem, parallel seats, and wedging device assuring equal bearing on all parts of seat Can be packed while wide open under pressure For steam working pressures up to 125 lb



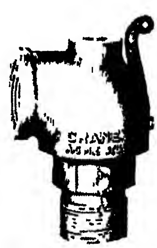
Crane Welding Fittings- Uniform in cross-section and radius, with ends accurately beveled to fit pipe or tubing Tangents facilitate welding in line





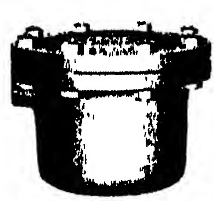
Crane Steam Trap-- A cast-iron trap of high capacity Simple working parts, removable without touching line Expels condensate in instantaneous action

Crane No. 512 Gem Union—A ground-joint union with brass to iron seat No gasket required Seat ring cannot loosen Taper threads. Easy to install 150 lb steam



Crane No. 1168 L. P. Pop-Safety Valve -A side-outlet valve of malleable iron, with brass trimmings, for steam heating boilers. Sturdy rigid base and ample clearance for tools Regularly set at 5, 10 or 15 lb, available at 3 to 25 lb

Crane No. 984 Air Vent—An automatic vent to remove air from water mains, hot water heating systems, tanks Operation similar to the No. 981 trap Simple, low in price



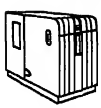
National Radiator Corporation

General Offices **Johnstown, Pa.**

Sales and Service Through These Branch Offices and Warehouses

BALTIMORE BOSTON BUFFALO CHICAGO CINCINNATI
CLEVELAND LEBANON MILWAUKEE NEW YORK
PHILADELPHIA PITTSBURGH RICHMOND WASHINGTON

National Cast Iron Oil Heating Boilers

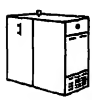


Available in either standard jacket, or extended--which provides rear cabinet for burner and controls. Built-in Taco heater, large steam space, water capacity, heating surface, and combustion chamber volume. Red, black and aluminum finish.

	Series [illegible]					
Boiler Size Sections	5	6	7	8	9	10
Bonded Rating Steam	[illegible]	400	[illegible]	[illegible]	[illegible]	720
Bonded Rating Water	[illegible]	[illegible]	[illegible]	[illegible]	1030	[illegible]
Heating Surface Sq Ft	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]

[illegible]

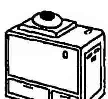
National Premier Vertical Steel Boilers



For either oil or stoker firing. Extended jacket providing front cabinet for burner, or standard. Rated under S.H.B.I. Code. Equipped with built-in Taco copper coils for domestic hot water. Finished in French Grey. Quality throughout.

Boiler Size	[illegible]	[illegible]
SHBI Ratings Steam	[illegible]	680
SHBI Ratings Water	[illegible]	1090
Heating Surface Sq Ft	[illegible]	40

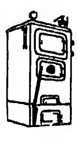
National Gas Boilers



Beautiful in appearance, splendidly engineered. French grey enamel finish, chromium base and dome.

	Series No. [illegible]							Series No. [illegible]										
Boiler Size Sections	3	4	5	6	7	8	9	4	5	6	[illegible]	8	9	10	11	12	13
Certified A.G.A. Rating Steam	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
Certified A.G.A. Rating Water	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]

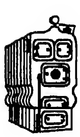
National Bonded Jacketed Square Boilers



Red baked-enamel jacket, black base and doors. Made in a variety of sizes to meet needs of every modern home. Burns any fuel.

	Series No. [illegible]						Series No. [illegible]				Series No. [illegible]				
Boiler Size Sections	4	5	6	7	8	9	[illegible]	6	7	8	[illegible]	6	7	8	9
Bonded Ratings Steam	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	1400
Bonded Ratings Water	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
Grate Area Sq Ft	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]

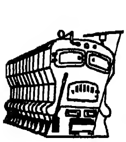
National Bonded Novus Sectional Boilers



An established favorite. Series 48 has divided sections, for easy portability and erection. Long fire travel, burns any fuel.

	Series No. [illegible]				Series No. [illegible]					Series No. 48						
Boiler Size Sections	5	6	7	8	[illegible]	[illegible]	[illegible]	8	9	6	7	8	[illegible]	10	11	[illegible]
Bonded Rating Steam	480	600	[illegible]	840	700	875	[illegible]	[illegible]	1400	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
Bonded Rating Water	790	990	1190	1390	1150	1440	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
Grate Area Sq Ft	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]

National Bonded Imperial Sectional Boilers



Originally marketed under the name of the Utica Imperial Sectional Boiler. Arched ribs give large amount of prime heating surface. Highly efficient with any type of fuel.

	Series No. 32					Series No. 42										
Boiler Size Sections	5	6	7	8	9	6	7	8	9	10	11	12	13	14	15	16
Bonded Rating Steam	1200	1500	1800	[illegible]	2400	2200	2700	[illegible]	[illegible]	[illegible]	4700	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
Bonded Rating Water	1980	2475	2970	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	8400	[illegible]	[illegible]	[illegible]	[illegible]
Grate Area Sq Ft	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]

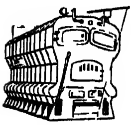
(*Maximum [illegible])

See also Page 872

National Radiator Corporation

HEATING EQUIPMENT

National Bonded Super-Smokeless Boilers

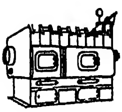


Have established an enviable reputation in the smokeless boiler field Efficiently burn low grades of fuel Noted for long, rotative fire travel

	Series No 24					Series No 13					
Boiler Size Sections	5	6	7	8	9	5	6	7	8	9	10
Bonded Rating Steam	750	1000	1250	1500	1750	1300	1650	2000	2350	2700	3050
Bonded Rating Water	1250	1650	2050	2450	2850	2150	2725	3300	3875	4450	5025
Grate Area Sq Ft	5.00	6.25	7.50	8.75	10.00	7.32	9.10	10.87	12.65	14.42	16.20

	Series No 10																		
Boiler Size Sections	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24
Bonded Rating Steam	2300	2800	3300	3800	4300	4800	5300	5800	6300	6800	7300	7800	8300	8700	9100	9500	9900	10300	10700
Bonded Rating Water	3795	4620	5445	6270	7095	7920	8745	9570	10395	11220	12045	12870	13695	14355	15015	15675	16335	16995	17655
Grate Area Sq Ft	12.03	14.38	16.71	19.08	21.43	21.77	21.43	21.43	21.77	21.77	23.77	23.77	23.77	23.77	23.77	23.77	23.77	23.77	23.77

National Bonded Low Water Line Boilers

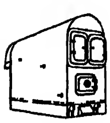


Will solve the head-room problem, in tide-water country, in theatre work, and everywhere where head-room is limited Also available in smokeless type

	Series No 80								Series No 40												
Boiler Size Sections	5	6	7	8	9	10	11	12	6	7	8	9	10	11	12	13	14	15	16	17	18
Bonded Rating Steam	850	1100	1350	1600	1850	2100	2350	2600	1800	2200	2600	3000	3400	3800	4200	4600	5000	5400	5800	6200	6600
Bonded Rating Water	1400	1815	2230	2645	3060	3475	3890	4305	2970	3630	4290	4950	5610	6270	6930	7590	8250	8910	9570	10230	10890
Grate Area Sq Ft	5.81	7.29	8.75	10.21	11.67	13.13	14.59	16.05	9.79	11.73	13.68	15.62	17.57	19.51	21.45	26.40	25.40	25.34	27.31	31.17	33.12

NOTE—A National Boiler Bond, issued and backed by a great independent surety company, guarantees material, workmanship, and the published Bonded Ratings of all National Bonded Boilers

National Premier Steel Boilers

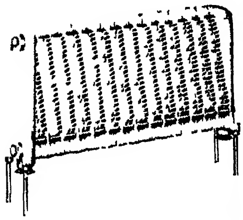


Commercial series 19 sizes, four types—Oil, or gas, stoker, direct hand fired, smokeless hand fired Residential series 11 sizes, three types Oil or gas, stoker, and direct draft hand fired Welded construction, A S M E Code; 15 lbs W P

Number of Boiler	317	318	374	375	376	431	435	436	491	495	551	554	613	673	674	753	793	853	854
Steam Ratings S H B I Type DB and SB Sq Ft	1800	2200	2600	3000	3500	4000	4500	5000	6000	7000	8500	10000	12500	15000	17500	20000	25000	30000	35000
Type OB and MB Sq Ft	2190	2680	3160	3650	4250	4860	5470	6080	7290	8500	10330	12150	15180	18220	21250	24290	30360	36430	42500
Water Ratings S H B I Type DB and SB Sq Ft	2880	3520	4160	4800	5600	6400	7200	8000	9600	11200	13600	16000	20000	24000	28000	32000	40000	48000	56000
Type OB and MB Sq Ft	3500	4280	5050	5840	6800	7770	8750	9720	11660	13600	16520	19440	24280	29150	34000	38860	48570	58280	68000
Heating Surface All Types	129	158	186	215	250	286	322	358	429	500	608	715	893	1072	1250	1429	1786	2143	2500
Grate Area Type DB and SB	7.9	8.9	9.7	10.5	11.4	12.2	13.4	14.5	16.4	18.1	20.5	22.5	25.6	28.4	30.9	33.2	37.4	41.2	44.7
Furnace Volume Type OB and MB	17.6	21.4	25.3	29.2	34.0	38.9	43.8	48.7	58.4	68.0	82.7	97.2	121.5	145.8	170.0	194.3	242.9	291.5	340.0
Width Overall All Types	32½	32½	39¼	39¼	39¼	45¼	45¼	45¼	51¼	51¼	57⅛	57⅛	63⅛	69⅛	69	75⅛	81⅛	81¼	87¼
Boiler Length Type DB SB and OB	5 3⅛	6 5⅛	5 5⅛	6 1⅛	7 0⅛	6 11⅛	7 8⅛	8 5⅛	7 0⅛	8 0⅛	8 4⅛	9 8⅛	9 3⅛	10 2⅛	11 8⅛	11 11⅛	13 7⅛	12 6⅛	14 4⅛
Type MB	5 3⅛	6 5⅛	5 5⅛	6 1⅛	7 0⅛	6 11⅛	7 8⅛	8 5⅛	7 0⅛	8 0⅛	8 3⅛	9 5⅛	9 0⅛	9 11⅛	11 3⅛	11 7⅛	13 1⅛	12 0⅛	13 8⅛
Boiler Length Overall Type DB SB and OB	6 6⅛	7 6⅛	6 5⅛	7 1⅛	8 0⅛	7 11⅛	8 8⅛	9 5⅛	8 2⅛	9 2⅛	9 6⅛	10 10⅛	10 6⅛	11 6⅛	11 0⅛	11 7⅛	15 3⅛	14 2⅛	16 0⅛
Type MB	6 6⅛	7 6⅛	6 5⅛	7 1⅛	8 0⅛	7 11⅛	8 8⅛	9 5⅛	8 2⅛	9 2⅛	9 5⅛	10 7⅛	10 3⅛	11 3⅛	12 7⅛	13 3⅛	14 9⅛	13 8⅛	15 4⅛
Boiler Height Overall Type DB SB and OB	5 7½	5 7½	6 7	6 7	6 7	7 0	7 0	7 0	8 2	8 2	8 6	8 6	9 4	9 6	9 6	9 10	10 2	11 5	11 5
Type MB	5 7½	5 7½	6 8	6 10	6 10½	7 2	7 3	7 3	8 6	8 8	9 0	9 3	10 4	10 7	10 10	11 11	12 0	13 11	14 6
Steam Outlet Size	6	6	6	6	6	6	6	6	8	8	8	8	8	8	8	8	10	10	10
Return Inlet Size	3	3	3	3	3	3	3	3	4	4	4	4	4	4	4	4	6	6	6
Safety Valve Size	1½	1½	2	2	2	2	2	2½	2½	2½	3	3	2 2	2 2½	2 2½	2 2½	2 2½	2 3	2 3
Water Line Height Type DB SB and OB	4 10	4 10	5 8	5 8½	5 8½	5 8½	5 11	5 11	6 9	6 9	7 1	7 1	7 11	8 0	8 0	8 3	8 3	9 7	9 7
Type MB	4 10	4 10	5 10	5 11½	6 0	6 1½	6 2	6 2½	7 1	7 3	7 7	7 10	8 11	9 1	9 4	9 9	10 1	12 2	12 9

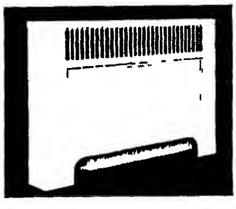
National Radiator Corporation

HEATING EQUIPMENT



National Aero Cast Iron Convectors

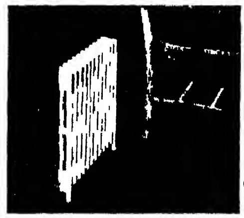
Proven dependability and permanence Made in eight types for recesses 3¾ in, 4⅜ in, 5⅞ in, 8 in and 9½ in deep Any length in multiples of 2 in Fins cast integral with tubes Deliver large volume of moderately warmed air



Type "RE" with Moulding

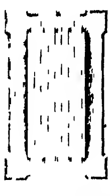
National Aero Enclosures

Enclosures with liners and removable fronts made for free standing, partial, or complete, concealment Plaster front type also available Front panels only, without liners, are made for complete concealment of convectors



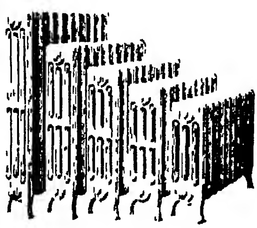
National Art Radiators

A small radiator offering truly artistic design Not an adaptation from older types Made in three widths 3, 4 and 5 tube, in heights from 19 in to 26 in Assembled in multiples of 1½ in Can be furnished legless May be used exposed or recessed



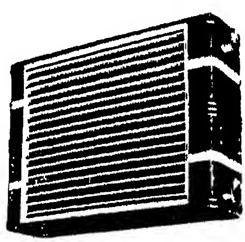
National Aero Wall Radiators

Used in factories, storehouses, garages, halls, etc Offers maximum heat in restricted space—leaves all floor space clear Can be assembled to fit available wall areas Made in 2 sizes– 7 and 9 sq ft —in each of two types Bars run short way of section in 7-A and 9-A, long way in 7-B and 9-B Units may be grouped as required



National Aero Tube Radiators

Made in three, four (illustrated), five, six and seven-tube styles, in varying heights Assembled with cast-iron push-nipples – proven protection against leaks - in any desired length Aero Radiators can be combined in any proportion with Aero Convectors



National Unit Heaters

Furnished in 15 different sizes for ceiling, wall or floor mounting Strong copper tubes fused to steel headers, copper fins imbedded in tubing, tested for 200 lb working pressure May be operated as recirculating fans for summer cooling Black and aluminum finish

Ask the nearest branch for complete technical information on National Radiators and Unit Heaters

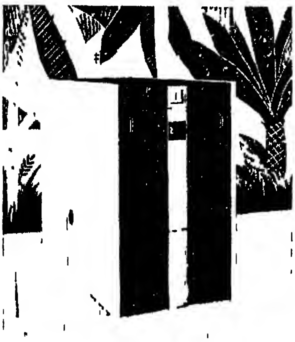
Weil-McLain Company

Manufacturing Division **Michigan City, Ind. and Erie, Pa.**

General Offices **641 W. Lake Street, Chicago**

NEW YORK OFFICES **501 Fifth Avenue**

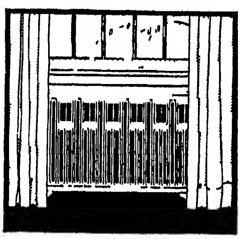
Prompt Weil-McLain Boiler and Radiator service is made conveniently available through local stocks carried by Weil-McLain Distributors in most of the important distributing centers



Weil-McLain BOILERS

Weil-McLain Boilers are made in various types and in a wide range of sizes to satisfy every heating need and demand The conventional line includes the Round, Jacketed, Square, Self-Feed and Smokeless types Illustrated at left is the new No 78 series De luxe Model Weil-McLain Boiler built especially to meet the requirements of automatic heating with oil, gas or stoker Equipped with built-in year around domestic hot water heater Thoroughly insulated Supplied with standard or extended jacket All openings for attaching automatic controls conveniently grouped at rear of boiler

Concealed Type



Equipped with or without Facilities for Humidifying

Raydiant

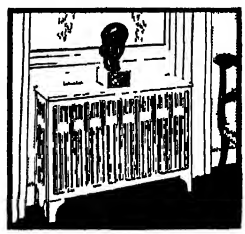
WITH HEAT RADIATING "LIVE" FRONT

Weil-McLain Raydiant Radiators supply a scientific blend of sun-like radiant warmth and convected heat Made entirely of cast iron and with full "live" front, they retain heat longer to increase the comfort of automatic (on-and-off) heating They make balanced heating easy in same installation with conventional radiators

CAMEO

Weil-McLain Cameo Tube Radiators are made in Senior, Junior and Wall Type

Cabinet Type



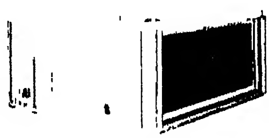
Equipped with or without Facilities for Humidifying

Weil-McLain CONDITIONERS

Weil-McLain Conditioners are designed to independently supplement radiator heat with these functions of air conditioning humidification, air cleaning, air circulation, ventilation and air tempering

The Weil-McLain Conditioner is an assembly of three distinct units, each readily removable These units are a blower assembly, a tempering and humidifying assembly and filters All are housed in a sturdily framed jacketed enclosure

W-36-1 Conditioner designed for smaller homes, W-36-2 for larger installations



W-36-1 Conditioner

W-36-2 Conditioner

Spencer Heater Company

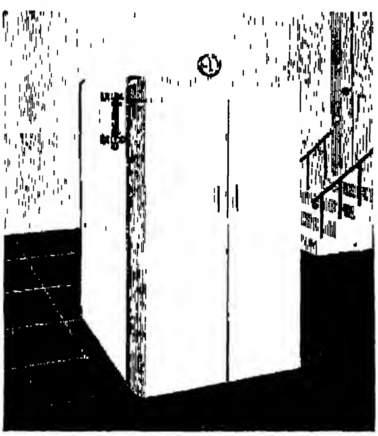
Williamsport, Pa.

Branch Offices and Representatives

New York, N Y	Boston, Mass	Albany, N Y	Syracuse, N Y
Philadelphia, Pa	Springfield, Mass	Binghamton, N Y	Allentown, Pa
Baltimore, Md	Washington, D C	Buffalo, N Y	Scranton, Pa

Spencer Automatic Magazine Feed Heaters are made in cast-iron sectional types for steam, vapor and hot water heating Also in steel tubular types for larger buildings There are sizes and capacities to provide economical and convenient heat—safe and sure—for every type of building

Spencer is the original magazine feed heater with a record of more than 38 years' successful operation They are sold and installed by all good heating contractors



Why Spencer Heaters perform so satisfactorily can best be explained by a brief inspection of their design and construction The Spencer principle—as illustrated in the cross sectional view—is simple

Once a day fuel is put into the magazine (A) It fills the sloping grate to the level of the magazine mouth (B) The fire bed always stays at (C) for as fast as fuel burns to ash (D) it shrinks and settles on the sloping grate (E) and more fuel feeds down automatically over the top of the fire bed Fuel feed is by gravity alone, in just the right amount to keep the fire always burning at its most efficient combustion point

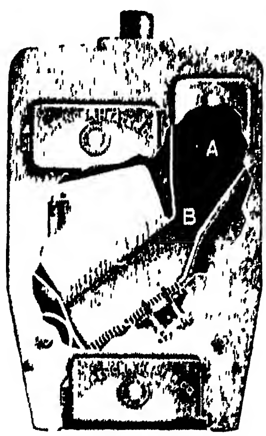
This explains why a Spencer Heater always gives the same uniform satisfying heat, and burns low cost coal economically These exclusive Spencer advantages are available in all types, J and L series, cast-iron sectional, burning low cost No 1 Buckwheat, the C-N series for larger fuels, or the steel magazine feed heaters, also for No 1 Buckwheat.

Coal - Coke - Gas - Oil Spencer J and L series cast-iron and steel tubular heaters are primarily designed to burn low cost No 1 Buckwheat anthracite and the C-N series Nut or Pea anthracite or coke

If at any time a home owner desires to burn more expensive fuels - oil or gas - a Spencer Heater will show an efficiency that is unsurpassed

Thermostats A Minneapolis Acratherm thermostat and electric damper motor are furnished as optional equipment

Jacketed Covering Attractive metallic jackets, as illustrated are available for Spencer Cast-Iron Heaters



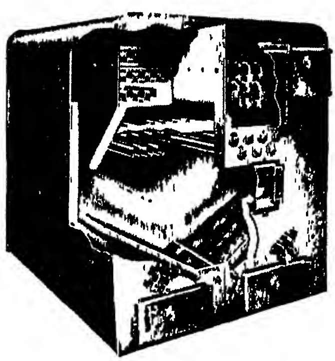
*The Spencer Heater Company guarantees that when installation is made in accordance with accepted standards, the heater will carry the number of feet of direct cast-iron column radiation listed This allows ample provision for heat loss in covered mains, risers and returns, and for peak loads

SPENCER STEEL TUBULAR MAGAZINE FEED BOILERS

For larger buildings we recommend Spencer Steel Tubular Boilers, also of the magazine feed type, burning low cost No 1 Buckwheat anthracite or coke

In the cross-section diagram illustrated here, part of the fire bed is cut away to show the sloping grates and the two magazines filled with fresh coal, ready to feed down automatically of its own weight to the fire These boilers are built in two vertical sections for ease in handling and installation—a great advantage on remodeling or replacement jobs, eliminating necessity of costly tearing out of walls, etc

Combination water tube and fire tube construction Built to *A S M E* standards

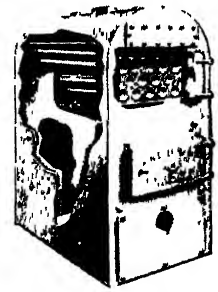


Sizes and Guaranteed Capacities

No	STEAM *Guaranteed Direct Cast-Iron Column Radiation Loads, Sq Ft	†E D R Rating
M6-6	2300	2875
M6-7	2600	3250
M6-8	2900	3625
M6-9	3200	4000
M6-10	3500	4375
M7-6	4000	5000
M7-7	4700	5875
M7-8	5400	6750
M7-9	6100	7625
M7-10	6800	8500
M8-6	8000	10000
M8-7	9750	12185
M8-8	11500	14370
M8-9	13250	16555
M8-10	15000	18740

†E D R ratings represent the attached net radiation load plus piping losses expressed in square feet of equivalent direct radiation (240 B t u steam, 150 B t u water) which may be placed on heater in accordance with accepted standards for economical operation

SPENCER STEEL BOILERS, FOR OIL, STOKER OR GAS



For more than 38 years Spencer has been building efficient, economical and dependable coal burning boilers With this background of experience, Spencer engineers have developed the Spencer Steel Boiler, for oil, gas or stoker

The high sustained efficiency of these boilers means adequate heat for low fuel cost Design is of the three-pass type Combustion chamber is amply large Built of open hearth steel boiler plate and electrunite tubes—a quality product for dependable service

Furnished with Taco indirect domestic water heating coils either of the storage tank or instantaneous type

Sizes and Guaranteed Capacities

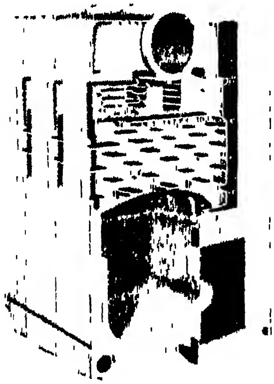
CAST-IRON SECTIONAL TYPE

	No	STEAM *Guaranteed Direct Cast Iron Column Radiation Loads, Sq Ft	STEAM †E D R Rating	WATER *Guaranteed Direct Cast Iron Column Radiation Loads, Sq Ft	WATER †E D R Rating
BUCKWHEAT BURNING	J-3	175	220	290	360
	J-4	265	330	440	550
	J-5	355	440	590	740
	L-105	390	490	645	800
	L-106	510	640	845	1050
	L-107	630	790	1045	1300
	L-205	550	690	910	1140
	L-206	725	910	1200	1500
	L-207	900	1130	1490	1860
	L-208	1075	1350	1780	2220
	L-209	1250	1570	2070	2580
	L-305	1150	1430	1900	2375
	L-306	1500	1870	2475	3095
	L-307	1850	2310	3050	3815
	L-308	2200	2750	3625	4535
	L-309	2550	3190	4200	5255
	L-310	2900	3630	4775	5975
	L-311	3250	4070	5350	6695
CHESTNUT BURNING	CN-502	280	350	460	575
	CN-502½	360	450	590	735
	CN-503	440	550	725	905
	CN-5K3	520	650	860	1070
	CN-504	600	750	995	1235
	CN-5K4	680	850	1130	1400
	CN-505	760	950	1265	1565
	CN-5K5	840	1050	1400	1730
	CN-506	920	1150	1535	1895

UNITED STATES RADIATOR CORPORATION

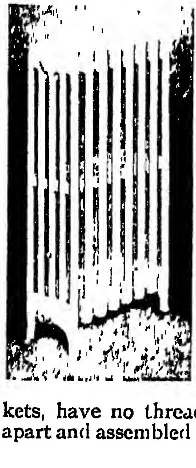
General Offices Detroit, Michigan • Branches and Sales Offices in Principal Cities

Detroit, Michigan



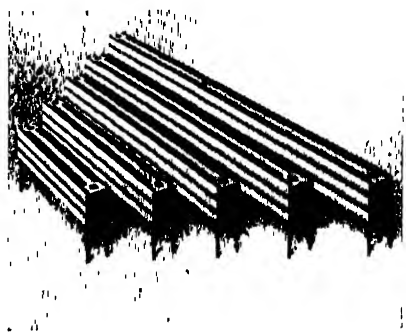
Illustrated above is the US-25 Boiler for horizontally fired oil burners The high efficiencies of this boiler are due to a method of heat absorption through new type extended rib heating surfaces The boiler is of the wet base construction which absorbs heat that would ordinarily be lost through the floor Year 'round domestic hot water is provided Construction is of cast iron A heavy jacket and a thick blanket of rock wool completely insulate the boiler

CAPITOL RADIATORS



Capitol Radiators give warmth where you want it and when you want it Homes heated with Capitol Radiators are comfortable, clean and healthful

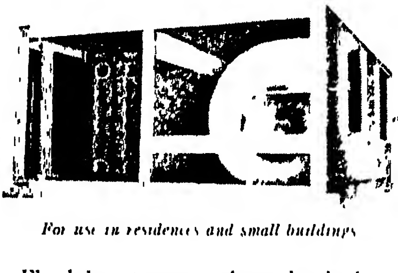
Capitol Radiators have the added advantage of assembly with extra heavy malleable iron push nipples, machined with hair-breadth precision, to form a tight, iron-to-iron joint They need no gaskets, have no threads to rust, are taken apart and assembled with the greatest ease



CAPITOL FINCAST CONVECTORS

Made entirely of cast iron
Made without joints
Cast in one piece
Many lengths and widths
Tappings, top, bottom and ends
Complete choice of enclosures

THE CAPITOL AIR CONDITIONER



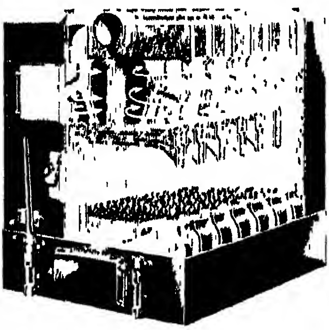
For use in residences and small buildings

Flexibility to meet the individual requirements of the building is the keynote of design in the Air Conditioner The owner may have a completely air conditioned building, he may have a partially air conditioned building with modern steam or hot water heating in less frequented rooms He may have a system that conditions the air in winter with a certain amount of cooling effect during the summer, due to circulation of air by a big quiet fan

UNITED STATES RADIATOR CORPORATION

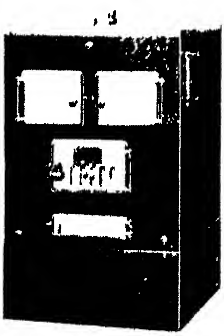
General Offices Detroit, Michigan • Branches and Sales Offices in Principal Cities

Detroit, Michigan



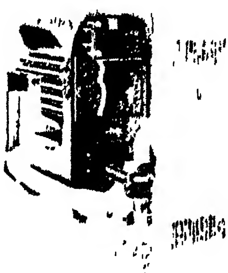
CAPITOL SQUARE BOILERS FOR ALL FUELS

Illustrated above is a Capitol Red Top Series "C" Boiler. These sectional, cast iron, all fuel boilers can be furnished either jacketed or unjacketed. Sections are connected with precision made slip nipples in accurately machined ports and the individual sections are ground to permit an iron to iron gas tight assembly. The large, smooth flueways are designed without sharp turns reducing friction permitting unrestricted gas travel. For easy cleaning, all flueways are fully accessible through the large front clean out doors. Controlled internal water circulation and large nipple ports insure complete separation of water and steam. All Capitol Red Top Boilers are heavily insulated against heat loss by a thick blanket of rock wool, reinforced with wire mesh to prevent slipping or sagging. To provide extra setting height or desired additional furnace volume when used with automatic stokers, Red Top Boilers can be furnished with extra high steel bases (illustrated below) which eliminate the necessity for pitting or building a brick or concrete base.



CAPITOL RED CAP BOILERS

Correctly designed flue passages force the hot gases to circulate through every section. The deep firepot provides the extra space needed for a long firing period and good combustion. The firepot walls assure a smooth, clean surface for heat absorption. A secondary air vent in the fire door carries the air down between the door and the hot baffle plate where it is distributed over the surface of the fire. Here the hot oxygen mixes with the volatile gases aiding complete and smokeless combustion.



PACIFIC STEEL BOILERS

This Pacific Oil Burning Steel Boiler is of the Scotch Marine type with its combustion chamber running the full length of the boiler. Thus the products of combustion must travel three times the entire length of the boiler. The combustion chamber is amply large to permit complete combustion before gases enter the tubes. It is shaped and proportioned for horizontal oil firing. The firing zone is completely surrounded by water backed surfaces contributing to greater efficiency. The Pacific is built with three inch tubes, fitted with flue gas deflectors to create the necessary turbulence so that maximum extraction of heat is effected.

THE BABCOCK & WILCOX COMPANY

85 Liberty Street, New York, N. Y.

Manufacturers of

Water-Tube Boilers | **Chain-Grate Stokers**

Oil Burners | **Seamless Steel Tubing and Pipe**

Branch Offices and Representatives in all Principal Cities

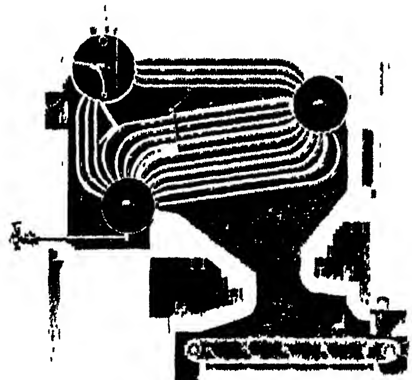
Type H Stirling Boiler

The Babcock & Wilcox Type H Stirling Boiler is a highly efficient unit built for moderate pressures at moderate prices and is designed to occupy minimum floor space and head room for the heating surface required

This boiler is built in four classes and 36 sizes ranging from 691 to 4980 sq ft of heating surface, and can be designed for operation with any fuel and every method of firing

The moderate price is due only to the simplicity of design, efficient production methods and superior shop equipment

Class	Heating Surface	Depth of Setting Ft, In	Width of Setting: Single Boiler, Ft, In	Width of Setting: Two Boilers in Battery Ft, In	Floor to Center of Mud Drum, Ft In	Floor to Face of Steam Outlet Ft In	Floor to Top of Boiler, Ft In	Size of Steam Outlet, In
	691	15-2	6-0	11-0	5 2	14-5½	13 3 1/8	5
	921	"	7-0	13-0	"	"	"	"
	1152	"	8-0	15 0	"	"	"	"
H-1	1382	"	9-0	17 0	"	"	"	"
	1612	"	10-0	19-0	"	"	"	"
	1843	"	11-0	21-0	"	"	"	"
	2073	"	12-0	23 0	"	"	"	"
	2304	"	13 0	25-0	"	"	"	"
	877	17-8	6-0	11-0	4 9¾	14 5½	13 3 1/8	5
	1169	"	7 0	13-0	"	"	"	"
	1462	"	8-0	15 0	"	"	"	"
H-2	1754	"	9-0	17-0	"	"	"	"
	2046	"	10 0	19 0	"	"	"	"
	2339	"	11 0	21-0	"	"	"	"
	2631	"	12 0	23-0	"	"	"	"
	2924	"	13 0	25-0	"	"	"	"
	1063	20-2	6-0	11-0	4 5½	14 5½	13 3 1/8	5
	1417	"	7-0	13 0	"	"	"	"
	1772	"	8 0	15 0	"	"	"	"
	2126	"	9 0	17 0	"	"	"	"
H-3	2480	"	10-0	19 0	"	"	"	"
	2835	"	11 0	21-0	"	"	"	"
	3189	"	12 0	23 0	"	"	"	"
	3544	"	13 0	25-0	"	"	"	"
	3898	"	14 0	27-0	"	"	"	6
	4252	"	15 0	29 0	"	"	"	6
	1245	22-8	6-0	11-0	4-1¼	14-11⅛	13 3 1/8	5
	1660	"	7-0	13-0	"	"	"	"
	2075	"	8-0	15-0	"	"	"	"
	2490	"	9-0	17-0	"	"	"	"
H-4	2905	"	10-0	19 0	"	"	"	"
	3320	"	11-0	21-0	"	"	"	"
	3735	"	12-0	23 0	"	"	"	"
	4150	"	13 0	25-0	"	"	"	"
	4565	"	14 0	27 0	"	"	"	6
	4980	"	15-0	29-0	"	"	"	6



Type H Stirling Boiler with Babcock & Wilcox Chain-Grate Stoker

The advantages of the Babcock & Wilcox Type H Stirling Boiler may be summarized as follows

Unusual steaming capacity for the floor space and head room required

Boilers may be set singly or in battery

Setting heights can be varied to suit any condition of firing

The choice of three locations for gas exit reduces cost of flues and breeching

Distribution baffles make effective all of the heating surface

Tube renewal is facilitated by correct tube spacing, and a tube removal door

Soot blowers can be readily installed to simplify thorough cleaning of all tubes

A superheater can be furnished without any change in the standard design or construction

The boiler is supported by a structural-steel framework entirely independent of the brickwork

Ample provision is made for free movement of parts due to expansion and contraction.

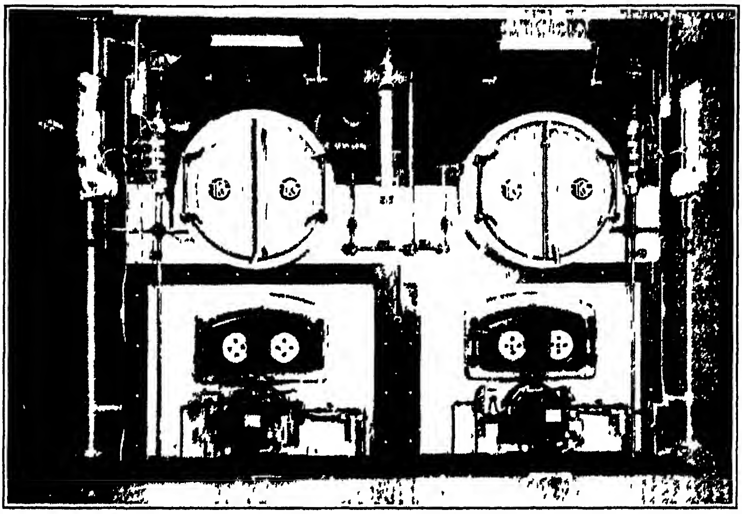
A complete table of sizes and dimensions, together with pertinent installation data, is contained in a new bulletin which will be sent upon request Simply ask for Bulletin G-8-B

The Bigelow Company

ESTABLISHED 1860

Boston, Mass | Main Office and Works | Philadelphia, Pa
Detroit, Mich | **New Haven, Connecticut** | Syracuse, N Y
New York, N Y | | Toledo, Ohio

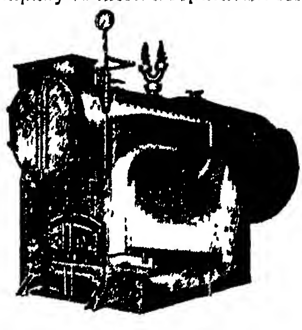
Manufacturers of Bigelow-Hornsby Water Tube Boilers, Bigelow Low Head Water Tube Boilers, Bigelow Three Drum Vertical Water Tube Boilers, Bigelow Horizontal Return Tubular Boilers, Bigelow Electric Steam Generators, Bigelow Scotch Type Boilers, Bigelow Two-Pass Boilers, Bigelow Manning Boilers, Bigelow Upright Boilers.

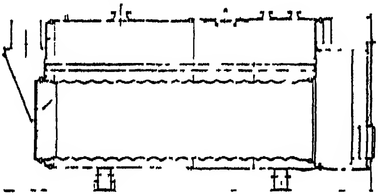


Two 60 hp Bigelow Two-Pass boilers used for heating the new Home Office Building of The Hartford Steam Boiler Inspection and Insurance Co, the largest insurer of steam boilers in the United States

Since 1860 it has been the endeavor of The Bigelow Company to merit a reputation for constructing boilers of a high standard of design, workmanship and material We are proud of the confidence shown in us by the use of Bigelow boilers in this building

The Bigelow Two-Pass Boiler is designed to meet heating and power requirements, especially where space limitations prevail It contains the recognized features of the well-known H R T boiler, but due to the small amount of brickwork required it can be installed at a lower complete cost The elimination of special brick shapes and staybolts in the furnace reduces the cost of maintenance to a minimum Built in units from 25 to 250 Hp for power service, and the 15 lb heating class in units from 3500 to 35000 sq ft of steam radiation





BIGELOW SCOTCH TYPE BOILER

Modeled after the Scotch Marine type of boiler, this boiler has many advantages to warrant its use in the industrial field for power and heating Self contained Compact Low water-line Built in units from 15 to 300 hp The 15 lb welded heating type built in units from 1800 to 35000 sq ft of steam radiation

Combustion Engineering Company, Inc.

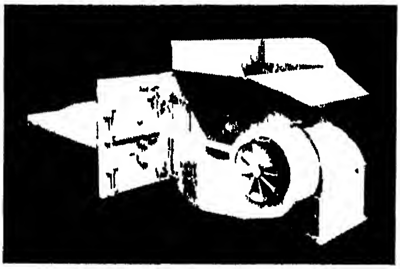
All Types Fire and Water-Tube Boilers
Complete Steam Generating Units

Mechanical Stokers
Pulverized Fuel Systems

200 Madison Avenue, **New York, N. Y.**

Canadian Associate COMBUSTION ENGINEERING CORP., LTD., MONTREAL

CE-SKELLY STOKER UNIT



The CE-Skelly stoker is designed for installation under boilers ranging from small heating units to power boilers developing up to 300 hp

Summary of Features

A Self-Contained Unit Hopper, fuel feeding and distributing mechanism, grate, windbox, driving mechanism and forced draft fan are combined in a compact unit

Hopper Made of rust- and corrosion resisting metal Non-clogging and easily removable Does not interfere with access to furnace doors Located at convenient height for filling with shovel

Coal Feed Screw conveyor, located entirely outside of retort and protected from heat, advances the coal from hopper to entrance of retort Reciprocating ram in retort continues the feeding and provides agitation of the fuel bed in the retort zone

Grate Surface Alternate fixed and moving grate bars Designed for correct air distribution and made of a special heat-resisting metal assuring low maintenance

Air Supply Integral forced draft fan, with inlet damper control, supplies air to windbox under the stoker Volume of air may be regulated by control to suit the rate of coal burning

Control Automatic control furnished as standard equipment

Application For new boilers or for existing boilers having obsolete or inefficient firing equipment Small clearances permit installation with only slight alterations in most cases

Operation Simple, easy, dependable Variable-speed transmission permits 16 rates of coal feed Control levers conveniently located

Write for New 24-page Catalog No. SU-5G

OTHER CE STOKERS

Nearly 11,000 CE stokers installed to date The CE line of larger stokers includes the following, of which Type E, Coxe and Green are among the leaders in their respective fields

Type K Stoker A single retort underfeed stoker for burning bituminous coals under boilers in the upper size range of the CE Skelly Stoker Unit

Type E Stoker A single retort underfeed stoker for burning bituminous coal under boilers up to about 600 rated hp

CE Multiple Retort Stoker For burning semi bituminous and bituminous coals under boilers up to the largest sizes

Coxe Stoker A traveling grate stoker for burning small sizes of anthracite, coke breeze and lignite.

Green Stokers A chain grate stoker available in both natural and forced draft types for burning non caking or free burning bituminous coals

CE BOILERS

All fire tube and water tube types in sizes ranging from 25 hp up to the largest Included are all designs formerly known by the trade names "Heine," "Walsh & Weidner" "Casey Hedges" and "Ladd."

Classified broadly, the various types of CE Boilers are as follows. BENT TUBE multi drum, four drum, three drum, two drum (complete steam generators). STRAIGHT TUBE sectional header, box header (cross drum and long drum) STEAM GENERATORS complete standardized units available in two types. FIRE TUBE hrt, vertical, internally fired, locomotive type MARINE sectional header, bent tube WASTE HEAT straight tube, bent tube

CE PULVERIZED FUEL SYSTEMS

Formerly known by the trade name "Lopulco," available in both direct fired and storage types for boilers ranging from 200 hp up The CE-Raymond Bowl Mill, a pulverizer of advanced design, may be used in either direct fired or storage installations

1-36b

Farrar & Trefts

Incorporated

Buffalo, N. Y.

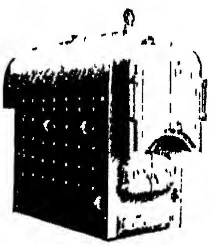
HEATING AND POWER BOILERS

Bison Compact Boilers
Bisonette Compact Boilers
Firebox Return Tubular Boilers
Firebox Locomotive Type Boilers
Scotch Marine Type Boilers
Vertical Boilers
Horizontal Return Tubular Boilers
Bison Two-Pass Return Tubular Boilers

Established 1864

STEEL PLATE CONSTRUCTION

Storage and Pressure Tanks
Receivers, Welded or Riveted
Steel Pipe, Welded or Riveted
Buoys, Welded or Riveted
Condensers and Kettles
Smokestacks and Breechings
Special Work in Stainless Steel, Everdur, Nickel, Aluminum or Monel Metal



The Bison Compact

The **F&T Bison Compact Welded Heating Boiler** is more than just another boiler. It has been designed carefully so as to have a large furnace volume, the proper volume of water, just the right amount of steam liberating surface, the correct volume for steam storage and a balanced circulation. The result is a remarkably steady water line. **A Balanced Boiler.**

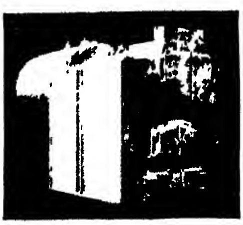
This boiler requires a minimum amount of floor space and is easy and inexpensive to install. It is reasonable as to first cost and economical in operation. Construction is in accordance with the *A.S.M.E.* Code for 15 lb. working pressure and boilers are designed for hand firing with anthracite or bituminous coal or for mechanical firing with oil, gas or stoker. There are various sizes available from 1800 to 35,000 sq. ft. of steam radiation, all ratings as required by the *Steel Heating Boiler Institute.*

The **Bisonette Compact Boiler** has the same characteristics as the larger Bison Compact Boiler. It has been designed for installation in large residences and small business establishments where the advantages inherent in a **Steel** boiler are desired.

Firebox Return Tubular Boiler

Firebox Return Tubular Heating Boilers are **Quality Boilers.** They are constructed to measure up to the high standards set by Heating Engineers and will give unfailing service under all conditions. Being economical to install and operate, they are highly favored by Architects and Engineers for heating Schools, Hospitals, etc.

There are two types of Firebox Boilers, the Up-Draft Type and the Down Draft Type. Both types are made of welded or riveted construction for heating purposes at 15 lb. working pressure and of riveted construction for power purposes at 100, 125 and 150 lb. working pressure in accordance with the *A.S.M.E.* Code. Sizes from 1800 to 35,000 sq. ft. of steam radiation, as rated by the *Steel Heating Boiler Institute,* are designed for hand firing with coal or for mechanical firing with oil, gas or stoker.



The Bison Two-Pass

The **Bison Two-Pass Return Tubular Refractory Lined Boiler** is favored by many Engineers because it **Eliminates Water Legs, Flat Sides and Staybolts.** This boiler is dependable for long years of continuous and economical operation. It stands up under heavy loads and provides that surplus of power so often needed.

This type of boiler is made of welded construction for 15 lb. working pressure and of riveted construction for 100, 125 and 150 lb. working pressure in accordance with the *A.S.M.E.* Code. Boilers are designed for ratings from 25 to 250 hp. and are inspected, tested, and stamped by a representative of a reliable insurance company before shipment. They are furnished for hand firing with coal or for mechanical firing with oil, gas, or stoker.

Fitzgibbons Boiler Company, Inc.

Established 1886

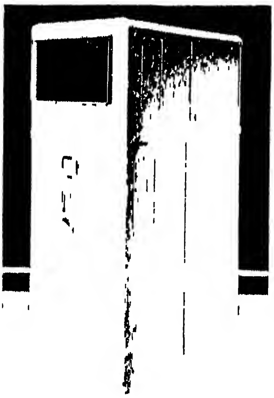
General Offices Architects Bldg., 101 Park Avenue

New York, N. Y.

Works OSWEGO N Y

Branches and Representatives in 59 Principal Cities

PRODUCTS STEEL HEATING and POWER BOILERS Types for all fuels and all heating systems. Sizes to meet heating requirements of all buildings. Built and rated in accordance with S H B I Code BOILER-AIR CONDITIONERS for residences of all sizes



FITZGIBBONS BOILER-AIR CONDITIONER

For Small and Medium Size Homes

Outstanding Features

Provides (1) cleaned, humidified tempered, circulated AIR, (2) Year 'round domestic HOT WATER, without a storage tank, (3) Economical steel boiler HEAT all in a *single, compact unit* requiring no more floor space than an ordinary heating boiler Operates with any good oil burner, gas burner or stoker The ideal "Split System" unit *Bulletin on request*

FITZGIBBONSAIRE

For Larger Buildings

A floor unit which combines with Fitzgibbons Steel Boilers for oil, gas or stoker firing to provide the same three services as the Boiler Air Conditioner described above *Bulletin on request*

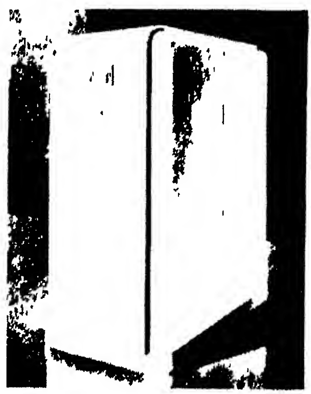
FITZGIBBONS OIL-EIGHTY AUTOMATIC

Residence Steel Boiler for Oil Burning

Ratings, Steam 12 Sizes 125 to 2680 Sq Ft

Outstanding Features

Internal domestic hot water supply coils (optional) **Tanksaver** supplies year-'round hot water without a storage tank, **Tankheater,** an efficient indirect water heater for tank installations **Combustrol,** automatically maintains balanced draft, diverts back drafts, prevents back firing **Thermalizer,** makes every tube do its full share of heat absorbing **Copper-Steel Construction** plate and tubes combines maximum strength with corrosion resistance **Attractively Jacketed.** Teams up with any good rotary or gun type burner to form a unit of **Highest Efficiency.** Burners may be located entirely inside jacket behind large, easily removable panels. *Catalog on request*



FITZGIBBONS GAS-EIGHTY AUTOMATIC For efficient, low cost heating with any good gas burner Has the same outstanding features as the OIL EIGHTY described above including the optional internal domestic hot water supply coils

FITZGIBBONS COAL-EIGHTY AUTOMATIC For domestic stoker firing. Full length jacket prevents escape of fly ash Awarded Certificate of Approval by the Anthracite Institute Ratings, Steam 125 to 2680 Sq Ft

FITZGIBBONS COAL-EIGHTY Jacketed boilers for coal, handfired Ratings, Steam 400 to 1000 Sq Ft

FITZGIBBONS R-Z-U JUNIOR

Multi-Service Steel Boiler

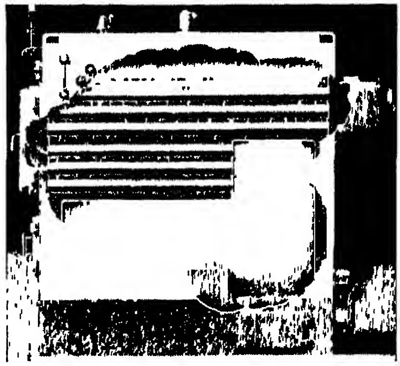
RATINGS, STEAM

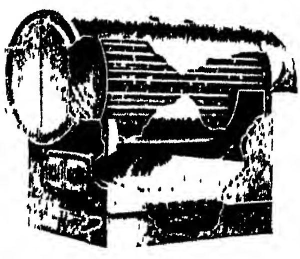
Coal Burning Type	750 to 3200 sq ft
Oil Firing Type .	1003 to 3893 sq ft
Stoker Firing Type	1003 to 3893 sq ft

Outstanding Features

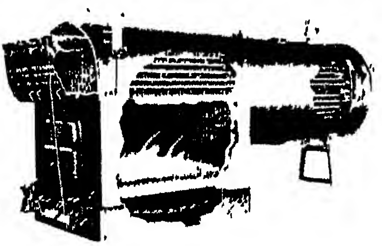
Tanksaver (optional) supplies year-'round hot water without a separate storage tank **Tankheater** (optional) a more efficient indirect water heater **Auxiliary Grate** (optional), for refuse disposal and stand-by heating duty in oil fired installations **Compact,** largest size will pass thru a 31 in doorway **Low Water Line,** eliminates need for a pit **Jacket** (optional), on all types

Descriptive Bulletin on Request

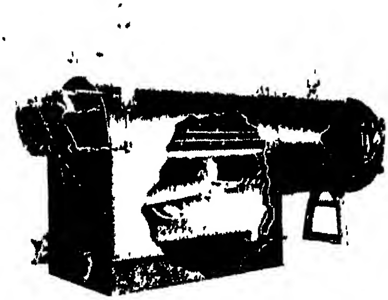




R Z-U



500 Series



600 and 800 Series

FITZGIBBONS Z-U
Steel Firebox Boilers

Built for 15 lb w s p *A S M E* Code
Up-Draft Type 1800 to 35,000 sq ft steam

FITZGIBBONS R-Z-U
Steel Firebox Boilers

The Z-U arranged for rear smoke outlet
Built for 15 lb w s p *A S M E* Code

UP-Draft Type	1800 to 35,000 sq ft steam
Smokeless Type	1800 to 35,000 sq ft. steam
Oil, Gas, Stoker Type	2190 to 12,500 sq ft steam

FITZGIBBONS "F" SERIES
Portable Riveted Firebox Boilers

Built for 100 lb w s p *A S M E* Code
Ratings, steam 1800 to 15,000 sq ft

FITZGIBBONS 500 SERIES
Portable Welded Firebox Boilers Return Tubular

Built for 15 lb w s p *A S M E* Code
Ratings, steam 3500 to 35,000 sq ft

FITZGIBBONS 700 AND "P" SERIES
Portable Riveted Firebox Boilers

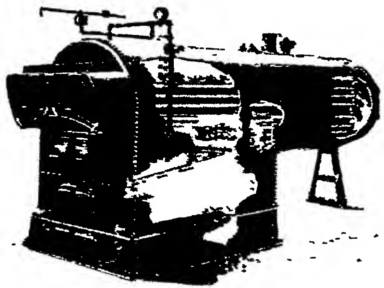
700 Series built for 15 lb w s p *A S M E.* Code
Ratings, steam 3500 to 35,000 sq ft
"P" Series built for 100 lb w s p *A S M.E* Code
Ratings, horsepower 25 to 250.

FITZGIBBONS 600 AND 800 SERIES
Smokeless Down-Draft Riveted Firebox Boilers

Built for 15 to 100 lb w s p. *A S.M E* Code.
Ratings, steam 3500 to 35,000 sq. ft

Descriptive Bulletins on any or all of above boilers will be mailed on request

Kewanee Firebox Boilers
Brickset Type



KEWANEE BOILER CORPORATION

Kewanee, Illinois

BRANCHES IN 61 PRINCIPAL CITIES

Steel Heating and Power Boilers, Water Heating Garbage Burners, Tabasco Heaters and Tanks.

KEWANEE STEEL HEATING BOILERS

Kewanee offers a dependable line of Steel Boilers built for heating every size building, with high efficiency burning any kind of fuel There are 350 standard sizes of Kewanee Boilers most of which are kept in stock, ready for immediate delivery

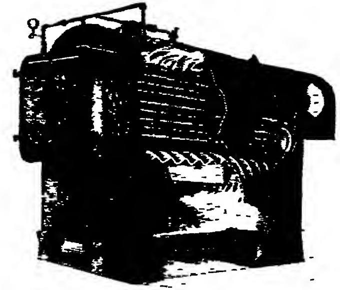
Sixty-eight years of intensive study and effort are back of Kewanee Boiler designs They are all constructed in our extensively equipped factory at Kewanee Illinois in conformity with these Codes *American Society of Mechanical Engineers* for construction and for rating with the *Steel Heating Boiler Institute* Simplified Practice

The Kewanee series include

HEAVY DUTY RIVETED FIREBOX TYPES *1 [illegible] to 35,000 [illegible] Brickset and portable settings Updraft and Downdraft [illegible] Furnace Single-pass tubes for [illegible] Two-pass tubes for [illegible]*

WELDED BOILERS *[illegible] to 35,000 [illegible] Direct Draft [illegible] Corrugated Crown Sheet [illegible]*

RESIDENCE STEEL BOILERS *[illegible] Hot Water [illegible]*



Kewanee Welded
Type C Boilers

SPECIFICATIONS—BRICK-SET UP-DRAFT BOILER

Boiler No		3	4	5	6	8	9	10	11	12	13	14	15	17	18	19	20
Rated Steam Capacity																	
Coal	Sq Ft	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
Oil Gas or Stoker	Sq Ft	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
Width and Length	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	72x15-0
Overall Height	[illegible]	47	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	7	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
Height of Water Line	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
Approximate Weight Coal	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]

SPECIFICATIONS—SMOKELESS DOWN-DRAFT BOILER

Boiler No		376	377	378	379	380	381	382
Rated Steam Capacity								
Coal	Sq Ft	3500	4000	4500	5000	6000	7000	8500
Oil Gas or Stoker	Sq Ft	4250	4550	5470	6080	7290	8500	10330
Width and Length	In x Ft In	42x8-3	42x9-3	48x8-7½	48x9-5	54x11-2½	54x12-11	60x12-8½
Overall Height	In	59	69	80	80	94	94	101
Height of Water Line	In	70	70	73	73	78	78	85
Approximate Weight Coal	Lb	6900	7500	8100	8500	10000	11100	13600
Oil	Lb	6200	6900	7400	8000	9100	10100	12500

Boiler No		383	384	385	386	387	388	389	390
Rated Steam Capacity									
Coal	Sq Ft	10000	12500	15000	17500	20000	25000	30000	35000
Oil Gas or Stoker	Sq Ft	12150	15180	18220	21250	24290	30360	36430	42500
Width and Length	In x Ft In	60x14-9	66x14-11	66x17-4	72x16-5	78x17-0½	78x20-7½	84x19-11	84x22-8
Overall Height	In	101	107	107	113	115	115	125	125
Height of Water Line	In	85	85½	88½	94½	95	95	107½	107½
Approximate Weight Coal	Lb	15300	16000	20500	22900	25100	29500	33600	37500
Oil	Lb	14100	16500	18900	21200	23200	27500	31500	35200

SPECIFICATIONS—PORTABLE UP-DRAFT BOILER

Boiler No		480	481	482	483	484	485	486	487	488	489	490
Rated Steam Capacity												
Coal	Sq Ft	6000	7000	8500	10000	12500	15000	17500	20000	25000	30000	35000
Oil Gas or Stoker	Sq Ft	7290	8500	10330	12150	15180	18220	21250	24290	30360	36430	42500
Width and Length	In x Ft In	54x11-5	54x13-1½	60x13-4½	60x15-5½	66x15-6½	66x18-0½	72x17-0	78x17-7½	78x21-3½	84x20-7	84x23-4
Overall Height	In	94	94	101	101	107	107	113	115	115	125	125
Height of Water Line	In	78	78	85	85	88½	88½	94½	95	95	107½	107½
Approximate Weight Coal	Lb	10200	11400	13300	14800	17300	19700	22000	24200	29400	32300	36100
Oil	Lb	9300	10400	12200	13600	15800	18100	20300	22300	26400	30200	33800

SPECIFICATIONS—TYPE "C" WELDED BOILER

*Boiler No		2773	774 2774	775 2775	776 2776	777 2777	778 2778	779 2779	780 2780	781 2781
Rated Steam Capacity										
Coal	Sq Ft	2200	2500	3000	3500	4000	4500	5000	6000	7000
Oil, Gas or Stoker	Sq Ft	2650	3160	3650	4250	4860	5470	6080	7290	8500
Width & Length	In x Ft in	36x5-10	36x6-4	36x6-10	36x7-8½	42x7-10½	42x8-6½	42x9-2	48x9-4½	48x10-7
Overall Height	In	77½	77½	77½	77½	83½	83½	83½	86½	86½
Height of Water Line	In	69	69	69	69	72	72	72	73	73
Approximate Weight										
700 Series, Coal	Lb		3900	4400	5000	5500	6000	6500	7500	8400
2700 Series Coal	Lb	3400	3800	4300	4800	5300	5800	6300	7200	8100
Oil	Lb	2900	3300	3700	4100	4600	5000	5400	6100	6900

*Boiler No		782 2782	783 2783	784 2784	785 2785	786 2786	787 2787	788 2788	789 2789	790 2790
Rated Steam Capacity										
Coal	Sq Ft	8500	10000	12500	15000	17500	20000	25000	30000	35000
Oil, Gas or Stoker	Sq Ft	10330	12150	15180	18220	21250	24290	30360	36430	42500
Width & Length	In x Ft in	54x9-11	54x11-2½	60x11-6½	66x12-3½	72x12-1½	72x13-4½	78x14-9½	84x14-2½	84x15-11½
Overall Height	In	99	99	108½	112	118	118	122	135	135
Height of Water Line	In	85	85	94	95	101	101	103	114	114
Approximate Weight										
700 Series, Coal	Lb	9700	11000	12900	14900	16600	18400	22000	25200	28400
2700 Series Coal	Lb	9400	10600	12500	14400	16100	17900	21200	24400	27500
Oil	Lb	8000	9100	10700	12300	13800	15400	18100	20900	23400

*Boiler Series 1773-1790 for Oil Gas, Stoker, Series 2773-2790 for Anthracite, Series 7L73-7L90 for Stoker, 2680 Ft to 42500 Ft

SPECIFICATIONS—RESIDENCE TYPE "R" BOILER — ROUND "R"

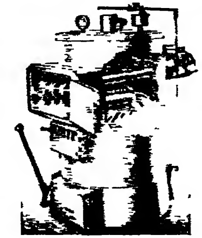
*Boiler No		742	743	745	746	747	748	735	736
Rated Steam Capacity, Coal	Sq Ft	790	1000	1350	1600	1780	1960	410	550
Oil, Gas or Stoker	Sq Ft	840	1120	1470	1900	2160	2380	540	680
Width and Length	In x In	32x39½	32x45½	32x45½	32x51½	32x57½	32x63½		
Diameter	In							23	26
Overall Height	In	59½	59½	70¼	70¼	70¼	70¼	55¼	59¼
Height of Water Line	In	48	45	58½	55½	58½	58½	44	46
Approximate Weight Coal	Lb	2150	2360	2800	3050	3300	3550	1040	1280
Oil	Lb	1900	2060	2500	2730	2920	3120	920	1130
Jacket, Crated	Lb	205	225	175	190	230	225	190	215
Boiler No., Square "R" Oil or Gas		**83R1**	**83R2**	**83R3**	**83R4**	**83R6**	**83R7**	**83R8**	**83R9**
Rated Steam Capacity	Sq Ft	931	1105	1326	1513	2091	2363	2552	2924
Approximate Weight	Lb	1600	1500	2000	2200	2500	2900	3100	3300

*Boiler Series 1742-1748 1735-1736 for Oil, Gas or Stoker 2742-2748, 2735-2736 for Anthracite
Kewanee Indirect Hot Water Heating Coils for Type C Square and Round R Boilers, 130 Sizes 75 to [illegible] Ga[illegible]

Kewanee Square R Oil or Gas Boiler

Kewanee Round R" and Square Type R Residence Heating Boilers for all fuels. Standard snug fitting jackets or Regal. Regal-Square styles for completely enclosing burners are available for Round R and 83R.

Hot Water Copper Coil 75 gal comes with Round R. Capacities up to 720 may be ordered.



Kewanee Round "R" Boiler

Lookout Boiler & Mfg. Company

Successors to the
HEGGIE SIMPLEX BOILER CO
Joliet, Illinois

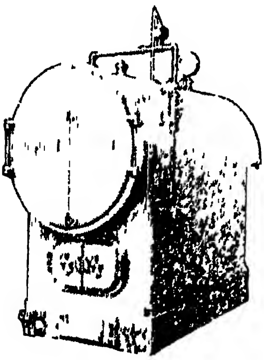
LOOKOUT
REG U S PAT OFFICE

Chattanooga, Tennessee

Represented in Principal Cities

HEGGIE SIMPLEX
REG U S PAT OFFICE

STANDARD SERIES
LOOKOUT HEGGIE-SIMPLEX STEEL HEATING BOILERS



Steam Boiler Assembly

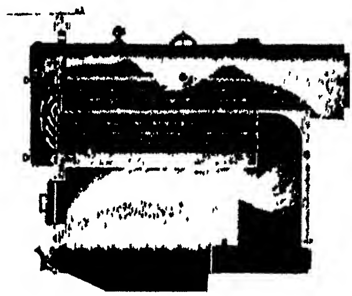
15-Lb. Steam Pressure

Steel Plate

Welded Joints

A.S.M.E. Construction

30-Lb. Water Pressure



For Burning Any Solid Fuel

STANDARD SERIES SPECIFICATIONS

Smokeless Arch Type — **Direct Draft Type** — **Mechanical Fired Type**

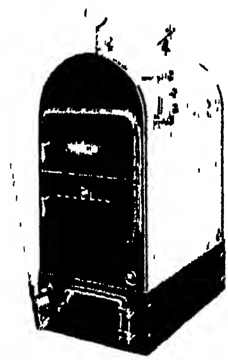
Smokeless Arch Type: Catalog Number Steam Boiler	Smokeless Arch Type: Steam Rating Sq Ft (S H B I)	Direct Draft Type: Catalog Number Steam Boiler	Direct Draft Type: Steam Rating Sq Ft (S H B I)	Mechanical Fired Type: Catalog Number Steam Boiler	Mechanical Fired Type: Steam Rating Sq Ft (S H B I)	Heating Surface Sq Ft (S H B I)	Grate Area Square Feet	Width of Boiler Inches	Length of Boiler, Overall Inches	Height of Base Inches	Height of Boiler on Base Inches	Height of Water Line on Base Inches	Steam Liberating Area Sq Ft	Water Capacity, Gallons	Size of Steam Outlet Inches	Size of Return Openings Inches	Diameter of Smoke Pipe Inches	Boiler Covering Required Sq Ft
134A	1690	334A	1690	734A	2060	121	8.2	34	67¼	12	72	62¼	9.8	171	6	3	18	66
134C	2110	334C	2110	734C	2570	151	10.5	34	79¼	12	72	62¼	12.3	214	6	3	18	78
134E	2530	334E	2530	734E	3080	181	10.4	34	91¼	12	72	62¼	14.8	257	6	3	18	89
140A	3110	340A	3110	740A	3770	222	12.4	40	80¼	12	79	69¼	13.9	280	6	3	20	92
140B	3500	340B	3500	740B	4250	250	13.7	40	86¼	12	79	69¼	15.3	308	6	3	20	98
140D	4120	340D	4120	740D	5000	294	13.7	40	98¼	12	79	69¼	18.1	364	6	3	20	110
140E	4500	340E	4500	740E	5470	322	13.7	40	104¼	12	79	69¼	19.5	392	6	3	20	116
147A	5000	347A	5000	747A	6080	358	14.7	47	101¼	12	86	72½	22.3	476	8	4	22	125
147B	5490	347B	5490	747B	6660	392	16.5	47	107¼	12	86	72½	21.0	513	8	4	22	133
147C	5960	347C	5960	747C	7240	426	16.5	47	113¼	12	86	72½	23.8	552	8	4	22	141
152A	6500	352A	6500	752A	7890	464	18.6	52	103¼	12	91	78½	23.3	551	8	4	24	136
152B	7000	352B	7000	752B	8500	500	18.6	52	109¼	12	91	78½	25.1	596	8	4	24	144
152C	7520	352C	7520	752C	9130	537	20.7	52	115¼	12	91	78½	26.9	641	8	4	24	152
152D	8040	352D	8040	752D	9760	574	20.7	52	121¼	12	91	78½	28.8	686	8	4	24	160
157A	9100	357A	9100	757A	11050	650	22.9	57	115¼	12	97	81¼	30.9	753	8	4	27	167
157C	10420	357C	10420	757C	12650	744	25.2	57	127¼	12	97	81¼	35.1	858	8	4	27	184
157D	11070	357D	11070	757D	13450	791	25.2	57	133¼	12	97	81¼	37.2	910	8	4	27	193
163A	12500	363A	12500	763A	15180	894	28.1	63	123¼	15	108	93	34.9	953	10	5	30	197
163B	13360	363B	13360	763B	16220	954	28.1	63	129¼	15	108	93	37.1	1014	10	5	30	206
163C	14200	363C	14200	763C	17240	1014	28.1	63	135¼	15	108	93	39.3	1076	10	5	30	215
163D	15000	363D	15000	763D	18220	1074	30.6	63	141¼	15	108	93	41.5	1137	10	5	33	224
170A	16020	370A	16020	770A	19450	1144	31.0	70	129¼	15	116	96¼	43.6	1269	10	5	33	230
170B	17020	370B	17020	770B	20670	1216	31.0	70	135¼	15	116	96¼	46.2	1347	10	5	33	240
170C	18030	370C	18030	770C	21900	1288	33.9	70	141¼	15	116	96¼	48.8	1423	10	5	36	250
178A	20000	378A	20000	778A	24290	1429	35.0	78	135¼	15	126	106¼	47.9	1524	10	5	36	258
178C	22390	378C	22390	778C	27180	1599	38.2	78	147¼	15	126	106¼	53.9	1751	10	5	36	281
184A	25000	384A	25000	784A	30360	1786	38.0	84	141¼	15	132	113	52.9	1831	10	5	40	291
184C	28010	384C	28010	784C	34020	2001	41.3	84	153¼	15	132	113	58.9	2042	10	5	40	314
184E	31010	384E	31010	784E	37660	2215	44.8	84	165¼	15	132	113	64.9	2255	10	5	40	337
184G	34010	384G	34010	784G	41290	2429	44.8	84	177¼	15	132	113	70.9	2467	10	5	42	361

To determine S H B I Ratings of Hot Water Heating Boilers, add 60 per cent to corresponding steam boiler rating

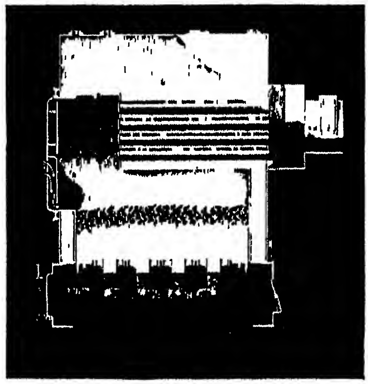
RESIDENCE SERIES
LOOKOUT HEGGIE-SIMPLEX STEEL HEATING BOILERS

Steel Plate **Welded Joints** **A.S.M.E. Construction**
15-Lb. Steam Working Pressure **30-Lb. Water Working Pressure**

Direct Draft Type for Hand Firing any Solid Fuel
Mechanical Fired Type for Oil, Gas, or Stoker



Steam Boiler Assembly with Jacket



For Burning Any Solid Fuel

RESIDENCE SERIES (Double Pass -2 In. Tubes) SPECIFICATIONS
Jacketed and without Jacket

Direct Draft Type **Mechanical Fired Type**

Direct Draft Type		Mechanical Fired Type																
Catalog Number Steam Boiler	Steam Rating Sq Ft (S H B I)	Catalog Number Steam Boiler	Steam Rating Sq Ft. (S H B I)	Heating Surface Sq Ft. (S H B I)	Grate Area Square Feet	Width of Boiler, Inches	Length of Boiler, Overall Inches	Height of Base, Inches	Height of Boiler on Base Inches	Height of Water Line, on Base, Inches	Steam Liberating Area Sq Ft	Water Capacity Full, Gallons	Steam Space Cu Ft	Size of Steam Outlet Inches	Size of Return Opening Inches	Size of Water Heater Openings	Diameter of Stack, Inches	Height of Stack Feet
2SJ1	370	2SCJ1	540	32	2 6	24	34¼	12	56	48	3 8	37	1 8	3	3	1½	10	30
2SJ2	510	2SCJ2	680	40	3 5	24	40¾	12	56	48	4 8	48	2 3	3	3	1½	10	30
2SJ3	670	2SCJ3	820	48	4 3	24	45¼	12	56	48	5 8	60	2 8	3	3	1½	10	30
2SJ4	800	2SCJ4	970	57	5 1	24	53¾	12	56	48	6 8	71	3 3	3	3	1½	10	30
2SJ5	950	2SCJ5	1160	68	5 4	27	47¼	12	65	55½	6 6	103	3 8	4	4	1½	12	40
2SJ6	1110	2SCJ6	1340	79	6 0	27	53¾	12	65	55½	7 7	122	4 5	4	4	1½	12	40
2SJ7	1300	2SCJ7	1580	93	7 0	27	60¼	12	65	55½	8 9	143	5 1	4	4	1½	12	40
2SJ8	1480	2SCJ8	1800	106	7 8	27	66¾	12	65	55½	10.0	163	5 8	4	4	1½	12	40
2SJ9	1620	2SCJ9	1970	116	7 9	30	60¼	12	72	61	10 1	175	6 7	5	5	1½	14	50
2SJ10	1830	2SCJ10	2230	131	9 0	30	66¾	12	72	61	11 3	199	7 6	5	5	1½	14	50

To determine S H B I Ratings of Hot Water Heating Boilers, add 60 per cent to corresponding steam boiler ratings

In addition to the STANDARD SERIES and RESIDENCE SERIES of Steel Heating Boilers, we make the following

HIGH FIREBOX SERIES Mechanical Fired Type Specifications given in Bulletin No 11[illegible]

P. O SERIES PORTABLE FIRE BOX BOILERS (Single Pass) -Down-draft Water-Grate Type Direct Draft Type Mechanical Fired Type Specifications given in Bulletin No 117

F P O SERIES PORTABLE FIREBOX BOILERS (Double Pass) -Down-draft Water-Grate Type Direct Draft Type Mechanical Fired Type Specifications given in Bulletin No 118

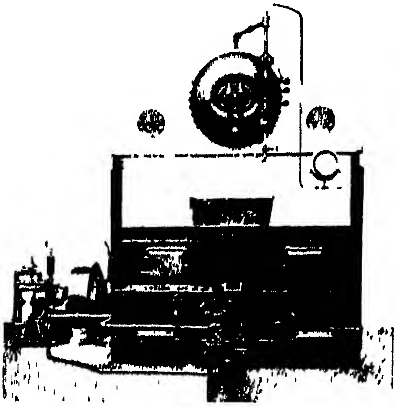
REFUSE BURNING WATER HEATERS Direct Draft Type Specifications given in Bulletin No 119

E. Keeler Company

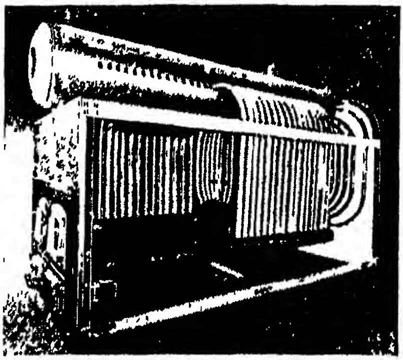
Williamsport, Pa.

ESTABLISHED 1864

Steel Boilers For Heating and Power Steel Stacks, Breechings and Plate Fabrications



Keeler Type "CP" Water Tube Boiler Equipped with Underfeed Stoker

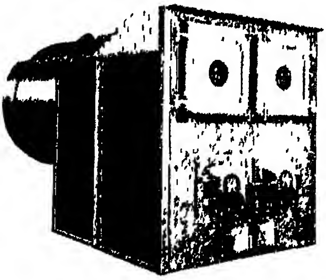


Keeler Type "CP" Water Tube Boiler Partly Assembled

The Keeler Type "CP" Water Tube Boiler Embodies These Important Features

1 It is completely steel encased and insulated
2 No brickwork required except for front wall and bridge wall
3 The sides of the furnace are water cooled
4 Clinkers cannot adhere to sides of furnace
5 Furnace maintenance costs are reduced to a minimum
6 Provides very large capacity in a given space.
7. Can be operated at high overloads without disturbing water level.
8 Has built-in soot blower Easy to clean interior or exterior
9. Made in large range of sizes in pressures up to 450 lb
10 Units as large as 250 hp can be shipped completely assembled
11. Units larger than 250 hp are shipped knocked down
12 Highly efficient with any method of firing

The E Keeler Company manufactures Straight Tube Water Tube Boilers, both long drum and cross drum types, Curved Tube Boilers, both three drum and four drum types, Return Tubular and Double Duty Fire Tube Boilers for every power or heating requirement



Keeler Double Duty Boiler

Keeler Double Duty Boiler Before Encased

Bulletins of Any Type Sent on Request

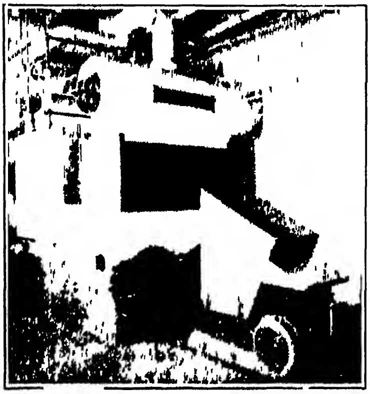
Union Iron Works

Manufacturers of UNION BOILERS For Heating and Power

Erie, Penna.

Representatives in All Principal Cities

HEATING BOILERS KNOCKED-DOWN WATER TUBE TYPE



Advantages

Power Boiler Performance now available in an improved water tube heating boiler for hand, gas, oil, or stoker firing. These boilers are similar to high pressure cross drum boilers for power plants and are specifically designed for heating service. They are built by a leading manufacturer of high pressure power boilers.

Higher Efficiency and Capacity in the same space conditions is possible as a water tube boiler is a better heat absorber than the fire tube type, therefore the UNION heating boiler has the definite advantage of producing more steam per square foot of heating surface.

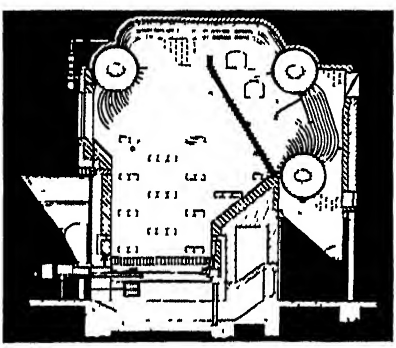
Ideal for Replacement Work as these boilers can be shipped knocked down and brought in through any ordinary building opening.

No Building Alterations Necessary to place this boiler over the foundations. This feature will appeal to the engineer, architect, or contractor who is making a replacement in an existing building.

Easily Assembled, No Field Welding is required. Can be shipped completely assembled or knocked down for field assembly. The only tool required is a tube expander, no welding equipment is needed.

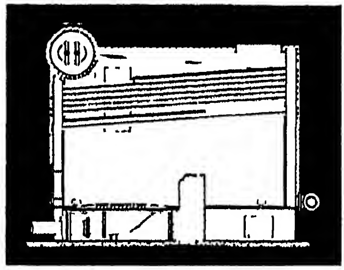
Union heating boilers are for either steam or hot water service and are built for steam ratings from 3,000 to 51,000 sq. ft., and for hot water from 1,800 to 81,000 sq. ft.

The Union Iron Works also builds fire tube and water tube boilers, including return tubular, portable fire tube, long drum and cross drum straight tube, low head and vertical bent tube boilers.



Union L-Type Bent Tube Boiler

Descriptive Bulletins on Union Boilers will be Mailed on Request

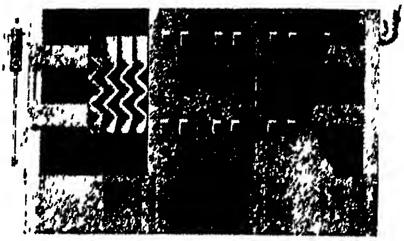


Union Heating Boiler

Waterfilm Boilers
Incorporated

154 Ogden Avenue, **Jersey City, N. J., U. S. A.**

BOILERS DESIGNED AND BUILT FOR AUTOMATIC FIRING



Send for full information today

DE LUXE SERIES

Boiler No	2		3		4		5	
Recommended Load Square Feet I D R Pickup included in these ratings	Steam 500'	Water 800'	Steam 800'	Water 1280'	Steam 1100'	Water 1760'	Steam 1400'	Water 2240'
Guaranteed B T U	120,000		192,000		264,000		336,000	
Maximum Firing Rate G P H	1.65 gal to 2 gal		2 gal to 2½ gal		2½ gal to 3 gal		3½ gal to 4 gal	

DE LUXE SERIES completely jacketed with either short to cover boiler only, or long to cover boiler, burner and domestic hot water tank

SECTIONAL SERIES

Single Series

No	Recommended Load I D R Steam	Recommended Load I D R Water	Guaranteed B T U	No	Recommended Load I D R Steam	Recommended Load I D R Water	Guaranteed B T U
1 5	1,250	2,000	330,000	1 12	3,000	4,800	750,000
1 6	1,500	2,400	390,000	1 13	3,250	5,200	810,000
1 7	1,750	2,800	450,000	1 14	3,500	5,600	870,000
1 8	2,000	3,200	510,000	1 15	3,750	6,000	930,000
1 9	2,250	3,600	570,000	1 16	4,000	6,400	990,000
1 10	2,500	4,000	630,000	1 17	4,250	6,800	1,050,000
1 11	2,750	4,400	690,000	1 18	4,500	7,200	1,110,000

Double Series

No	Steam	Water	Guaranteed B T U	No	Steam	Water	Guaranteed B T U
2 8	4,000	6,400	990,000	2 25	12,500	20,000	3,030,000
2 9	4,500	7,200	1,110,000	2 26	13,000	20,800	3,150,000
2 10	5,000	8,000	1,230,000	2 27	13,500	21,600	3,270,000
2 11	5,500	8,800	1,350,000	2 28	14,000	22,400	3,390,000
2 12	6,000	9,600	1,470,000	2 29	14,500	23,200	3,510,000
2 13	6,500	10,400	1,590,000	2 30	15,000	24,000	3,630,000
2 14	7,000	11,200	1,710,000	2 31	15,500	24,800	3,750,000
2 15	7,500	12,000	1,830,000	2 32	16,000	25,600	3,870,000
2 16	8,000	12,800	1,950,000	2 33	16,500	26,400	3,990,000
2 17	8,500	13,600	2,070,000	2 34	17,000	27,200	4,110,000
2 18	9,000	14,400	2,190,000	2 35	17,500	28,000	4,230,000
2 19	9,500	15,200	2,310,000	2 36	18,000	28,800	4,350,000
2 20	10,000	16,000	2,430,000	2 37	18,500	29,600	4,470,000
2-21	10,500	16,800	2,550,000	2 38	19,000	30,400	5,630,000
2 22	11,000	17,600	2,670,000	2 39	19,500	31,200	5,750,000
2 23	11,500	18,400	2,790,000	2 40	20,000	32,000	5,870,000
2-24	12,000	19,200	2,910,000	2 41	20,500	32,800	5,990,000

Electrol Incorporated

FINE OIL HEATING EQUIPMENT EXCLUSIVELY SINCE 1918

931 Main Ave, **Clifton, N. J.**

Conversion Burners, Boiler-Burner Units, Air Conditioning Units

Quiet, All Electric ELECTROL OIL BURNER

Electrol-Kewanee Heating Unit

A complete boiler burner unit consisting of (1) exclusive design, horizontal, welded copper bearing steel firetube boiler, (2) Electrol oil burner mechanism, and (3) domestic water heater Induced draft, low stack temperature Integral flame Low water and limit controls Burns heavy furnace oil Guaranteed minimum 80 per cent over all efficiency Simple, fully automatic Attractive, sound and heat insulated jacket

Unit	Cap I D R Steam	Gal Oil per Hour	Gals D H W Three Hours	Over-all Dimensions In		
				L	W	H
E K 5	750	1 6	50	47 1/4	34	55 1/4
E-K 10	1250	2 7	80	60 1/2	34	55 1/4
E-K 15	1750	3 75	100	75	34	55 1/4

Electrol Oil Burner

Famous for nearly 20 years for dependable, economical operation in any boiler or furnace Curved air chamber produces rotating air and ball shaped flame Built in Master Control, a safety that cannot become clogged with soot Continuous electric ignition Cut Off Valve at Nozzle New parts fit any existing models eliminating obsolescence Burns low cost fuel oil One moving part Simple, rugged, fully automatic Summer Winter Dom H W. with indirect water heater

Model Sizes	Gal Oil per Hour		Capacity E D R Steam
	Min	Max	
EC	1 3	3 1/2	200 1100
EJ	1 3	2 1/2	200 800*
EU	2	8 1/2	600 2800
EM	12	24	2500 10,000

Electrol Conversion Oil Burners should be selected to heat a boiler to its full rated capacity regardless of the load that may be connected to the boiler

*Standard Equipment with the Model EJ is 1140 rpm motor With 1725 rpm motor, capacity is same as model EC

Electrol-American, Series No. EA 6 Heating Unit

A complete automatic boiler-burner unit for the moderate-sized home Boiler of cast iron construction Specially designed heat absorbing surfaces Biltin-Taco Water Heater, capacity 40 gals (100 gal heater can be supplied) Includes safety controls, Low Water Cut Off, safety valve and pressure limit switch Boiler trim water column, combination pressure and vacuum gauge on steam boilers, altitude gauge and thermometer on hot water boilers Fired by Electrol Model TCV Burner Burns low cost fuel oil Heavily insulated, sound proof jacket finished in beige enamel Burner, smokehood, water heater and controls completely enclosed

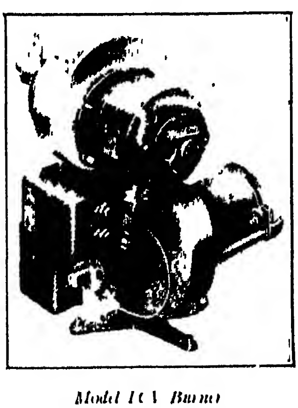
No of Boiler	Installed Radiation Boiler will Carry Sq Ft (4500 Btu Transmission Rate)		Lbs Oil Hr (4500 Btu Transmission Rate)	Capacity Taco Water Heater Gals	Overall Dimensions Ins		
	Steam	Water			L	W	H
EA6-4	355	570	8 5	40	47 1/16	29 3/4	54 5/8
EA6-5	435	695	10 3	40	50 11/16	29 3/4	54 5/8
EA6-6	515	825	12 1	40	54 1/16	29 3/4	54 5/8
EA6-7	595	950	13 5	40	57 11/16	29 3/4	54 5/8
*EA6-8	675	1080	14 9	40	61 1/16	29 3/4	54 5/8
EA6 9	755	1205	16 3	40	64 11/16	29 3/4	54 5/8

*100 gals at extra cost

Electrol Model TCV Burner Vertically Mounted

The vertical mounting of the motor, fan housing and oil pump, makes possible the use of the TCV Burner with jacketed boilers and furnaces

It embodies all the unique Electrol Features.



Model TCV Burner

Model Size TCV	Dimensions, Ins			Gals, Oil per Hr	
	H	W	L	Min	Max
	25 1/4	21	19 3/4	1 3	3 5

Capacity Equivalent Direct Radiation	
Steam	Hot Water
200-1100	350-1800

See also Pages 846-847

S. T. Johnson Co.

Branches
San Carlos, Calif
San Francisco, Calif
Sacramento, Calif
Philadelphia, Pa

Executive Offices and Factory

940-950 Arlington Avenue, Oakland, Calif.

OIL BURNERS FOR EVERY HEATING AND POWER PURPOSE

EXPERIENCE

The S T Johnson Co , established in 1904, is a pioneer in the use of liquid fuels for every heating and power purpose

PRODUCTS

Manual, Semi-Automatic and fully Automatic Rotary Oil Burners built in various sizes to meet the requirements of any domestic, commercial or industrial application

Oil Fuel Combination Boiler Burner units furnishing heat or hot water or both in combination for homes

Gas fired Boiler Burner units furnishing heat and domestic hot water where desirable

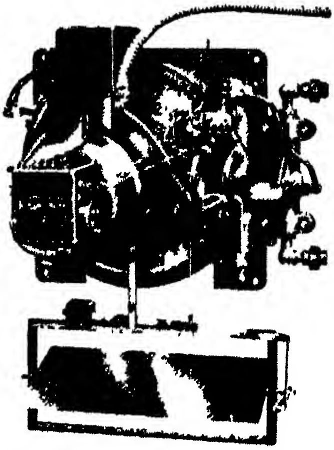
Combination Oil and Gas Burners for commercial and industrial applications

ARCHITECT AND ENGINEER SERVICE

Through the Johnson Engineering Department your heating or power problems will be studied without obligation and full assistance given in preparing layouts and estimates

BULLETINS

Loose-leaf bulletins on Johnson Oil Burners or boiler burner units and bricking and installation plans for boilers and furnaces most generally encountered are available from the factory or Johnson distributors in principal cities



Type 30-AV I F S A M I

Type 30-AV Johnson Oil Burner Swung back, Out of Firing Position Without Disconnecting Any Pipe, When It Is Desirable to Make an Inspection or Mechanical Adjustment

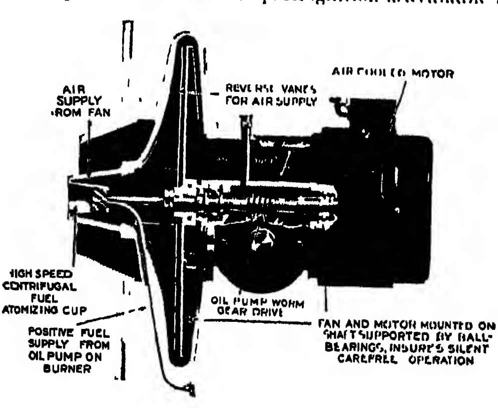
COMMERCIAL FULLY AUTOMATIC ROTARY TYPE OIL BURNERS

Types 30-AV and 30-H

These automatic burners are made in 5 sizes for boilers up to 11,700 sq ft of steam radiation Controlled automatically by a thermostat, steam pressure, water temperature, or a combination of these devices as the application may require

Standard ignition is electric gas The gas may be natural, manufactured or compressed gas in containers

On special order electric spark ignition is available for fuels not heavier than No 3 on smaller burners.



Type 30-AV uses up to No 5 fuel without preheating and is equipped with the Johnson viscosity compensating valve furnishing a constant flow of fuel, regardless of fuel temperatures or viscosities, to the burner without building excessive pressure in the lines or putting undue strain on the pump or motor The valve incorporates a spring loaded floating piston with three orifices registering with delivery ports to the burner atomizer, to the suction side of the pump and to the oil return to storage. Adjustments are made by moving a lever around a notched quadrant

which changes the size of the metering orifices

Type 30-H is equipped with an automatic oil preheater and features that make possible the efficient use of No 6 (Bunker C) fuel oil. The preheater maintains warm oil enabling burner to start instantly at all times.

Johnson Low Fire Starting and Modulated Firing Accessories. These features are incorporated in Type 30-AV LFS & MF and Type 30-H LFS & MF Johnson Burners. Also available for Type 28 Burner.

Low Fire Starting automatically reduces oil and air to a minimum for starting purposes and during a short pre-determined period, increases the fuel and air to full flame, assuring smooth positive starting.

Modulated Firing takes charge of the burner after full flame is established and automatically varies the flame to suit boiler demands through control of air and fuel. At shut down periods primary and secondary air are closed thus conserving furnace heat. Fuel savings have run as high as 7 per cent with this type of operation.

Laddi' Du-All

SEMI-AUTOMATIC TYPE 28

A manually ignited burner made in five sizes for boilers up to 41,700 sq ft of steam radiation. Built in two types: one with built in pump for single installations, the other for use with separate Johnson pump equipment on multiple installations. Fuel oils (preheated when necessary) as heavy as No 6 may be used.

'LADDI' DU-ALL

Combination Boiler-Burner Unit for Small and Medium Sized Homes

Furnishes heat or domestic hot water or both in combination at low cost.

In the combination units fuel is only consumed for the production of domestic hot water during warm weather.

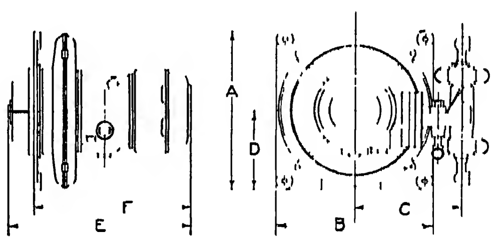
Made in three sizes: 100,000, 120,000, and 140,000 Btu per hour output.

'Laddi' Du All Units rest directly on the floor. Diameter at the base is 21 in on all three sizes, while the heights run 4½ ft on No 1, 5 ft on No 2 and 5½ ft on No 3.

General Specifications of 'Laddi' Du-All

Vertical 5/16 in copper bearing steel boiler shell electrically welded. Tested to 150 lb hydrostatic pressure. Horizontal fire box with pre-cast refractory combustion chamber. 1¼ in seamless extra heavy copper bearing steel fire tubes. 6 in reversible chimney connector for either top or back connection. Completely insulated with mineral wool and heavy steel enameled jacket. Johnson Type B burner, 110 volt a c, 60 cycles.

Burns Diesel fuel oil or No 3 U S Standard. Unit fully assembled and tested before shipment. Finish of enamel in Royal Purple and Brewster Green. Fire tubes easily accessible.



S. T. Johnson Co. Rotary Oil Burners Types 28, 30-AV, 30-H

Burner Size No	Capacity Steam *Radiation, Sq Ft Min	Max	Boiler Hp Rating	Gal Oil per Hour Min	Max	Size Motor Hp	Dimensions in Inches A	B	C	D	E	F
2½	825	2775	20	2	7	⅓	14	13⅞	9½	6 5/16	17	15
3½	1400	6950	50	3	15	½	16 3/16	15⅞	10¼	8¼	17½	15½
4½	2775	13900	100	6	30	¾	17	19½	13	9 5/16	21	19¼
5½	5550	27800	200	12	60	2	21½	21½	13	11	24	20
6½	8325	41700	300	24	90	3	21½	21½	13	11	25	20½

*Boiler output. Above burners available without built in pumps in same sizes and capacities.

Kelvinator

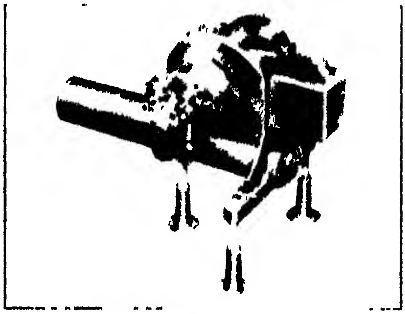
Division of Nash-Kelvinator Corporation

AUTOMATIC HEATING

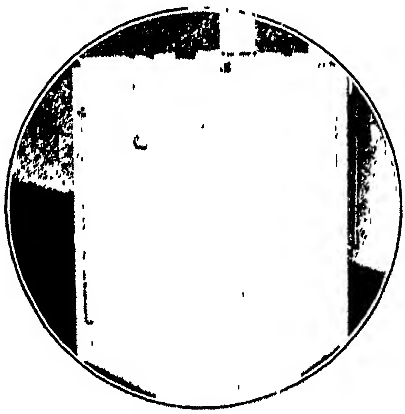
Representatives in all Principal Cities

Factories in Detroit, Michigan, and London, Ontario

Kelvinator Oil Burners

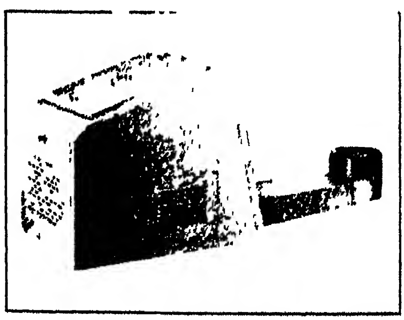


The advanced design of **Kelvinator Oil Burners** is based on Kelvinator's 23 years of experience in the temperature control field. Outstanding features include a **fuel control unit** that can be regulated during installation to meet the exact needs of the individual heating plant, and an **adjustable air shutter** that permits accurate setting. The reliable **non-pulsating pump** is unusually quiet, and the **fan housing** in direct alignment with the air supply tube promotes noiseless operation of the **high-capacity blower**. An **air turbulator** inside the air supply tube assures an intimate mixture for complete combustion, and a special **air agitator** gives a soft, floating flame which can be adjusted to conform to the shape of the individual combustion chamber. Radio interference and ignition failure are obviated by the use of an **ignition transformer** of advanced construction.



Pressure-type Oil Burners. For installation in all types of heating plants. Three sizes, for equivalent steam radiation loads of 350-750 sq ft, 750-1300 sq ft, and 1300-2130 sq ft. Uses **AOBA** number 1, 2, or 3 fuel oil.

Boiler-Burner Units: For steam, hot water, or vapor heating systems. Three sizes, covering requirements of 500 to 780, 800 to 1230, and 1080 to 1635 sq ft of equivalent steam radiation loads. Uses **AOBA** number 1, 2, or 3 fuel oils. Built-in water heater for year-'round domestic hot water supply. Burner can also be used in warm air conditioning units.



Automatic Coal Burners: For installation in all types of heating plants. Burns lower grades of bituminous coal. Feeds fuel (underfeed principle) as heating load requires. Automatic time relay starts burner regularly to prevent fire from going out. Five sizes in capacities of 875, 1250, 1875, 2500, and 3750 sq ft of equivalent steam radiation.

See also Pages 867-869

Micro-Westco, Inc.

Bettendorf, Iowa

Branch Offices

90 West St., New York City
333 S. Mich. Blvd., Chicago
221 State St., Boston, Mass.
1120 Chestnut St., Philadelphia, Pa.

Model D

Model G

Model F

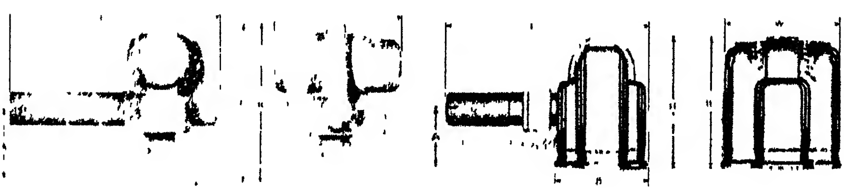
Bettendorf Burners are all electric, fully automatic and incorporate the high pressure method of atomization. Listed as standard by the Underwriters Laboratories Inc. to burn domestic fuel oils, Nos. 1, 2 and 3, and approved by the fire prevention bureaus of all principal cities.

Manufactured in four models with a range of capacity from ⅞ of a gallon to 9 gal. per hour. For dimensions, ratings etc. see chart below.

All Bettendorf Burners are built around a rigid and heavy one-piece main housing which provides an accurate and rigid means of attaching the various parts, assuring perfect alignment of all moving parts, as well as positive interchangeability of parts.

The Bettendorf is positively not a "One Feature" burner. Instead it combines those proven features of modern design for the efficient burning of oil in all types of applications.

Micro Westco, Inc., extends the services of its Engineering Department to Architects, Heating Engineers and Contractors on Oil Heating Problems.



Model	Dimen. A Min	Dimen. A Max	Dimen. H Min	Dimen. H Max	Dimen. L Min	Dimen. L Max	Dimen. B	Dimen. W	Motor HP	Cap. GPH	Cap. Steam Rad.	Cap. H.W. Rad.	Cap. Warm Air Leader Pipe
D	7½ in.	11½ in.	19 in.	23 in.	24 in.	23 in.	12 in. dia.	20 in.	1/6	2 to 4	1000 sq. ft.	1600 sq. ft.	1400 sq. in.
F	6½ in.	9 in.	17 in.	20 in.	21 in.	30 in.	14 in.	18 in.	1/8	1.35 to 2.5	625 sq. ft.	1000 sq. ft.	875 sq. in.
G	6¼ in.	8½ in.	16 in.	18 in.	20 in.	29 in.	10 in. dia.	18 in.	1/8	1.35 to 2.5	625 sq. ft.	1000 sq. ft.	875 sq. in.
HD	8½ in.	12½ in.	22 in.	26 in.	33 in.	38 in.	15 in. dia.	22 in.	1/3	5 to 9	2250 sq. ft.	3600 sq. ft.	

Norge Division

Borg-Warner Corporation

606-670 E Woodbridge Street, **Detroit, Michigan**

Norge Whirlator Oil Burner

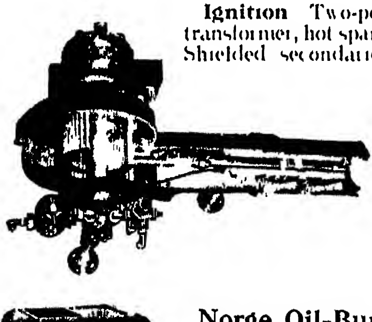
Serviced through franchised distributors and dealers throughout the United States

Norge Whirlator Oil Burner is a "Pressure-Atomization-type" burner built in six sizes, Model 8, handles 800 sq ft of steam radiation or its equivalent, Model N-18, handles 1800 sq ft, Model N 28, 2800 sq ft, Model N-38 handles 3800 sq ft, Model N-68, handles 6800 sq ft, Model N-88, handles 8800 sq ft

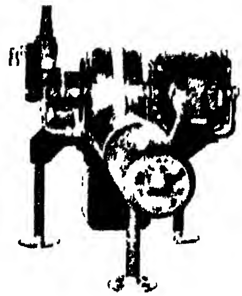
The "Whirlator" is a patented design of multiple vanes, rotating entire mass of air in a smooth even movement, to provide ad-mixture for high combustion efficiency

Nozzle atomizes 28 deg -30 deg Baume, known as No 3 A O B A specifications, or lighter oils

Motors – (Model N-8) Split Phase, 110 volts, 60 cycle, alternating current, 1750 rpm Model N-8, 1/10 hp motor Model N-18, 1/6 hp, Model N-28, 1/5 hp Larger models powered in proportion



Ignition Two-pole transformer, hot spark Shielded secondaries, built in radio interference eliminator Tested for maximum breakdown of 15,000 volts



Controls Minneapolis Honeywell Series 10, low voltage room thermostat, type R 117 3 Protectoray (stack mounted) or combustion safety device and boiler or limit control (either high or low voltage, as desired)

Pump Rotary type Direct flexible spring coupling to motor shaft

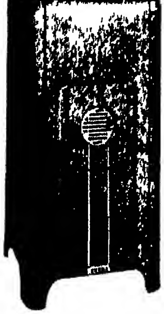
Fan Fabricated light aluminum, mounted on motor shaft No intermediate connections

Strainers A series of two with ample capacity

Accessibility Fan or blower housing, and inspection cover in one light weight casting.

Legs Three sturdy legs, easily adjustable

Approvals Listed as standard by Underwriters' Laboratories, and approved by leading state and municipal safety boards of fire departments



Norge Oil-Burning Space Heaters

Built in three sizes, both standard and Deluxe models Standard models finished in wrinkled lacquer Deluxe models in Stippled Brown porcelain Norge double pot type burner uses average run of No 1 fuel oil or lighter Blue flame sleeve type of heater where public preference requires this type

Model	37-6	50 6	75 6
Height	39-in.	44 in	52 in
Width	20 in	24 in	28 in
Depth (overall approx)	25 in	27 in.	33 in
Maximum Oil consumption per hour	3/8-gal.	1/2 gal	3/4 gal
Btu per hour	37 500	50,000	75 000
Shipping weight (approx)	155 lb	223 lb	313 lb

See also Page 873

Butler Manufacturing Company

1282 Eastern Ave — **Kansas City, Missouri**

THE AUTOMATIC BUTLER COAL STOKER

14 Sizes. Feeding Capacities 18 to 1,000 Lb Per Hour. Hopper Capacities Up to 1,400 Lb. Boiler Capacities up to 262 Hp Steam Radiation up to 29,089 Sq Ft Hot Water Radiation Up to 46,542 Sq Ft.

Controls Minneapolis-Honeywell Thermostats, magnetic switch and relay transformer or M-H Relay, limit controls either pressuretrol, aquastat or vaporstat

Motor Standard high grade ball bearing for any standard voltage, frequency, phase or direct current

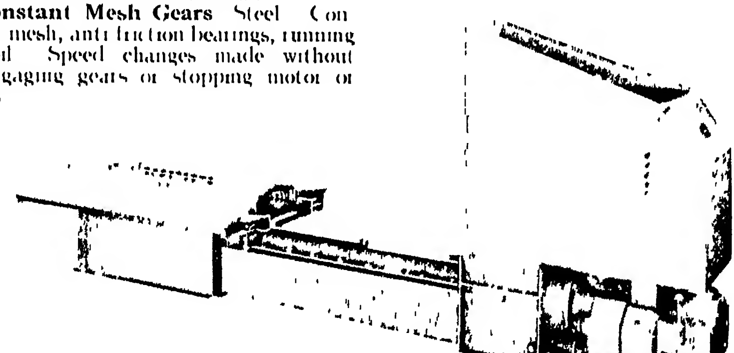
Fan Cast aluminum mounted independently on own ball bearings not on motor

Drive Three center, double V-belt system drives transmission and fan

Constant Mesh Gears Steel Constant mesh, anti friction bearings, running in oil Speed changes made without disengaging gears or stopping motor or gears

Zoned Air Control Steel wind box under retort with air damper Permits controllable volume of air to retort zone and to entire zone of live side grates Result is uniformly active fire bed over entire grate area and not just in retort zone

Hopper Vertical and equipped with up and down and sidewise agitator to insure positive coal feeding into feed screw inlet



Coal Duct Cast iron, equipped with removable serrated jaw breaker plate to trap foreign matter accidently entering coal duct.

Feed Screw Straight flight cast of special alloy steel torsion tested to withstand tremendous stresses

Pulsator Gear driven directly off screw shaft Adds ram feeding action to screw feeding Distributes fuel evenly over fire bed Affords two way agitation to fuel mass to keep heavy coking formations broken up prevents blow holes increases combustion efficiency.

Retort Chrome nickel alloy, narrow tuyere blocks and side grates easily replaced by removing locking rods Retort suspended to allow for expansion and contraction without setting up stresses, which otherwise distort retort or crack its setting.

Self-Coaling Bin To Burner Models Designed with side fuel flow on separate worm screw which carries fuel from bin to stoker screw Operated by independent motor and gear drive Automatic trip control regulates flow of coal into stoker base only as required by firing conditions Also supplied with dust tight steel bins shipped in sections to be bolted together

Complete Manual Gives detailed capacities on coals of various Btu content, combustion space requirements, installation dimensions and drawings, heating cost tables and performance records Write Stoker Division Authorized distributors widely located

Iron Fireman Manufacturing Company

Automatic Coal Burners

Portland, Oregon

Factories PORTLAND, ORE, CLEVELAND OHIO, TORONTO, CANADA

Retail Branches or Subsidiaries

CHICAGO, ILL MILWAUKEE, WIS ST LOUIS, MO NEW YORK N Y MONTREAL, CAN

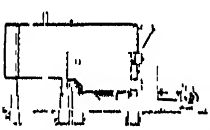
Dealers in Principal Cities and Towns in the United States and Canada

Representation in numerous foreign countries

IRON FIREMAN Automatic Coal Burners

"Forced Underfiring" Principle -Iron Fireman "Forced Underfiring" is based on the scientific principle of feeding fuel to the fire from below, under forced draft From the conveyor screw coal enters the firebox under the fire and is gradually forced upward into the flame As the coal approaches the fire, it is gradually heated The volatile gases are distilled off in the presence of an excess of oxygen and are thoroughly ignited while passing through the incandescent fuel bed This insures complete combustion The ash is fused into clinkers which are easily removed

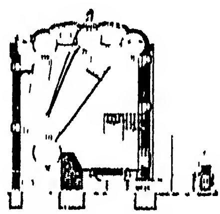
Advantages - Iron Fireman saves money and increases heating plant efficiency in four major ways (1) Cuts fuel costs, (2) Reduces labor costs, (3) Provides steady, even heat or power, (4) Eliminates the smoke nuisance



Typical Installation Down Draft Firebox Boiler

Installation and Sizes Iron Fireman is made in a range of hopper and bin feed sizes for commercial heating and power boilers developing up to 500 hp and also for homes It can be installed quickly in practically any solid fuel boiler or furnace, old or new Machines are shipped complete from the factory All parts are standard and interchangeable

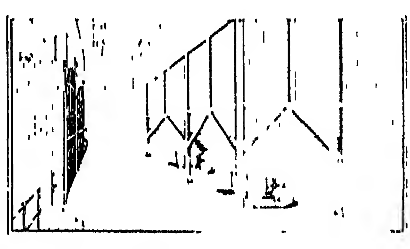
Features of Design and Construction Construction and operation of the Iron Fireman are characterized by simplicity throughout Outstanding features of design and construction are (1) Pressed steel construction (2) Special patented transmission three speeds and neutral Gears run in bath of oil (3) Electric motor standard make (4) V belt drive (5) Safety shear pin protects mechanism from damage (6) Quiet ball bearing fan supplying forced draft to fire (7) Automatic fire banking damper conserves fuel and holds fire in proper condition when stoker is idle (8) Positive pneumatic fume eliminator an auxiliary air supply that insures positive movement of all gases through the fire (9) Volumeter that supplies exactly the amount of air needed for perfect combustion regardless of fuel bed conditions or type of coal used (10) Sectional retort especially designed to allow for heat expansion (11) Dead plates of heavy iron and ribbed (12) Sectional, self cleaning tuyere blocks. (13) Conveyor screw cast of special Iron Fireman alloy steel from one piece pattern (14) Automatic electric controls designed for and used exclusively on Iron Fireman



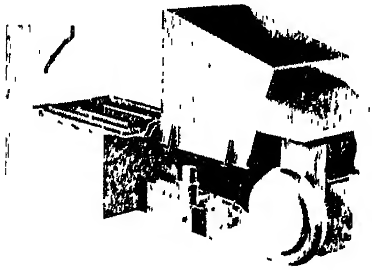
Typical Installation, Four Drum Water Tube Boiler

Automatic Controls Iron Fireman starts and stops at the command of sensitive, accurate automatic controls Directing controls govern stoker operation according to demands of time, tempera

Iron Fireman in Operation in Horizontal Return Tubular Boiler, Low Bridge Wall



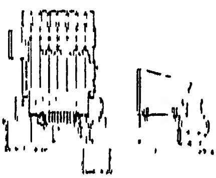
Multiple Installation of Bunker Feed Stokers



Commercial Model For Heat or Power

ture, or pressure An example of the superiority of Iron Fireman directing controls is the "Syncro Stat" which provides automatic control of day and night temperature Other directing controls include pressure regulators, hot water and furnace regulators, and the "Timetactor," a device which runs the Iron Fireman during predetermined intervals in order to keep the fire alive during mild weather

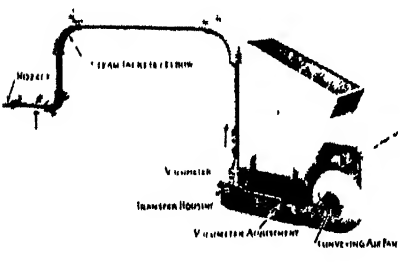
The most important unit of the operating control system is the motor driven relay switch This device starts and stops the stoker motor at the command of the Syncro Stat or other directing controls In the case of the larger stokers a magnetic operating switch works in conjunction with the relay switch



Typical Installation Cast Iron Boiler

Iron Fireman for Homes The Iron Fireman residential model employs "Forced Underfiring" principle the same as larger machines, with simplified operation

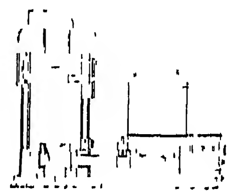
Can be recommended for any steam, hot



Pneumatic Spreader Model For Boilers Developing up to 400 H P

Domestic Installation Coal Flow model that carries coal direct from bin to fire

water, vacuum, or warm air furnace Quickly installed Hopper and bin-feed models for both bituminous and anthracite coal Anthracite models have been tested and approved by The Anthracite Institute



Typical Installation Domestic Boiler

ENGINEERING SERVICE

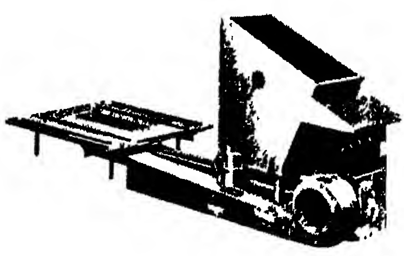
The Iron Fireman organization is nationwide Trained men backed by one of the largest manufacturing organizations in the field are at your service to help you with the experience and practical heating information gained through servicing thousands of boiler rooms and heating plants in all parts of the country

Any Iron Fireman engineer will gladly call and submit any additional information requested

CATALOG AND INFORMATION

Catalogs give full information about the Iron Fireman Descriptive folders give special data about installation in particular types of industries and in homes

Secure them by addressing the factory or any Iron Fireman representative



Industrial "Poweram" Model For Heat or Power

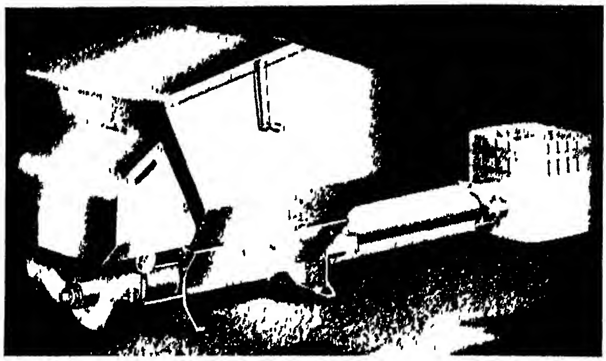
Combustioneer Division

of The Steel Products Engineering Company

AUTOMATIC COAL BURNERS FOR HOME AND INDUSTRY

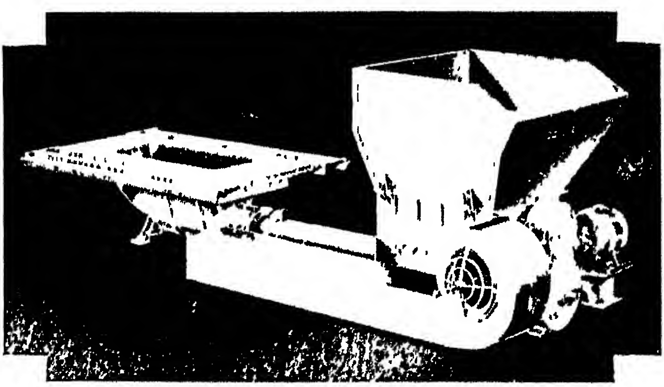
Springfield, Ohio

Combustioneer Automatic Coal Burners are designed for homes and industry. They are the result of many years of experience in the design, construction and operation of coal burning equipment. Air and fuel in correct proportions are automatically fed into the fuel bed. Uniform temperature, utmost convenience, comfort, automatic healthful, safe warmth always assured.



A Domestic Model of Combustioneer

The Domestic model of **Combustioneer** has been developed for homes and small buildings. It is economical in fuel consumption, provided with accurate automatic controls and employs the **Combustioneer Breathing Fuel Bed** and **Automatic Respirator**. Fits all types of small domestic heating plants. Industrial models have been designed to meet the requirements of boilers up to 300 hp. Forced underfeed principal of firing. Intermittent feed gear box. Automatic air control. Sectional tuyeres and dead plates.



An Industrial Model of Combustioneer

Complete information, specifications and engineering data furnished on request.

The Marley Company

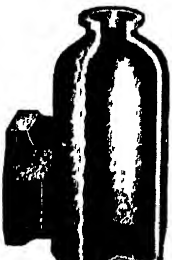
1915 Walnut St., Kansas City, Mo.

Representatives in Principal Cities

Manufacturers of

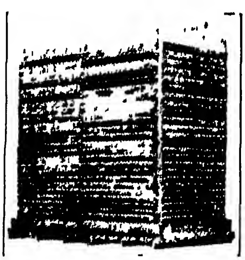
Marley Patented Spray Nozzles, Spray Towers, Spray Deck Towers, Spraycoil Towers, Forced Draft Towers, Spray Ponds, Louvre Fence and all Accessories, Atmospheric Heat Exchangers.

Marley Patented Spray Nozzles



Furnished in a wide variety of sizes and capacities for service with spray ponds, cooling towers, air washers, brine lofts, aerators, etc. Marley nozzles are of the centrifugal type with a long whirl chamber, free, non-clogging passages and no moving parts. The large venturi inlet is so designed as to impart the proper water whirl, producing a full free break up at discharge orifice with minimum friction loss. **Marley Bulletin No. 59.**

Marley Atmospheric Spray Towers



Cast iron louvre posts, slip fit red wood louvres, extremely effective down spray distribution system plus standardization of design, are a few of the outstanding features of this type of Marley cooling tower. Completely shop fabricated and correctly engineered, these towers provide a water cooling unit of rugged construction, good efficiency and low maintenance at minimum cost. **Marley Bulletin No 68**

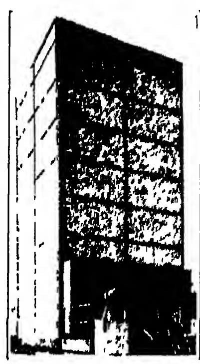
Marley Two-Piece Spray Nozzles

For air washers, brine spraying, etc. Capacities and angle of spray can be varied over wide limits. All sizes made with removable base plugs and machined internally. **Marley Bulletin No. 59**

Marley Small Series Forced Draft Towers

Furnished with steel shells and basins for indoor or outdoor operation. Completely shop fabricated for field assembling. These towers meet such requirements as complete elimination of drift, positive cooling performance regardless of location, extreme thermal efficiencies and quiet operation. Fans, drives, drift eliminators and filling, as well as operating features and efficiency, have been developed by the Marley engineers to a high state of perfection. **Marley Bulletin No 67.**

Marley Standard Forced Draft Towers



Obtainable in any desired capacity. Made of either wood or steel. Features include zig zag eliminators, multiple banked spray filling, double cased shells, fan dampers, cast iron cased inlet fan rings, adjustable pitch fan blades, flexible and independent operating control of both water and air. **Marley Bulletin No. 80.**

Marley Humidifying Spray Nozzles

Bronze construction, accurately machined, easily taken apart. Produce fine mist readily absorbed by the air. Easily attached to city water supply. Average room requirement one nozzle to every 150 sq ft floor space. **Marley Form No. 188.**

Binks Manufacturing Co.

Plant and Executive Offices

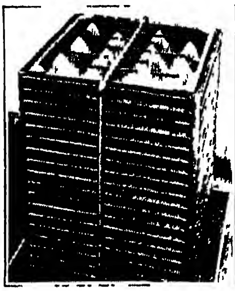
3114-3140 Carroll Avenue, **Chicago, Ill.**

Branch Offices

Cleveland Ohio	1157 Leader Bldg	New York N Y	[illegible] Lafayette St
Detroit, Mich	2832 E Grand Blvd	Philadelphia Pa	[illegible] Lombard St
Los Angeles, Calif	2111 East 26th St	Pittsburgh Pa	906 Chamber of Commerce Bldg
Milwaukee Wis	713 N Fourth St	San Francisco Calif	[illegible] Harrison St
New Orleans La	329 Balter Bldg	Windsor Ontario, Canada	

Representatives in Principal Cities

COOLING TOWERS SPRAY NOZZLES COOLING PONDS
SPRAY PAINTING EQUIPMENT



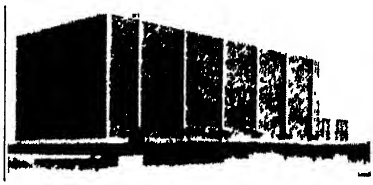
"Binks" Atmospheric Spray Cooling Towers

"Binks" Atmospheric Spray Cooling Towers are made in a wide range of sizes to handle capacities from 5 to 1200 gpm. They are used and recommended for all types of industrial water cooling work and are suitable in the standard sizes for refrigeration plants ranging from 2 to 240 tons and for Diesel engine plants ranging from 30 to 3600 hp.

The towers consist of a heavy shop welded copper bearing steel frame, genuine wrought iron manifold with welded feeder arms and bronze nozzles. All metal parts are hot dip galvanized after fabrication.

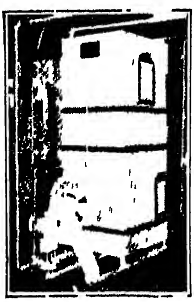
Louvres consist of clear all heart redwood or galvanized steel, to suit requirements.

Due to standardized construction features, "Binks" Atmospheric Spray Towers may be quickly erected on the job, for either ground or roof installation from a simple elevation print furnished with each unit, without skilled factory supervision.



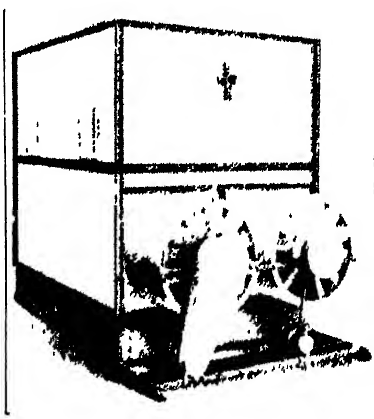
All bolts and nuts for assembling are cadmium plated, louvres being inserted to the tower frame in slip fit louvre retainer channels requiring the use of no bolts, nuts or any other fastening.

There are more than 2000 "Binks" Atmospheric Towers now in operation.



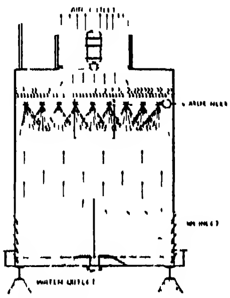
"Binks" Indoor Forced Draft Cooling Towers

"Binks" Indoor Forced Draft Cooling Towers are made in standard sizes from 5 gpm up to and including 300 gpm. They are extensively used for the cooling of jacket water for Diesel engines and large capacity air compressors, for condensing water for refrigeration machines and various industrial processes.



This type of equipment is exceptionally well adapted in locations where outdoor mounting of towers on building roofs would be extremely costly due to piping the installation to and from the tower. These units may be conveniently placed adjacent to the process with provision being made for fresh air inlet to the tower, such as open doors or windows, whereas the saturated air is conveyed to the outdoors through a duct.

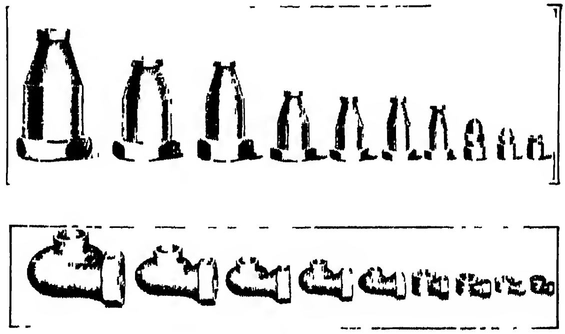
Write for Complete Catalog and Prices on any "Binks" Equipment in which you are interested.



Diagrammatic sketch of Binks Type "K" Tower

SPRAY-RITE

ROTOJET

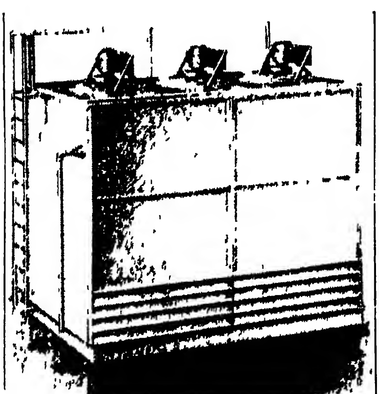


"Binks" Spray Nozzles
Made in all types for every cooling and industrial need

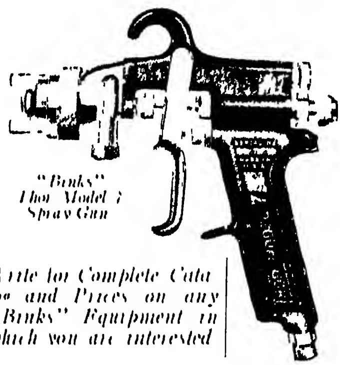
"Binks" Type "K" Induced Draft Towers

The "Binks" Type "K" Induced Draft Towers are designed for outdoor mounting either on ground or building roofs and are made in sizes from 25 to 800 gpm capacity. They are extensively used for all industrial water cooling requirements and standard sizes are suitable for refrigeration work ranging from 6 to 200 tons and for Diesel engine service from 90 to 3000 hp.

Fan assemblies are mounted to top of tower directly connected to geared head motors which are splash, moisture and weatherproof. Capacities of fans from 3200 to 96,000 cfm, depending on tower size, and are extremely quiet in operation, drawing air upward through tower from open louvre sections at bottom. This principle of mechanical draft tower construction eliminates wind noises in the casing and is representative of one of the most efficient mechanical draft towers made. They are constructed entirely of galvanized steel of heavy gauge, complete with ¼-in pans, and are shipped knocked down with complete assembly drawings for field installation which can be accomplished by ordinary labor.



Spray Painting Equipment



"Binks" Thor Model 7 Spray Gun

Write for Complete Catalog and Prices on any "Binks" Equipment in which you are interested

Consult Binks for any Spray Painting or Finishing Needs

Spray guns of all types

Compressed air supply systems

Compressed air cleaning systems

Spray booths and exhaust systems.

Water wash spray booths, for reclaiming porcelain enamel; also for synthetic enamel and other materials

Paint supply systems individual pressure tanks or pipe line circulating systems

Conveyor systems for handling items to be sprayed

Complete accessory equipment for the finishing department

E. B. Badger & Sons Co.

75 Pitts Street — **Boston, Mass.**

Engineers and Manufacturers

N Y Office, 271 Madison Avenue

Representatives in Principal Cities

PRODUCTS and SERVICES **Corrugated Copper and Stainless Steel Expansion Joints, Pipe Bends, Chemical Apparatus, Copper and Sheet Metal Work, Copper Boilers and Hot Water Tanks, Engineers on Process Work.**

BADGER DIRECTED FLEXING EXPANSION JOINTS

Since the Badger Corrugated Type of Expansion Joint was first placed on the market more than 40 years ago, several important improvements have been incorporated in this joint by E B Badger & Sons Co, Engineers, each adding to its life and utility

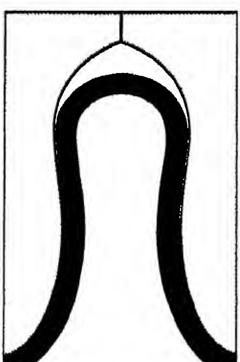
1. Adoption of equalizing rings
2. Machine method of making uniform corrugations
3. Use of seamless copper tubes
4. Improved machine method of manufacture resulting in minimum forming stresses
5. Monel sleeves for protection against superheated steam
6. Welding end joints
7. Adoption of scientific heat treatment
8. Use of special deoxidized copper
9. Directed Flexing
10. Use of Stainless Steel

Advantages of the Corrugated Joint

Eliminates packing, avoids steam losses No Servicing, Compact, Accessible, No Manholes needed

Directed Flexing

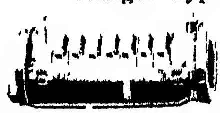
This feature, exclusive with joints made by E B Badger & Sons Co has brought about a marked increase in the life of the corrugated type of joint Corrugations are "all curved" and when flexing actually wrap and unwrap themselves in a progressively controlled movement The rings guide as well as limit Factory tests and actual service have proved thoroughly that the life of Directed Flexing Joints now is much longer

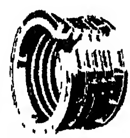


Different Types Available

Both flanged and welding end joints are available in 4 in size and larger In the flanged type for smaller sizes we can furnish a 4 in joint fitted with companion flanges, bolts and gaskets, for whatever size is required In the welding type we can furnish a 4 in joint fitted with swaged beveled nipples for whatever size is required All joints can be equipped with telescoping monel metal sleeves to protect against superheated steam

Flanged Types





Fitted with alignment bars on the 4 and 5 in sizes. Choice of standard or extra heavy flanges From 1 to 3 in expansion, inclusive

On 6 in. pipe and larger, no alignment bar is used Standard 125 lb or extra heavy 250 lb flanges as required. From 1 to 4 in expansion, inclusive

Welding Types

For use with both saturated and superheated steam Single type from 4 in. up, to take care of 1 in expansion or more Pipe nipples welded directly into pipe line.

Double units for 2 in expansion or more Equipped with service outlets if desired Complete units furnished, mounted on base plate, with anchor and guides

Single and Multiple Corrugated Expansion Joints for Low Pressure

For use between turbine or engine exhausts and condenser or on low pressure lines. Excellent for absorbing shock and vibration. Flanges in round, oval or rectangular shapes Guaranteed up to 30 lb pressure

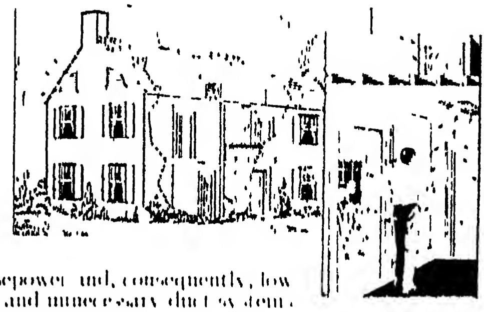
American Coolair Corporation

COOLING AND VENTILATING FAN SYSTEMS

3601 Mayflower Street, Jacksonville, Florida

PRODUCTS

Silent Reversible Belt Drive Cooling and Ventilating Fans for homes, stores, offices, factories, etc.
Quiet, high speed direct drive fans

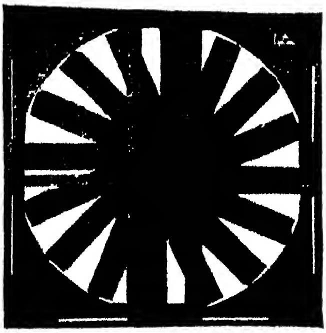


[illegible]

DIFFERENTIATE clearly between cooling and ventilating. Cooling requires complete air changes as rapidly as every half minute to every minute and one half; ventilation requires air changes only every two to five minutes, depending on the building. Coolair has pioneered home cooling by air circulation.

One of the outstanding advantages of COOLAIR ventilating fans is that they will deliver large quantities of air under free delivery with small horsepower and, consequently, low operating cost. Elimination of costly and unnecessary duct systems will often more than pay for the COOLAIR equipment. It is important to provide ample exhaust and intake opening to avoid restricting the air volume. Net free air passage area should never be less than area of fan. For maximum quietness free area should be enough larger than fan to assure a grill velocity of not more than 750 fpm.

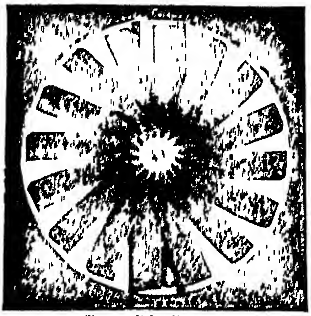
Description



Standard Type. Front View
Diameters Six to Nine Feet
Performance Data on Request

COOLAIR Belt Drive Fans meet the ventilating and cooling requirements of any size building. They are just as efficient when blowing in as when exhausting. The V belt drive permits use of standard electric motors operating at the most efficient speed, and easily replaced in event of motor trouble. Slower speed motors are costly, difficult to replace and less efficient. With simple exchange of a motor the COOLAIR unit can be converted from a silent, slowly turning residence unit using a small motor, to a high speed industrial unit using a large motor.

With a reversible motor, air can be blown into or exhausted from the building at will. Air can be pulled from the cool side of the building in the morning and the fan reversed when the sun moves to the opposite side in the afternoon.



Type "B" Front View

Over-All Dimensions (In.)

Size	Height and Width	Depth
2B	30 5/8	11 1/2
2½B	36 5/8	11 1/2
3B	42 5/8	12 1/2
3½B	49	15
4B	55 1/8	15 1/2
4½B	61 1/8	16
5B	67 1/8	16 1/2
*6	75 1/4	28
*7	87 1/2	31 1/2
*8	99 1/2	34
*9	112	34

*Standard type frame
Note Depth Approximate

Capacities and Weights Coolair Belt Drives

Fan Size	Hp	Watts per Hour	Fan RPM	Cu Ft Air per Min
2B	*1/6	190	427	3250
	†1/4	280	582	4450
2½B	*1/4	300	413	5750
	†1/3	420	460	6400
	‡1/2	570	507	7000
3B	*1/4	300	305	7800
	†1/3	420	340	8700
	†1/2	570	385	9800
	‡3/4	800	448	11000
3½B	*1/3	420	268	11200
	†1/2	570	290	12100
	†3/4	800	350	14600
	‡1	1060	392	16300
4B	*1/2	570	259	15800
	†3/4	800	277	17000
	‡1	1085	306	18800
	‡1½	1450	369	22500
4½B	*1/2	570	194	17000
	†3/4	800	225	19700
	†1	1085	260	22700
	‡1½	1450	286	25000
	‡2	1900	329	28000
5B	3/4	800	195	23000
	†1	1085	211	25000
	†1½	1450	251	29600
	‡2	1900	277	32800
	‡3	2850	324	38000

*Very quiet †Quiet ‡Industrial
Data in accordance with Standard Code Test American Society of Heating and Ventilating Engineers

Advantages

(1) Low initial cost Great air delivery per dollar of power

(2) Economical operation Use of a small motor at the most efficient speed

(3) Quiet performance Great volumes of gently moving air without objectionable noise or drafts

(4) May be equipped with reversible motors to exhaust or blow in at will

(5) Air volume controlled by means of two speed pulley on smaller units

(6) Rubber insulated motor support Adjustable to belt tension

(7) Equipped with SKF ball bearings in dustproof, grease packed housings

(8) Easy to install The steel blades and frame are durable, yet light in weight

(9) Standard motors Enclosed when specified. No vent holes are necessary to keep them cool

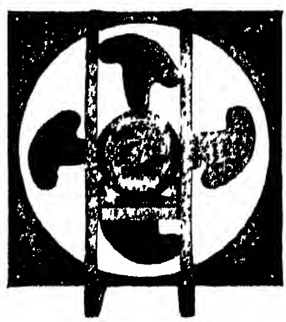
(10) Blades are individually mounted, inexpensive to replace

(11) Type "B" units, when equipped with ball bearing motors, are suitable for operation in any position

(12) Blades of Type "B" belt drive units are non overloading

(13) Residence units are light, spring suspended for ultra quiet operation

Coolair Direct Drive Fans



Coolair Direct Drive
Data on Request

Covered by U S Patent No 1,855,660, designed for maximum air per horsepower with minimum noise

Type "B" Direct Drive COOLAIR Fans are made in four sizes At 1200 rpm for residence kitchens, small stores, etc At 1800 rpm for cooling, drying, for removing dust and fumes Motors totally enclosed, long service types The 1/6 hp and larger are ball bearing, will operate in extremes of position, temperature and humidity. Cushion mountings

Coolair Residence Unit and Spring Suspension covered by U S Patent No. 1,992,112

DeBothezat Division

American Machine and Metals, Inc.

Executive and Sales Offices 100 Sixth Ave , New York, N. Y.

Factory EAST MOLINE ILL

Branch Offices and Representatives in All Principal Cities

Type H of the "Selective Series" 30 in , 3 hp 1140 r p m 8800 c f m against 1 in static pressure

Fans and Blowers are guaranteed to have non overloading power characteristics Complete operating safety under varying working conditions is assured

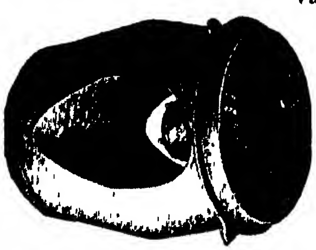
Disc Pressure Fans "Selective Series" Meet all pressure requirements efficiently made in various sizes from 8 in to 10 ft Bulletin SS 101 contains complete technical data on L-Type for low pressures and HI and H Types for high pressures

Giant Fans Are made in 5 ft to 10 ft diameters and are directly connected to motor, chain or Tex rope drive Technical data concerning these fans is contained in Bulletin SS 101

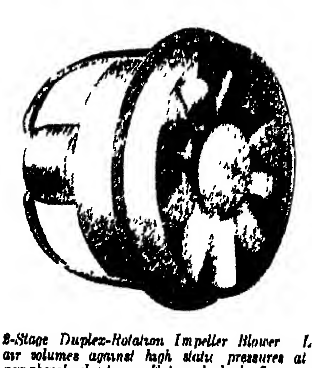
Bifurcators Can be placed in a straight duct without introducing right angle turns in the ducts or requiring extended shafts and bearings. As motor is located outside of air stream, safe, care free operation is assured

Vari-Speed Fans Speed selection is secured through variable pitch motor pulley Quiet Compact Reliable

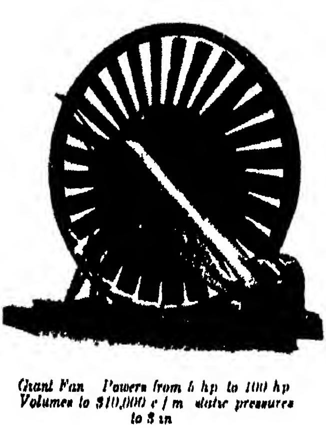
Duplex-Rotation Impeller Blowers Consist of disc pressure fan wheels rotating in opposite senses so designed that the rotational losses are completely eliminated The unit is directly driven by a DUPLEX-ROTATION motor, which eliminates belts and gears It is equipped with ball bearings throughout and is suitable for horizontal or vertical mounting This unit can replace any fan of same diameter now producing objectionable noise, without alterations Impeller Blowers are also supplied with belt or gear drives Write for Bulletin DR 101



Bifurcator For special ventilating jobs where fumes of excessive temperature corrosive or explosive character are to be removed



2-Stage Duplex-Rotation Impeller Blower Large air volumes against high static pressures at low peripheral velocities Extremely high efficiency over entire range of operation



Giant Fan Powers from 5 hp to 100 hp Volumes to 310,000 c f m static pressures to 3 in

Autovent Fan & Blower Company

1809-23 N Kostner Ave , **Chicago Illinois**

FANS BLOWERS **UNIT HEATERS**

Autovent "31" Series Propeller Fans - Will not churn air or overload the motor Especially recommended for economical ventilation Ruggedly constructed Capacities from 500 to 38,000 cfm Write for bulletin No 200

Acid-Moisture Proof, Explosion Proof Fans Wherever acid fumes or excess moisture laden air exists, the Acid Moisture Proof Propeller Fans equipped with fully enclosed motors and fan wheels with Bakelite coating (or of special construction) are recommended Wherever explosive gases and chemical fumes must be removed, Vapor Explosion Proof Propeller Fans (underwriters label "Class I group D") with fan wheels of copper, brass or other non ferrous metals must be installed Write for bulletin No 201

AUTOVENT "31 SERIES" PROPELLER FANS

Constant or Two Speed Alternating Current Multiphase
220 or 440 Volts 60 Cycle

Size Type	Motor Hp	Cfm Max	Appr Rpm Max	Appr Shpg Wt Lb	Size Type	Motor Hp	Cfm Max	Appr Rpm Max	Appr Shpg Wt Lb
1631MN	1 8	1700	1140	64	3031MN	1/3	7600	1140	220
1631MR	1 5	1950	1725	66	3631MI	1/2	10000	690	335
1831MN	1 8	2350	1140	77	36M31L	3/4	12500	850	390
1831MR	1 4	2950	1725	90	4231MI	1 0	13500	690	510
2031MN	1 4	3450	1140	115	4831MI	1½	18900	690	650
20M31R	1 4	3550	1725	115	5431MI	2	21500	575	890
24M31I	1 6	3850	850	145	6031MI	2½	25600	480	1200
2431MN	1 3	4300	1140	150	7231MI	3	38000	480	1500
3031MI	1 4	6000	850	210					

"BW" (Bucket Wheel) Propeller Fans A slow speed operating fan Provides efficient ventilation for unlimited uses, from 500 to 10,000 cfm Sturdily constructed, minimum power, available at low, quiet operating speeds Write for bulletin No 202

Autovent Uniblade Volume Blowers Motor driven universal discharge, for fume hoods, chemical labs, processing, drying, forced draft, etc Handles low volumes of air at medium pressures Wheels range from 6 in to 11 in dia same design as heavy duty blowers Can be mounted on floor, wall or ceiling Direct Connected Blowers for general ventilating applications, available in wheel diameters up to 25 in Bulletin No. 300 Forward Curve Belt Driven Blowers No 301

Autovent "V" Belt Driven Unit Blowers Forwardly curved blade type with smooth air flow and quiet bearings Motor mounted on steel pedestal, integral with blower housing, making a compact unit Air delivery can be decreased or increased if desired Interchangeable motors Sturdily constructed of sheet steel, rolled lock seams Write for bulletin No 300. Backward Curve Belt Driven Blowers No 302

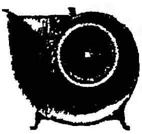
Autovent Super-Type Steam Unit Heaters This suspended type heater forces air circulation and directs warm air to lower part of room Heating element of flat copper fins, attached to seamless, drawn copper tubes Tubes run vertically for perfect drainage No welded or brazed joints in coil or header Fans have non-overloading power feature Motor furnished to requirement Write for bulletin No. 101

Bayley Blower Company

1817 S. Sixty-Sixth Street **Branches in Principal Cities** **Milwaukee, Wis.**

Builders of Heating, Ventilating, Cooling, Purifying, Humidifying and Air Washing Equipment, Exhaust and Drying Apparatus, Mechanical Draft and Blast, Fans and Blowers of all Types

Bayley Plexiform Fan:



Is a multi-blade fan for supplying air for heating and ventilating systems, manufacturing processes, drying systems, forced and induced draft systems It is suitable for handling high or low temperature gases at medium or low pressure Will deliver maximum quantities requiring minimum space with great economy

This is a distinct Bayley product, high class material and workmanship, properly designed to avoid excessive vibration and overstressing of parts Inlets and outlets are properly sized for maximum delivery and maximum efficiency Fans are furnished in single or double width of any required arrangement and with sleeve or anti-friction bearings

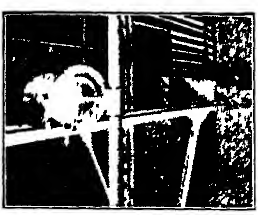
Aeroplex Fan:

Is of high speed design with self limiting power characteristics Application parallel to the Plexiform Fan Highly efficient and quiet in operation

Bayley Exhausters and Pressure Blowers:

Type "B" exhaust fan is for heavy duty, handling refuse from industrial and textile plants Type "SE" is used in handling smoke, fumes and dust-laden gases Type "H" for high-pressure work These units are highly efficient and of high class design and workmanship

Bayley Turbo Air Washers, Humidifiers and De-Humidifiers:



The Bayley Turbo Air Washer Showing Turbo Atomizer and Eliminator

The Turbo Atomizer used in the Bayley Washer produces a steady, fine spray Water at low pressure is delivered to the center of a rapidly revolving cone-shaped rotor provided with atomizing pins set in its periphery This atomizer requires very little attention, and will operate successfully under low water pressure The orifices are large and this atomizer, unlike high pressure nozzles, cannot clog

Bayley Chinook Heating Sections:

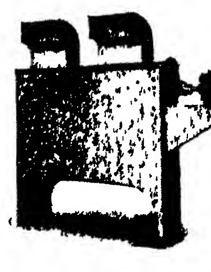
The Chinook section is used with blast heating, ventilating and drying systems, and is suitable for high or low pressure steam circulation The base is divided into two chambers Steam enters (see cut) the lower chamber, rising through ⅜-in pipes located within the 1¼-in pipes leading from the upper chamber Condensation takes place in the larger pipes, the water falling into the upper chamber and draining away through the return outlet The Chinook can be repaired in the middle of the bank without breaking steam connections or taking down a section

Shipped assembled in smaller sizes, and knocked down in the larger units May be installed in horizontal or vertical position

Bayley Chinookfin Heating Sections:

Are the same design as the Chinook Heaters, using heavy gauge copper fin tubes As compared with Chinook it is much lighter and occupies less space

Bayley Plexfin Unit Heaters:



This unit incorporates Chinookfin radiation and Plexiform or Aeroplex fans. The fan assembly including top plate and motor is removable as a unit for maintenance and inspection The heating element is a removable unit Casing all welded extra heavy gauge This is an exceptionally high grade unit at a moderate price

Buffalo Forge Company

450 Broadway, Buffalo, N. Y.

Branch Offices

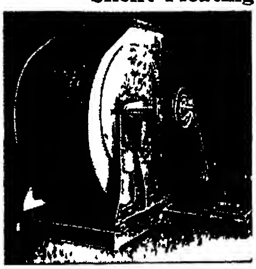
ALBANY 611 Standard Bldg	KITCHENER ONT Canadian Blower & Forge Co Ltd
ATLANTA 724 First National Bank Bldg	KNOXVILLE—C F [illegible] 515 Empire Bldg
BOSTON 486 Main St Melrose Sta	LOS ANGELES 708 Pershing Square Bldg
CHARLOTTE—J W Fraser & Co 124 Brevard Court	MINNEAPOLIS 519 Foshay Tower
CHICAGO 20 N Wacker Drive	NEW ORLEANS—Devlin Bros 1003 Maritime Bldg
CINCINNATI 622 Broadway	NEW YORK 39 Cortlandt St
CLEVELAND 418 Rockefeller Bldg	PHILADELPHIA 220 S Sixteenth St
DALLAS 515 South Harwood	PITTSBURGH 431 Fulton Bldg
DAVENPORT 505 Security Bldg	SAN FRANCISCO—Moore Machinery Co 550 Fifth St
DENVER—Hendrie & Bolthoff Mfg & Sup Co 1655 Seventeenth St	ST LOUIS 1596 Arcade Bldg
DES MOINES 214 Old Colony Bldg	SALT LAKE CITY Salt Lake Hdwe Co
DETROIT—Coon DeVisser Co 2051 W Lafayette Blvd	SEATTLE 500 First Ave , S
GREENVILLE—George R Morgan 316 Walker Bldg	TOLEDO 1922 Linwood Ave
KANSAS CITY 515 Dwight Bldg	TUCSON ARIZ Tidmarsh Engrg Co
	WASHINGTON D C 820 Woodward Bldg

PRODUCTS: Heating and Ventilating Equipment, including: Unit Heaters, Multiblade Fans, Pipe Coil Heaters, Buffalo Air Washers, Buffalo Unit Air Washers, Buffalo Unit-Coolers, Drying Equipment, Mechanical Draft Fans, Air Preheaters, Exhaust Fans, Blowers, Dust Collectors, Disc Fans, Spray Nozzles.

Buffalo Ventilating Fans

No matter what type of fan your work calls for, there is a Buffalo Fan of that type, of the right size, quiet, efficient, practical Write for our fan catalogs

"Limit-Load" Conoidal Fans with Silent Floating Bases

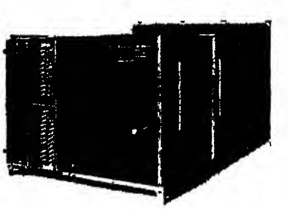


The Buffalo "Limit-Load," high-efficiency, non-overloading ventilating fan mounted on the silent, insulated floating fan base eliminates all motor and fan vibration, making an almost noiseless installation

Breezo Ventilating Fans

Breezo fans provide efficient, inexpensive ventilation on any job where they may be used to exhaust into the open. Made in sizes from 8 in to 36 in diameter.

Buffalo Comfort Conditioning Cabinets



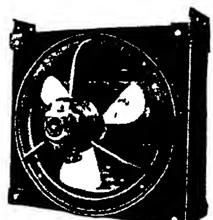
–for cooling and heating Heating coils are two-row copper fin type capable of heating from 70°F to 135°F with 2 lb steam Cooling capacities—from 3 tons up

Buffalo Air Washers

Buffalo Air Washers are in use in thousands of buildings, many for more than thirty years Bulletin 480 gives details

Buffalo Unit Heaters

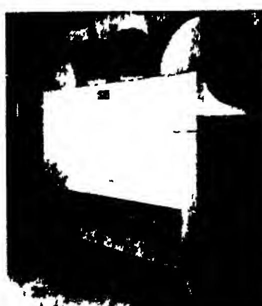
Gas Units: Made in suspended and floor types All models equipped with full automatic safety features Provide clean, silent heat at moderate cost



Breezo-Fin Steam Unit

Steam Units: Both suspended and floor type units in a large range of capacities

Buffalo Unit Coolers



Suspended Type: Quiet unit coolers for use with cold water, brine, methyl chloride or Freon Compact, simple, inexpensive

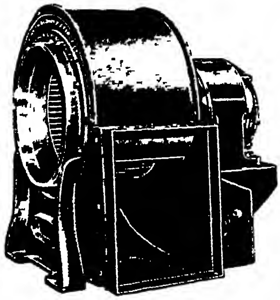
Floor and Flat Suspended: Available for same kinds of refrigerants, but with larger capacities

Champion Blower & Forge Co.

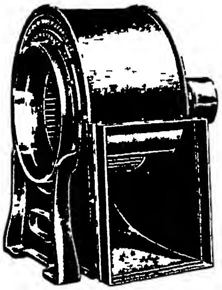
Manufacturers and Engineers

Plant and General Offices· Lancaster, Pa.

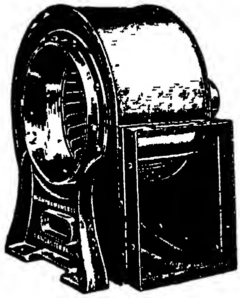
Manufacturers of Blowers, Ventilating Fans, and Exhaust Fans for Air and Material; and Blast Gates.



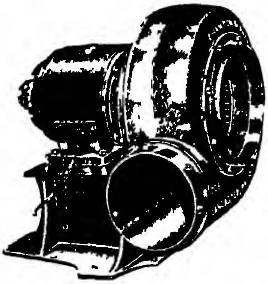
Type "SE"
Electric Driven Fans
Sizes—6 to 60 in Wheels



Type "S" Manivane
Belt Driven Fans
Sizes—6 to 60 in Wheels



Type "BC"
Backward Curve Fan
Sizes—12 to 60 in Wheels

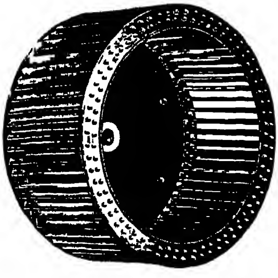


Type "CE" Manivane Fans
For Direct Work and
Quiet Operation

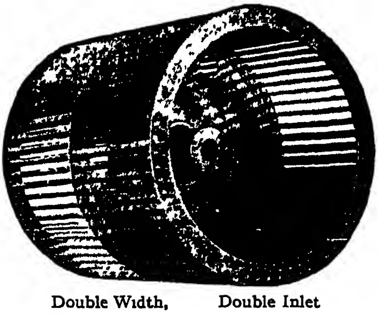
Champion Type "SE" Fans are arranged with direct connected motor mounted on convenient size motor base, adjustable to fit center distance of motor to which fan is directly attached In all other details constructed like Type "S" fans

Champion Type "S" Manivane Steel Plate Fans are specially adaptable for supplying air or for exhausting purposes handling air, smoke gas or fine dust directly through the fan or for heating, cooling, ventilating or drying purposes

The Backward Curved Blade Wheel on Type "BC" Fans allows a speed which permits the use of direct-connected standard speed Electric Motors



Single Width, Single Inlet.



Double Width, Double Inlet

Type "C" Manivane Blast Wheels

Built strong and substantial to withstand air resistance. Free from vibration and noise. Especially adapted for oil burner, stoker, general heating and ventilating purposes where noise is objectionable

ILG Electric Ventilating Company

Propeller Fans, Blowers, Unit Heaters, Air Conditioning Equipment

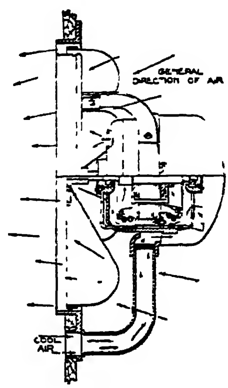
2880 North Crawford Avenue, **Chicago, Ill.**

Sales Representatives in all Principal Cities

ILG SELF-COOLED MOTOR PROPELLER FANS

Used everywhere for removal of foul air, fumes, heat, etc Features include the patented Ilg self-cooled motor dynamically balanced, vibration-free, bucket-type wheel, and a strong one-nameplate responsibility Motor, wheel and frame are all Ilg-built

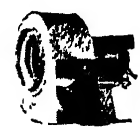
The self-cooled motor design combines the low operating cost of the open motor with the protection of the fully enclosed motor The fan action draws clean air through vent pipe from outside, circulates it through motor (follow the arrows) and exhausts it The Ilg motor stays clean, cool Ilg fan sizes range from 12 to 72 in in A C and D C Ratings are in accordance with the Standard Test Code of the A S H V E and the National Association of Fan Manufacturers



The Motor that Breathes

OTHER ILG PRODUCTS

Ilg Universal Multiblade Blowers—Type B Universal Blowers combine compactness, quietness and efficiency Motor is recessed in side of blower, requiring no separate base Multiblade wheel is mounted on motor shaft There is no inlet bearing Available also for belt drive Capacities 1750 cfm to 70,000 cfm, single and double width

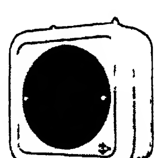


Ilg Type "B" Volume Blowers—A new design in small volume, low pressure blowers Light weight, quiet running Dynamically balanced multiblade wheel is mounted on motor shaft Motor and steel housing are supported by cast-iron base Universal discharge Available in 11 capacities, 180 cfm to 2100 cfm

Ilg Type "BC" Universal Blower—Motor load remains constant over unusually large variation in air volume and static pressure Type "BC" comes in direct connected drive (*left*), also belted drive with ball bearings (*right*) Blower wheel has backward curved blades riveted to side and back plates Universal discharge Available in 11 sizes

Ilg Unit Heaters—Steam and Electric—Copper tube and fin construction and enclosed self-cooled motor Ilg-built throughout For steam or hot water Tested with 500 lb hydrostatic pressure Available in 67 capacities, also, Ilg electric unit heaters for all electric operation, available in 20 sizes

Ilg Cooling and Air-Conditioning Units—Self-contained Ilg Spot Koolers in ½ ton cooling capacities with water or air-cooled compressors Also central unit systems using floor cabinet or ceiling suspension units with remotely located compressor For cooling, dehumidifying, and recirculating Also, for heating and humidifying



Albany, N Y
Atlanta, Ga
Baltimore Md
Boston, Mass
Buffalo, N Y
Camden N J
Chicago, Ill
Cincinnati O
Cleveland O
Detroit, Mich
Greensboro, N C
Hartford, Conn
Indianapolis, Ind
Kansas City Mo
Los Angeles, Cal
Milwaukee Wis
Minneapolis, Minn
Montreal, P Q

B. F. Sturtevant Co.

Hyde Park, Boston, Mass.

Sturtevant

REG U PAT OFF

PLANTS LOCATED IN

CAMDEN, N J — HYDE PARK, MASS — FRAMINGHAM, MASS
STURTEVANT, WIS — GALT ONT — BERKELEY, CALIF

Newark, N J
New York, N Y
Pittsburgh, Pa
Portland, Me
Portland, Ore
St Louis, Mo
San Francisco, Cal
Seattle, Wash
Spokane, Wash
Syracuse, N Y
Salt Lake City, U
Toronto, Ont
Washington, D C

A M Lockett & Co
New Orleans, La
Houston, Texas
Dallas, Texas

The Cooling and Air Conditioning Corporation

DIVISION OF B F STURTEVANT COMPANY

HYDE PARK BOSTON, MASS

ATLANTA CAMDEN CHICAGO LOS ANGELES NEW YORK

DATA ON HEATING, VENTILATING, AIR CONDITIONING AND VACUUM CLEANING EQUIPMENT FOR ARCHITECTS, ENGINEERS, CONTRACTORS

The publications listed below have been prepared to aid the architect, engineer and contractor in the selection of proper equipment for industrial, public, and private buildings of all types and sizes If you do not have all of these publications in your file we will gladly send copies upon request

COOPERATION

Sturtevant Engineers, located at each of the offices listed are always ready to cooperate with architects, engineers and contractors in the selection of equipment suitable for any prospective installation

VENTILATING FANS

CATALOGUE No

- 271-2 Multivane Fans (Forwardly curved blade type)
- 381-2 Silentvane Fans (Backwardly curved blade type)
- 414 Rexvane Fans (Radial blade type).
- 332-1 Ventilating Sets (direct motor-driven centrifugal type fans for ventilating small rooms Capacities: 80 to 1600 c f m)
- 400-5 Direct-connected Fans and Blowers (Propeller Fans, Window Fans for kitchens and offices, Centrifugal Fans from 80 to 6460 c f m.; Portable Gas-Engine-Driven Fans, Coal Burning Blowers, Forge Blowers, Dust Blowers)
- 422 Roofvane Ventilators

CATALOGUE No

- 345 Carbon Monoxide Asphyxiation and Its Prevention

MISCELLANEOUS HEATING AND VENTILATING EQUIPMENT

CATALOGUE No

- 377-1 Unit Ventilators
- 395-2 Rexvane Speed Heaters (Floor type unit heaters)
- 396-3 Speed Heaters (Suspended Type Unit Heaters)
- 382 Coal Burning Blowers

AIR CONDITIONING EQUIPMENT

CATALOGUE No

- 295-1 Air Washers
- AC 101 Industrial Air Conditioning
- 398 Comfort Air Conditioning
- 389 Household Humidifiers
- 378 Filticooler (Compact, high efficiency air washer For filtering, washing, humidifying, cooling, dehumidifying. Used principally for public buildings and factories).
- 383 Humidifilter (A humidifying and filtering unit primarily adapted to central systems Used for both comfort and industrial processing work).
- 384 Suspended Type Air Conditioning Units
- 401-1 Railway Air Conditioning

VACUUM CLEANERS

CATALOGUE No

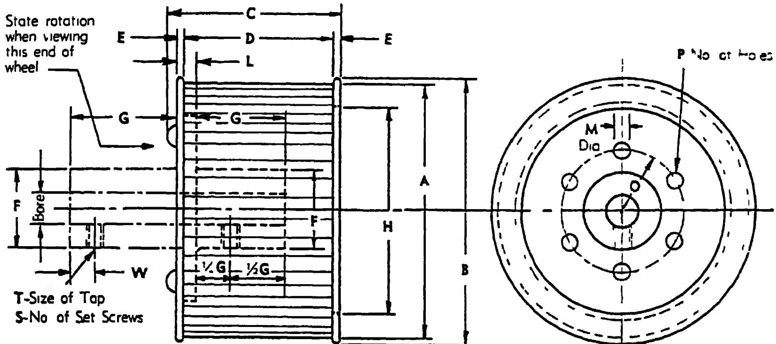
- 397-1, 368-1 Central System Vacuum Cleaners
- 413 "Vortex" Portable Vacuum Cleaners

The Torrington Mfg. Co.

50 Franklin Street, **Torrington, Conn.**

PRODUCTS—All Aluminum Blower Wheels and Disc Propeller Type Fans

SINGLE WIDTH SINGLE INLET BLOWER WHEELS



Pat 1,700,017
1-22-29

Hub can be furnished for inside or outside of wheel

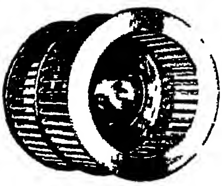
No	Nominal Size Dia	Width	No of Blades	Actual Blade Dia A	Actual O D B	C	Actual Blade Width D	E	F	G	H	L	Bore	O	M	P	T	S	W	Weight With Hub
00A	3	x 1 1/4	27	3	3 1/4	1 3/8	1 3/32	3/32	3/4	1/2	2 5/16	7/32	1/4	19/32	1/8	4	10-32	1	1/4	3 oz
00	3	x 1 1/2	27	3	3 1/4	1 23/32	1 7/16	3/32	3/4	1/2	2 5/16	7/32	1/4	19/32	1/8	4	10-32	1	1/4	3 1/2 oz
1/8	3 5/8	x 1 1/2	45	3 5/8	3 3/4	1 47/64	1 31/64	5/64	7/8	5/8	2 61/64	3/16	1/4	Spun-Hub			10-32	1	1/4	4 1/2 oz
0	4 1/2	x 2 1/4	34	4 1/2	4 7/8	2 9/16	2 7/32	1/8	1 1/8	3/4	3 1/2	15/64	3/8-1/2	1 1/16	3/16	6	1/4-20	1	1/4	8 1/2 oz
0A	4 1/2	x 2 1/4	45	4 1/2	4 7/8	2 9/16	2 7/32	1/8	1 1/8	3/4	3 1/2	15/64	3/8-1/2	1 1/16	3/16	6	1/4-20	1	1/4	9 1/2 oz
0B	4 1/2	x 3	45	4 1/2	4 7/8	3 3/32	2 3/4	1/8	1 1/8	3/4	3 1/2	15/64	3/8-1/2	1 1/16	3/16	6	1/4-20	1	1/4	11 1/2 oz
1/2B	5	x 2 1/4	45	5	5 11/32	2 9/16	2 1/4	1/8	1 1/8	3/4	3 7/8	15/64	1/2	1 1/8	3/16	6	1/4-20	1	5/16	13 oz
1/2A	5	x 2 3/4	45	5	5 3/8	3 1/8	2 3/4	1/8	1 1/8	3/4	3 7/8	15/64	1/2	1 1/8	3/16	6	1/4-20	1	5/16	12 1/2 oz
1/2	5	x 3	38	5	5 3/8	3 1/4	2 15/16	1/8	1 1/8	3/4	3 7/8	15/64	1/2	1 1/8	3/16	6	1/4-20	1	5/16	12 1/2 oz
1/2C	5	x 4	38	5	5 3/8	4 5/16	4	1/8	1 1/8	3/4	3 7/8	15/64	1/2	1 1/8	3/16	6	1/4-20	1	5/16	13 1/2 oz
1D	6	x 1	31	6	6 1/8	1 11/32	1	1/8	1 1/8	7/8	4 11/16	17/64	1/2	1 7/16	3/16	6	5/16-18	1	3/8	13 oz
1A	6	x 2	32	6	6 1/8	2 1/4	1 29/32	1/8	1 1/8	7/8	4 11/16	17/64	1/2	1 7/16	3/16	6	5/16-18	1	3/8	14 oz
1E	6	x 2	45	6	6 1/8	2 1/4	1 29/32	1/8	1 1/8	7/8	4 11/16	17/64	1/2	1 7/16	3/16	6	5/16-18	1	3/8	15 oz
1F	6	x 2 1/4	45	6	6 1/8	2 1/2	2 5/32	1/8	1 1/8	7/8	4 11/16	17/64	1/2	1 7/16	3/16	6	5/16-18	1	3/8	15 oz
1	6	x 3	31	6 1/16	6 1/8	3 1/4	2 29/32	1/8	1 1/8	7/8	4 11/16	17/64	1/2	1 7/16	3/16	6	5/16-18	1	3/8	15 1/2 oz
1C	6	x 3	45	6	6 1/8	3 1/4	2 29/32	1/8	1 1/8	7/8	4 11/16	17/64	1/2	1 7/16	3/16	6	5/16-18	1	3/8	15 1/2 oz
1B	6	x 3 7/8	38	6	6 1/8	3 31/32	3 5/8	1/8	1 1/8	7/8	4 11/16	17/64	1/2	1 7/16	3/16	6	5/16-18	1	3/8	1 # 1 1/2 oz
*1 1/4A	7 1/2	x 2	47	7 1/4	7 1/2	2 3/8	2	1/8	1 5/16	1 1/4	5 13/16	17/64	1/2	1 5/16	3/16	6	5/16-18	1	3/8	1 # 4 1/2 oz
*1 1/4B	7 1/2	x 3	37	7 1/4	7 1/2	3 1/4	2 7/8	1/8	1 5/16	1 1/4	5 13/16	17/64	1/2	1 5/16	3/16	6	5/16-18	1	3/8	1 # 4 3/4 oz
*1 1/4	7 1/2	x 3 7/8	47	7 1/4	7 1/2	4	3 5/8	1/8	1 5/16	1 1/4	5 13/16	17/64	1/2	1 5/16	3/16	6	5/16-18	1	3/8	1 # 8 1/2 oz
*1 1/4C	7 1/2	x 4 1/2	36	7 1/4	7 1/2	4 3/4	4 3/8	1/8	1 5/16	1 1/4	5 13/16	17/64	1/2	1 5/16	3/16	6	5/16-18	1	3/8	1 # 9 oz
1 1/2A	9	x 2	47	9	9 1/4	2 5/16	1 29/32	1/8	1 1/2	1 5/8	7 5/8	9/32	1/2-5/8	2 25/64	3/16	8	5/16-18	2	1/2	2 # 1 oz
1 1/2D	9	x 2 3/4	48	9	9 1/8	2 7/8	2 1/2	1/8	1 1/2	1 5/8	7 5/8	9/32	1/2-5/8	2 25/64	3/16	8	5/16-18	2	1/2	2 # 2 oz
1 1/2B	9	x 3	47	9	9 1/8	3 1/4	2 7/8	1/8	1 1/2	1 5/8	7 5/8	9/32	1/2-5/8	2 25/64	3/16	8	5/16-18	2	1/2	2 # 4 oz
1 1/2C	9	x 3 7/8	59	9	9 1/8	4	3 5/8	1/8	1 1/2	1 5/8	7 5/8	9/32	1/2-5/8	2 25/64	3/16	8	5/16-18	2	1/2	2 # 5 oz
1 1/2	9	x 4 1/2	47	9	9 1/8	4 27/32	4 15/32	1/8	1 1/2	1 5/8	7 5/8	9/32	1/2-5/8	2 25/64	3/16	8	5/16-18	2	1/2	2 # 8 oz
1 3/4	10 1/2	x 5 1/4	47	10 1/2	10 13/16	5 19/32	5 1/8	5/32	2	2	8 13/16	9/32	5/8-3/4	2 61/64	7/32	10	3/8-16	2	9/16	3 # 12 oz
2	12	x 6	64	11 15/16	12 7/16	6 3/8	5 7/8	3/16	2	2	10 1/8	25/64	3/4-1	3 21/32	7/32	10	3/8-16	2	3/4	5 # 4 oz
2 1/2	15	x 7 1/2	64	15 1/16	15 5/16	7 15/16	7 3/8	3/16	2 1/2	2 1/4	12 15/16	25/64	1	4 1/4	17/64	8	3/8-16	2	3/4	10 # 0 oz

Aluminum is light in weight—reduces starting torque and minimizes power consumption—reduces resonance—requires no protective finish—resists corrosion—is ideal for a smooth and quiet operating wheel

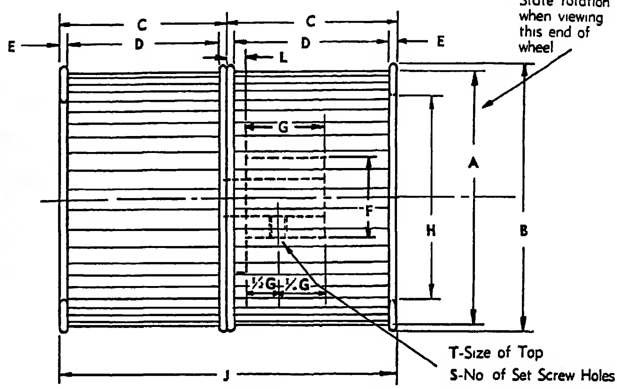
The Torrington Mfg. Co.

50 Franklin Street, **Torrington, Conn.**

DOUBLE WIDTH DOUBLE INLET BLOWER WHEELS



Pat 1,700,017
1-22-29



No	Nominal Size Dia x Width	No of Blades	Actual Blade Dia A	Actual O D B	C	D	E	F	G	H	J	L	Bore	T	S	Weight With Hub
00A	3 x 2½	27	3	3¼	1 3/32	1 3/32	3/32	¾	½	2 3/16	2 9/16	7/32	¼	10-32	1	4¾ oz
00	3 x 3	27	3	3¼	1 5/16	1 7/16	3/32	¾	½	2 5/16	3¼	7/32	¼	10-32	1	5¾ oz
⅛	3⅝ x 3	45	3⅜	3¾	1 11/16	1 31/64	5/64	⅞	⅝	2 41/64	3 31/64	5/16	¼	10-32	1	7¼ oz
0	4½ x 4½	34	4½	4⅞	2 15/32	2 7/32	⅛	1⅛	¾	3½	4 15/16	15/64	⅜-½	¼-20	1	14 oz
0A	4½ x 4½	45	4½	4⅞	2 15/32	2 7/32	⅛	1⅛	¾	3½	4 15/16	15/64	⅜-½	¼-20	1	1#
0B	4½ x 6	45	4½	4⅞	3	2¾	⅛	1⅛	¾	3½	6	15/64	⅜-½	¼-20	1	1# 3¼ oz
½B	5 x 4½	45	5	5 11/32	2½	2¼	⅛	1⅛	¾	3⅞	5	15/64	½	¼-20	1	1# 5 oz
½A	5 x 5½	45	5	5⅜	3 3/16	2¾	⅛	1⅛	¾	3⅞	6	15/64	½	¼-20	1	1# 5 oz
½	5 x 6	38	5	5⅜	3 3/16	2 15/16	⅛	1⅛	¾	3⅞	6⅜	15/64	½	¼-20	1	1# 5 oz
½C	5 x 8	38	5	5⅜	4¼	4	⅛	1⅛	¾	3⅞	8½	15/64	½	¼-20	1	1# 7¾ oz
1D	6 x 2	31	6	6⅛	1¼	1	⅛	1⅛	⅞	4 11/16	2½	17/64	½	5/16-18	1	1# 5½ oz
1A	6 x 4	32	6	6⅛	2 5/32	1 29/32	⅛	1⅛	⅞	4 11/16	4 5/16	17/64	½	5/16-18	1	1# 8 oz
1E	6 x 4	45	6	6⅛	2 5/32	1 29/32	⅛	1⅛	⅞	4 11/16	4 5/16	17/64	½	5/16-18	1	1# 7½ oz
1F	6 x 4½	45	6	6⅛	2 13/32	2 5/32	⅛	1⅛	⅞	4 11/16	4 13/16	17/64	½	5/16-18	1	1# 7½ oz
1	6 x 6	31	6 1/16	6⅛	3 5/32	2 29/32	⅛	1⅛	⅞	4 11/16	6 5/16	17/64	½	5/16-18	1	1#12 oz
1C	6 x 6	45	6	6⅛	3 5/32	2 29/32	⅛	1⅛	⅞	4 11/16	6 5/16	17/64	½	5/16-18	1	1#12 oz
1B	6 x 7¾	38	6	6⅛	3⅞	3⅝	⅛	1⅛	⅞	4 11/16	7¾	17/64	½	5/16-18	1	1#15 oz
*1¼A	7½ x 4	47	7⅛	7½	2¼	2	⅛	1 5/16	1¼	5 13/16	4½	17/64	½	5/16-18	1	2# 3 oz
*1¼B	7½ x 6	37	7⅛	7½	3⅛	2⅞	⅛	1 5/16	1¼	5 13/16	6¼	17/64	½	5/16-18	1	2# 4¼ oz
*1¼	7½ x 7¾	47	7⅛	7½	3⅞	3⅝	⅛	1 5/16	1¼	5 13/16	7¾	17/64	½	5/16-18	1	2#10 oz
*1¼C	7½ x 9	36	7⅛	7½	4⅝	4⅜	⅛	1 5/16	1¼	5 13/16	9¼	17/64	½	5/16-18	1	2#11½ oz
1½A	9 x 4	47	9	9⅛	2 5/32	1 29/32	⅛	1½	1⅝	7⅝	4 5/16	9/32	½-⅝	5/16-18	2	3# 4 oz
1½D	9 x 5½	48	9	9⅛	2¾	2½	⅛	1½	1⅝	7⅝	5½	9/32	½-⅝	5/16-18	2	3# 7 oz
1½B	9 x 6	47	9	9⅛	3⅛	2⅞	⅛	1½	1⅝	7⅝	6¼	9/32	½-⅝	5/16-18	2	3# 8½ oz
1½C	9 x 7¾	59	9	9⅛	3⅞	3⅝	⅛	1½	1⅝	7⅝	7¾	9/32	½-⅝	5/16-18	2	3#13½ oz
1½	9 x 9	47	9	9⅛	4 21/32	4 13/32	⅛	1½	1⅝	7⅝	9 7/16	9/32	½-⅝	5/16-18	2	4# 2 oz
1¾	10½ x 10½	47	10½	10 11/16	5 7/16	5⅛	5/32	2	2	8 11/16	10⅞	9/32	⅝-¾	⅜-16	2	5#15½ oz
2	12 x 12	64	11 15/16	12 7/16	6¼	5⅞	3/16	2	2	10⅛	12⅛	25/64	¾-1	⅜-16	2	8#11 oz
2½	15 x 15	64	15 1/16	15 5/16	7¾	7⅜	3/16	2½	2¼	12 15/16	15½	25/64	1	⅜-16	2	16# 8 oz

HOUSING SCROLLS. We do not manufacture housings but have prepared drawing with tables of dimensions for Scrolls for each size wheel Copy will be mailed upon request.

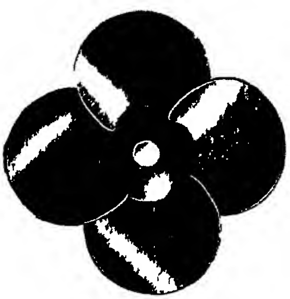
CAPACITY TABLES. Capacity rating and curve sheets showing static pressure, air deliveries and brake horsepower at various speeds will be sent upon request

The Torrington Mfg. Co.

50 Franklin Street, **Torrington, Conn.**

AIRISTOCRAT
SILENT FANS

Patent No 2 021 707



Deluxe Model—For desk, ceiling and wall bracket fans Built with round center disc in size 8 in —10 in —12 in —16 in

Standard Model—Same as Deluxe model but with conventional type spider for economy

Pressure Model—Same blade shape as Deluxe Model but designed for moderate pressures where quiet operation is desired Sizes 8 in —10 in —12 in —14 in —16 in

All models have 4-blades for clockwise rotation Hubs of brass, blades of steel or aluminum—any finish

VARIPITCH PRESSURE FANS

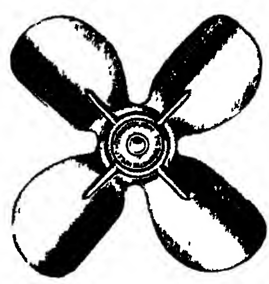
A newly patented variable pitch propeller fan blade will be marketed in 1937 Details furnished upon request

Prices and ratings for all fans furnished upon request

AUTOCRAT FANS

Torrington Autocrat fans for auto heaters and windshield defrosters have been the standard ever since these devices were first marketed Made in sizes 3 in — 4 in —4½ in —5 in —5¼ in —5½ in —6 in —6¼ in — 6½ in. All four blades Also 7 in 5-blade.

Made in one piece of cold rolled steel or aluminum with brass hubs, complete with set screw. ¼ in bore is standard Either clockwise or counter-clockwise rotation (expressed when looking at air delivery side of fan) White nickel is standard finish for steel blades



Send for Bulletin with prices, detailed specifications and ratings.

L. J. Wing Mfg. Co.

Branch Offices in Principal Cities

59 Seventh Avenue, New York, N. Y.
Phone CHelsea-3-0026

Factory NEWARK, N J

Wing Featherweight Unit Heaters—Floodlights of Heat

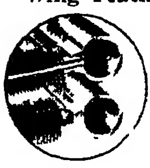
Vertical downward discharge of heat—the original and unique feature of Wing Featherweight Unit Heaters—allows distribution of heat in any direction or number of directions from a single heater Thus one heater effects distribution that could be accomplished only by a number of one-direction heaters

Heating expense is considerably reduced because this heater located close to the ceiling, takes warm air from the top of the building, where the heat would be wasted, and delivers it in the work area

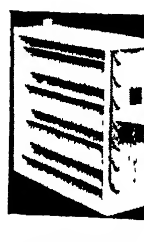
Wing Featherfin Heating Elements described elsewhere on this page, are employed in all Wing Unit Heaters

Made in various capacities and for low and high ceilings Discharge outlets to suit any condition *Bulletin H-6*

Wing Featherfin Heating Sections—For heating air for any purpose by steam or hot water



Detail of Wing Featherfin Heating Element showing Compression Union Tube Connection



Variable-Temperature Heating Section

The Wing Featherfin Heating Element is extremely light in weight It is of the fin-and-tube extended-surface type, and offers very low resistance to air flow Its hairpin shape avoids expansion and contraction strains Each tube is easily replaced in case of accidental damage

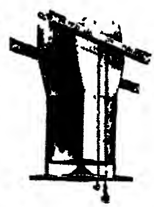
Sections available for any final air temperature with any steam pressure

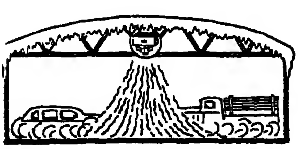
Used separately for any heating purpose and in Wing Unit Heaters Tested at 1000 lb pressure *Bulletin H-6*

WING VARIABLE-TEMPERATURE HEATING SECTIONS permit precise control of air temperature without by-passing and without throttling steam No freezing *Bulletin VT-2*

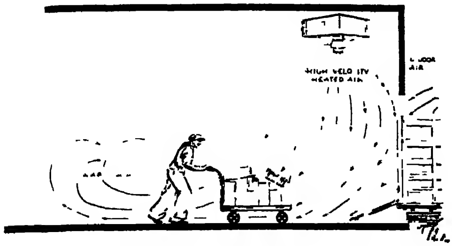
Wing Utility Heaters—This Wing Heater—a light-weight suspended unit heater—delivers heated air in one general direction Has same powerful fan and rugged heating element as Wing Featherweight Unit Heaters *Bulletin U-4*

Wing Industrial Fog Eliminators—Eliminate fog, odor and fumes in dyeing, bleaching and finishing plants in the textile industry, in creameries, pasteurizing, bottling and distributing plants in dairy industry, in canning, bottling and packing plants in the food industry, in chemical works, paper mills, steel pickling plants, etc No ducts are required *Bulletin FE-11*





Wing Garage Heaters—For effective and economical heating of garages Located above the open aisles, they deliver a column of heated air down to the floor where it spreads in every direction, putting the heat where it is needed Sometimes cut heating costs in half *Bulletin G-1*



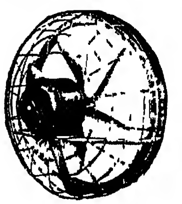
Wing Door Heaters—For instantaneously heating inrush of cold air at doorways of garages bus terminals freight houses, docks piers shipping and receiving departments

The unit is suspended from the ceiling over each opening A powerful fan drives down a column of heated air that warms the entering cold blast Connected to existing steam line *Bulletin D-1*

Wing Featherfin Process Heating Units—For manufacturing processes such as drying, aging, etc, requiring the recirculation of the heated air Motor or turbine located outside air current *Bulletin P-2*

Wing-Scruplex Safety Ventilating Fans—A propeller type fan that will deliver air against static pressure, quietly, efficiently and safely

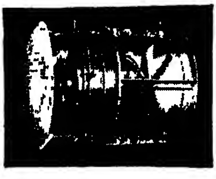
The screw design propeller moves the air forward in straight lines with minimum eddy, assuring high static efficiency Sizes 10 to 60 in, capacities range from 950 to 100,000 cfm *Bulletin F-6*



Solid Blade Type

Wing Type COM Blowers (High Static Pressures at Low Speeds)—Efficiently produce high static pressures at low speed Equipped with constant-speed motor and built-in damper, permitting variation of air delivery over a wide range with decreasing horsepower

For hand-fired, stoker-fired, oil-burning or pulverized-fuel boilers Frequently eliminate all duct work *Bulletin CO-7*

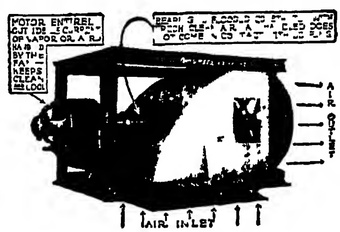


Wing Steam Turbines for driving pumps, fans, etc *Bulletin S-9*

Wing Kildraft—Warms air entering doors of stores, public buildings, etc *Bulletin KD-1*

Wing Safety Cooling Fans for cooling men and products in industry *Bulletin CF-12*

Wing - Scruplex Exhausters — Made for either horizontal or vertical operation—top, bottom or side intake The motor is entirely outside the exhaust housing, therefore always easily accessible—kept clean and cool For any ventilating need where air must be drawn or forced through ducts *Bulletin 78-A*



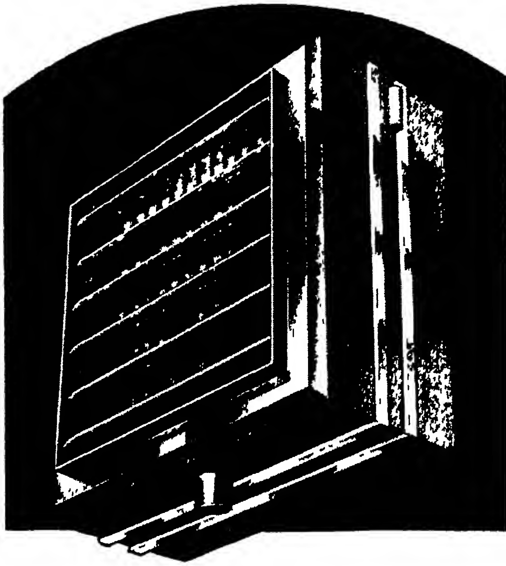
Wing Motor-Driven Blowers—For low-pressure heating boilers and small power boilers Increase boiler capacity, permit close control and allow use of lowest-cost fuel The type EM Blower is equipped with fully enclosed, dust-proof motor complete with speed-regulating rheostats and automatic controls *Bulletin M-66*

Wing Turbine-Driven Blowers—Increase boiler capacity, maintain constant steam pressure and permit complete combustion of low-cost fuels No duct work is necessary Applied to hand-fired, stoker-fired, oil burning or pulverized-fuel boilers The exhaust steam is free from oil and can be used for heating or processes *Bulletin T-97*

Airtherm Manufacturing Company

1474 South Vandeventer, **St. Louis, Mo.**

THE ENGINEERED LINE OF UNIT HEATERS



AIRTHERM IMPROVED UNIT HEATERS

Backed by the experience of thirty years in unit heater construction, the new Airtherm line represents a great advance in unit heater construction Many new and exclusive features have led to the instantaneous acceptance of this line by important industries

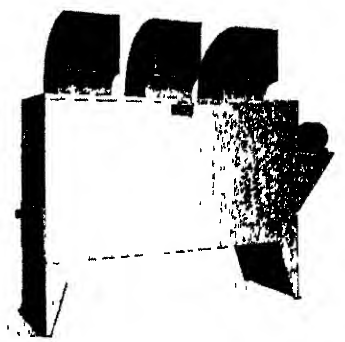
High quality materials, sound engineering principles, advanced construction methods, all combined with modern styling, have resulted in a product of unquestioned merit Study all unit heaters before you buy Your choice will be Airtherm

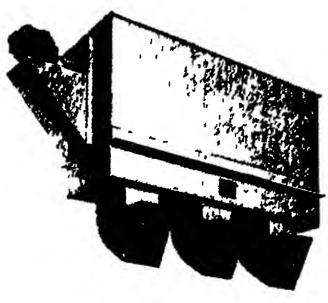
The Airvector *(above)*

Airtherm's latest contribution to the propeller fan type unit heater field Many exclusive features, including ribbed "Sound-Proofed" Cabinets, "No Pendulum Action" method of motor mounting, modern styling, and other exclusive features Complete range of sizes for any job

The Airheator *(right)*

Airtherm's blower fan type unit heater for floor or ceiling mounting Available in a variety of assemblies, welded construction, quiet operation, and high efficiency Airtherm Unit Heaters are the choice of leading engineers





The Airblanket *(left)*

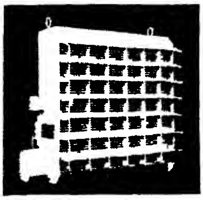
An outstanding innovation in unit heater construction and performance The Airblanket is now being used where ceilings run as high as 70 ft , and has shown amazing performance under every circumstance Investigate the Airblanket for any job over 18 ft ceiling, or any other installation requiring absolute control of warm air discharge Write for Catalog A-37.

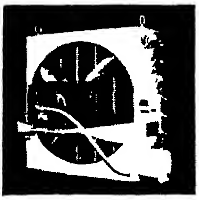
Electric Air Heater Company

Division of The American Foundry Equipment Co.

677 Byrkit Street. **Mishawaka, Indiana, U. S. A.**

Manufacturers of Industrial and Domestic Electric Air Heaters



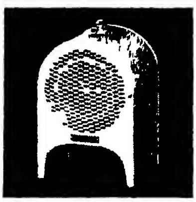


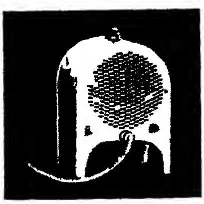
Industrial Heater

Industrial Electromode Heaters

This type electric air heater design includes all the principles of steam unit heaters The difference is that the source of heat supply is a set of wires instead of a Boiler House, piping and steam traps Controlled by thermostats, electric air heaters are economical, easily installed and heat is available within one minute at any time, and cut off instantly when no longer required

Model	Ratings			60 F Temp Rise		45 F Temp Rise		Fan Diam
	KW	BTU	EDR	CFM	FPM	CFM	FPM	
8-3	3 0	10245	42 7	155	515	207	690	6½
11-4½	4 5	15367	64 0	233	382	310	596	
11-6	6 0	20490	85 0	310	508	412	660	9
14-10	10 0	34150	142 0	517	488	684	645	12
14-15	15 0	51225	213 0	776	732	1034	975	
20-25	25 0	85375	356 0	1290	533	1720	712	18
20-35	35 0	119525	498 0	1800	750	2400	1060	
27-45	45 0	153675	640 0	2325	570	3066	748	24
27-60	60 0	204900	854 0	3100	760	4133	1000	
32-90	90 0	307350	1280 0	4656	727	6108	954	30
32-120	120 0	409800	1708 0	6208	968	8280	1294	





Portable Heater

Portable Electromode Heaters

This portable electric air heater meets the demand for portable heaters as well as for permanent installations This type may be equipped with a built-in thermostat for temperature control. A special switch can be added which will permit running the fan without heat A waterpan can be added so that 1 to 3 pints of water may be evaporated per hour to give humidity control

Model	KW	BTU	EDR	60 F Temp Rise		45 F Temp Rise		Face Area Sq Ft	Diam Fan	Net Wt
				CFM	FPM	CFM	FPM			
A-15	1 5	5122	21 3	75	220	103	310	0 327	7¾	20
A-20	2 0	6830	28 4	100	305	138	420			
B-30	3 0	10245	42 7	150	340	207	468	0 442	9	25
B-40	4 0	13660	56 9	200	450	276	625			
C-50	5 0	17075	71 1	260	435	345	575	0 60	11½	32
C-60	6 0	20490	85 3	310	515	415	692			

1-Kwhr = 3415 Btu = 14 23 sq ft Equiv in Dir Radiation 60 cycle A C is Standard—other cycles and D C at extra cost for motor *Important* Specify Voltage and kind of current when ordering

Bilt-In Heater

Bilt-In-Wall Type Electromodes

This unit is designed to fit the heating requirements of the home, apartment, office or summer (winter) home The grille is of an attractive design and may be finished with any color to harmonize with interior decoration scheme. This heater is installed on the standard spaced stud just above the baseboard and controlled by a thermostat it gives automatic controlled temperature in each or all rooms.

Send for Data Book No. 337

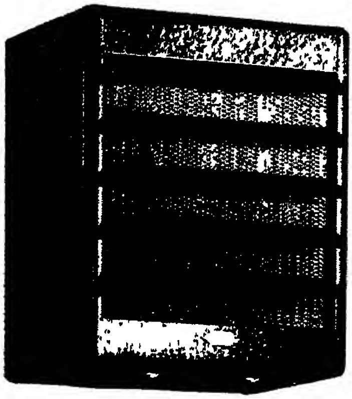
Fedders Manufacturing Co.

HEAT TRANSFER SPECIALISTS SINCE 1896

57 Tonawanda Street, **Buffalo, N. Y.**

Branches and Representatives in All Principal Cities

Unit Heaters, Air Conditioning Surface and Unit Conditioners, Unit Coolers, Commercial and Household Refrigeration Equipment



Patents 1 970,105 2,025 426

FEDDERS UNIT HEATERS

Quiet operation, high efficiency heating element with streamline tubes and specially designed individual convoluted fins, full protection against destructive expansion stresses within the heating element and between element and cabinet, the strength of curved non-diaphragmatic headers, extremely rugged electrically welded mono-piece steel cabinets are results of advanced engineering inherent in Fedders Unit Heaters

Their handsome appearance makes their high heating ability available to commercial as well as industrial applications

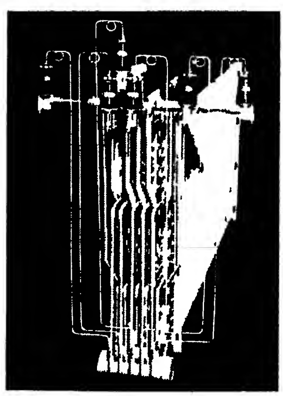
Fedders Unit Heaters are built in a complete line of sizes and capacities as listed below

Model No	3023	3141	3143	3181	3183	3211	3261	3273	3311	3372	3411
EDR	75	100	125	150	175	200	225	250	300	350	400

Model No	3421	3472	3615	3666	3671	3866	3871	3911	3961	3971
EDR	450	500	600	700	800	900	1000	1100	1200	1300

FEDDERS AIR CONDITIONING SURFACE

For refrigerant, cold water, steam or hot water Fedders design assures proper distribution throughout the coil and keeps pressure drop to a minimum by reducing the length of tubing in series Copper tubes and copper fins provide maximum heat transfer efficiency Catalogs contain full information and figuring data on the complete range of sizes and capacities



Patents
1,974,631, 1,987,948,
2,011,379

FEDDERS THERMOSTATIC EXPANSION VALVES

Multiple use of adjustable, Fedders High-Capacity Thermostatic Expansion Valves in conjunction with Fedders refrigerant manifolding, doubly assures correct distribution of refrigerant

COMPLETE WORKING DATA

Fedders catalogs are working data books giving complete specifications, dimensions and performance data Write for copies covering products that fit your requirements

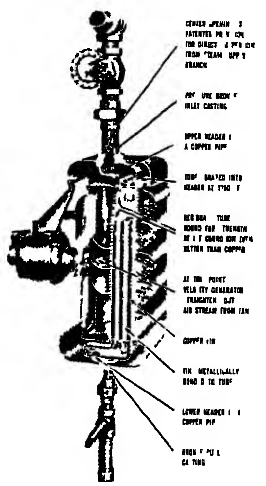
Modine Manufacturing Co.

17th and Holburn Streets, **Racine, Wisconsin**

Branches in All Principal Cities

MODINE UNIT HEATERS

Application—Not only successfully used for industrial, garage and similar space-heating applications, but ideally suited also to show-rooms, stores, oil stations, churches, school rooms, resi-



dential recreation rooms, etc They have also effected important economies in the drying of paint, leather, enamels, paper, wood, etc, by speeding up and bettering the drying process

Description—Exclusive features that not only make the Modine a better unit

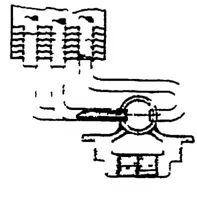
MODINE UNIT HEATERS CAPACITIES AND DIMENSIONS

(In Inches)

Model No	Over-all Height	Width	Depth Less Motor	EDR	CFM	Motor RPM
76	10½	9¾	5¾	76	187	1550
126	16	13	8	126	456	1590
152	18	15	8	152	540	1590
181	18	15	8	181	770	1140
204	18	15	8	204	735	1140
238	18	15	8	238	731	1140
275	19½	18	9	275	1052	1140
352	22½	18	9	352	1425	1140
440	22½	18	9	440	1320	1140
542	23	22	9	542	1710	1125
620	26½	22	9	620	2140	1120
710	26½	22	9	710	2230	1120
903	27⅞	26	10½	903	3000	1125
1163	30⅜	26	10½	1163	4050	1110
1300	34⅜	26⅜	8	1300	5010	1125
1545	30⅛	52½	11	1545	5400	1125
2015	30 5/32	56½	11	2015	6540	1110

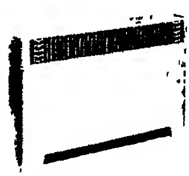
All above models are available with variable speed motors Units for hot water application also available

structurally but assure more effective economical distribution of heat (1) The **Expansion Bend,** given each tube of the condenser before entering lower header provides for free expansion, thus eliminating expansion strain from being transferred to header tanks (2) **Velocity Generator** for redirecting and controlling the heated air stream assures greater possible throw consistent with comfortable heating (3) **Direct Pipe Suspension** greatly facilitates horizontal redirection of the heated air stream, permitting full 360 deg rotatability Also means easier installation at less cost—no brackets, pipe rods or straps being necessary Write for Unit Heater **Catalog 236.**



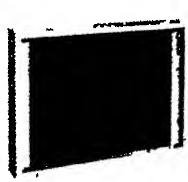
MODINE COPPER CONVECTORS

The new Modine Copper Convector furnished in four types Concealed, Recessed, Floor and Wall Cabinet, is designed for use on steam, vapor, vacuum or hot water systems Styled for beauty of line and proportion, the enclosure is adapted to easy color application Interchangeable, die-cast grille segments in four patterns make it possible to design a grille which will harmonize with any interior Grilles also available in plated finishes —**Catalog 136.**



MODINE COOLING COILS

Developed for use in connection with central system cooling and air conditioning plants, Modine Cooling Coils, Cold Water Type, are installed in conjunction with a blower fan and duct work They are adaptable where cold water or a non-corrosive brine is used as the cooling medium.—**Catalog 336.**



OTHER MODINE PRODUCTS

Modine also offers complete lines of Blast Heaters Unit Coolers and Air Conditioning Equipment Data will gladly be furnished on request

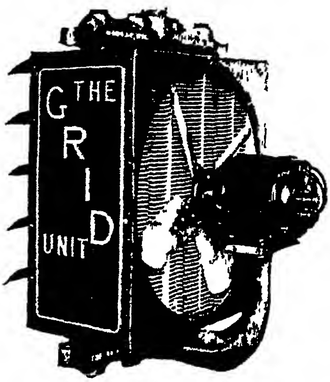
The Unit Heater and Cooler Co.

Wausau, Wisconsin

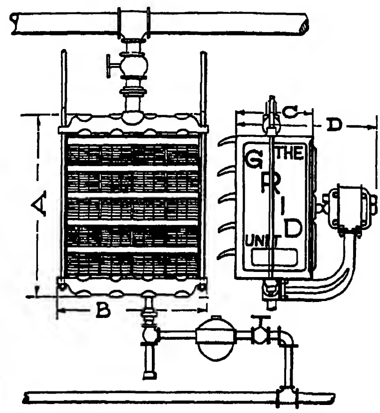
Offices in Principal Cities

MANUFACTURERS OF THE GRID UNIT

(PATENTED)



Sturdy—Permanent
The Grid Unit Heater
(A product of the D J Murray Mfg Co)



Typical Installation of Grid Unit Heater

GRID UNIT HEATER DATA

Model No	Dimensions Inches A	B	C	D	Face Area Sq Ft	Motor Hp	Motor RPM	Vol at Fan	Capacities 5 Lb Press 60° Air BTU	Final Temp	Approx Shipping Weight	Pipe Sizes Supply	Outlet
1000	16	12¾	9⅜	16	86	1/20 1/30	1750 1150	578 405	29,400 23,300	107 113	85	1¼"	1¼"
1200	18	14½	11¼	17½	1 04	1/20 1/30	1750 1150	711 515	46,000 35,000	119 122	120	1¼"	1¼"
515	22	18	11½	20	1 65	1/10 1/20	1750 1150	1290 1000	59,750 54,850	102 110	165	1½"	1¼"
1500	22	18	11½	20	1 65	1/10 1/20	1750 1150	1450 1015	77,500 65,500	109 118	210	1½"	1¼"
520	27	23¼	11½	21½	2 80	1/6 1/10	1150 850	2500 1910	102,500 91,100	97 103	290	2"	1¼"
2000	27	23¼	11½	21½	2 80	1/6 1/10	1150 850	2500 1835	148,000 113,000	114 117	320	2"	1¼"
525	32	28½	11½	28	4 50	1/2 1/4	1150 850	4200 3370	166,400 147,500	94 100	340	2"	1¼"
2500	32	28½	11½	28	4 50	1/2 1/4	1150 850	4200 3375	225,000 197,000	108 113	440	2"	1¼"
530	38	33	13¾	29	6 50	1/2	820	5300	260,500	105	560	2½"	1¼"
3000	38	33	13¾	29	6 50	1½ 1/2	1150 850	8100 6350	394,000 341,000	104 109	725	2½"	1¼"

Use Grid Unit Heaters—

For either low or high Steam Pressure
Cast Aluminum Fins—Long Life—No Leaks—Not effected by electrolytic action
More Air changes—Positive Circulation
Lower outlet temperatures and large Air Volume
Reduced Fuel Cost.
Low Maintenance Expense

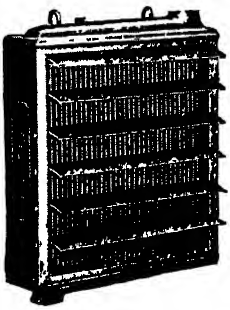
Send For Bulletin on Other Units Not Listed.

YOUNG RADIATOR Company

Racine, Wis.

Representatives in All Principal Cities

Manufacturers of Young Unit Heaters—Convection Heaters—Blast Units—Unit Coolers—Evaporators—Commercial Heat Transfer Units



Model "SH" Unit Heater

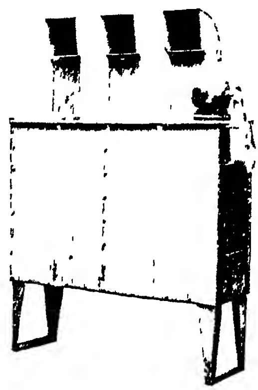
There are 24 different models of **Suspended Unit Heaters** equipped with constant or variable speed motors **Floor Type Unit Heaters** have capacities from 120,000 to 800,000 Btu per hour

Blast Units and **Commercial Heat Transfer Surfaces** may be used for various applications operating in conjunction with heating, cooling, or air conditioning systems

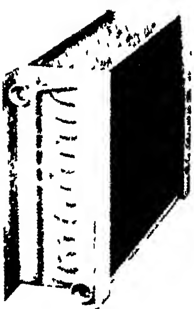
Unit Coolers are used with cold water, brine, or any common refrigerant **Water Coils** for cooling with water or brine in many sizes and capacities

Evaporators for use with Freon or Methyl chloride are scientifically designed to distribute the liquid with one or a group of expansion valves

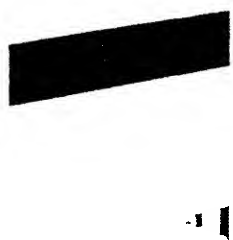
There are four distinct types of **Convection Heaters**—free standing cabinets, recess cabinets, wall-hung cabinets, and plastered-in enclosures. All models incorporate the **Streamaire** design with large elliptical tubes which make Young convectors outstanding on any type of heating system.



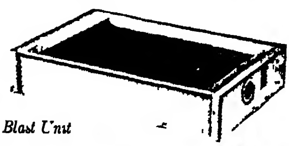
Model "FH" Unit Heater



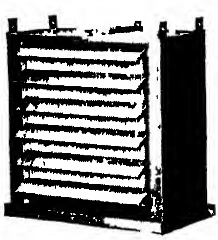
Water Coils



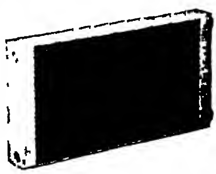
Streamaire Convection Heaters



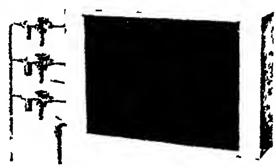
Blast Unit



Unit Coolers



Commercial Heat Transfer Unit



Evaporator

AEROFIN CORPORATION

850 Frelinghuysen Avenue
Newark, N J

Manufacturers of AEROFIN

The Standardized Light-Weight Heat Exchange Surface

11 West 42nd Street, NEW YORK

Land Title Building	**United Artists Building**	**Burnham Building**
PHILADELPHIA	**DETROIT**	**CHICAGO**

Aerofin is the modern Standardized Light-Weight Encased Fan System Heating and Cooling Surface originated by *Fan Engineers* to meet the present and future requirements of this highly specialized field All Standard AEROFIN Units are furnished as completely encased Units, ready for pipe and duct connections The patented casings are built of pressed steel and are exceptionally strong and rigid, protecting the Unit from all the strains of pipe connections and expansion or contraction in service The casings are flanged on both faces, top and bottom, and template punched for bolting together adjacent Units, or for duct connection

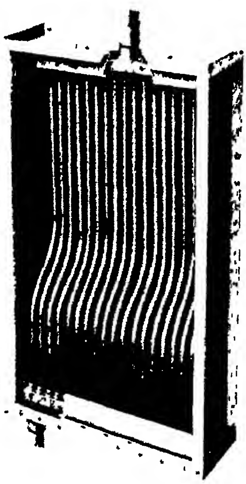


Fig 1

Flexitube Aerofin (Fig 1) supplants the original non-corrodible Low Pressure AEROFIN

Flexitube AEROFIN is distinguished from all other developments by its off-set tubes, so arranged as to absorb all expansion and contraction strains

Headers—Cast bronze or aluminum

Tubing—5/8 in O D copper, admiralty or aluminum

Joints—Where admiralty or copper tubes are used together with bronze headers tubes are brazed to headers using Mueller patented joint Where both aluminum tubes and headers are used tubing is welded to headers

Casings—Copper, aluminum or galvanized iron

Design—Units are constructed with headers on opposite ends making possible installation of units with tubes horizontal or vertical

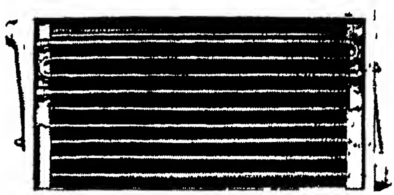


Fig 2

Universal Aerofin: (Fig 2) is distinguished by its "S" bend construction of tubing, the units being designed with steel headers on opposite ends, the ends of the "S" bends being connected thereto by compression nuts, the bends taking care of the expansion and contraction of the tubing

This type of surface is recommended where close control is desired

Headers—Pressed steel

Tubing—1 in O D Copper, admiralty or aluminum

Casings—Copper, aluminum or galvanized iron

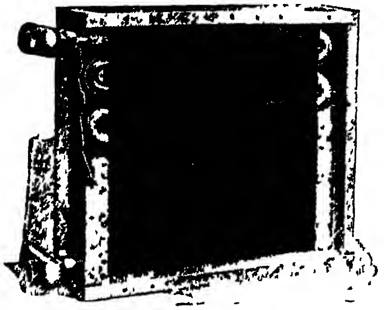


Fig 3

High Pressure Aerofin: (Fig 3) AEROFIN is of the continuous tube design being recommended where extremely high pressures of steam are used

Headers—Pressed steel

Tubing—1 in O D Copper, aluminum or admiralty

Casings—Copper, aluminum or galvanized iron

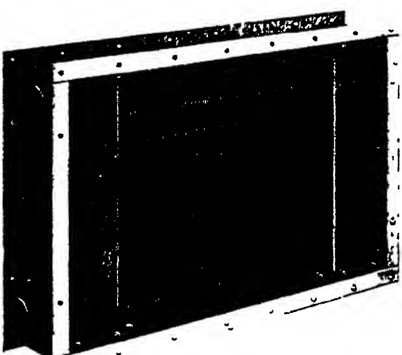


Fig 4

Booster Aerofin: (Fig 4) is of the continuous tube design and is recommended where small volumes of air are used, or where it is designed to raise the air temperatures in branch ducts, etc

Headers—Cast iron

Tubing—$\frac{5}{8}$ in O D Copper or aluminum

Casings—Copper aluminum or galvanized iron

Aerofin Encased Booster Units: (Fig 4) For horizontal or vertical air flow Six sizes, 150 to 1624 c.f m

For either horizontal or vertical air flow

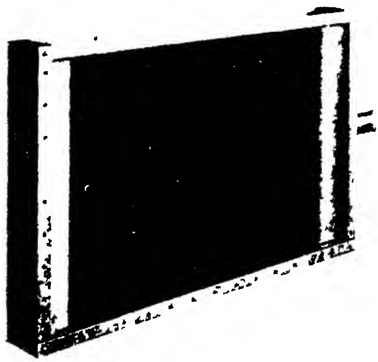


Fig 5

Narrow Width Aerofin: (Fig 5) recommended for water cooling or for flooded Freon systems Made in straight tubes only with headers on opposite ends, joints between headers and tubing being brazed Construction similar to Flexitube AEROFIN

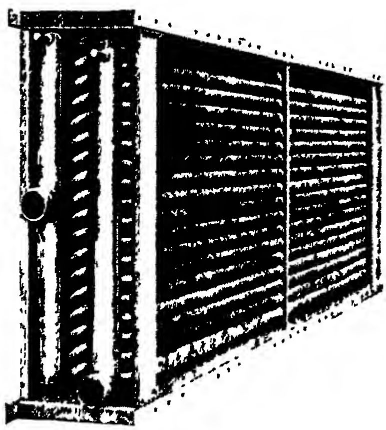


Fig 6

Aerofin Continuous Tube Water Coils: (Fig 6) are designed for air cooling by circulating cold water through the AEROFIN and air over the extended fin surface These units can be made for either horizontal or vertical air flow

Tubes and fins are made of copper, completely tinned with a permanent metallic bond between fin and tubes Headers are made of one-piece cast bronze and casings of heavy galvanized iron or copper

Every unit is tested to 1000 lb hydrostatic pressure

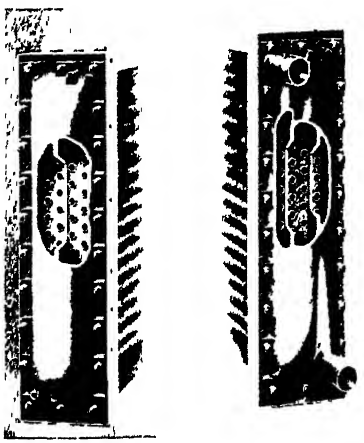


Fig 7

Aerofin Cleanable Tube Units: (Fig 7) for cooling only and all made with headers removable so as to permit of cleaning out tubes Recommended for use where sediment or scale forming chemicals are present in the cooling water

Headers—Cast iron
Tubing—Copper or admiralty
Casings—Copper or galvanized iron

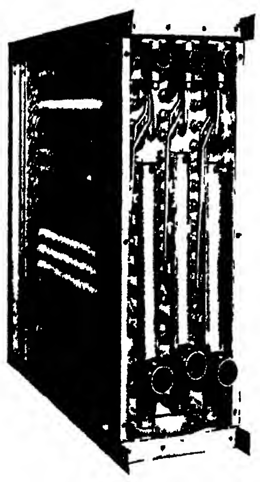


Fig 8
End plate removed showing distributing and suction headers

Aerofin Direct Expansion Units: (Fig 8) Row Control Type—Recommended for use where cutting on or off rows of tubes in direction of air flow is desired Suitable for use with Freon or Methyl-Chloride

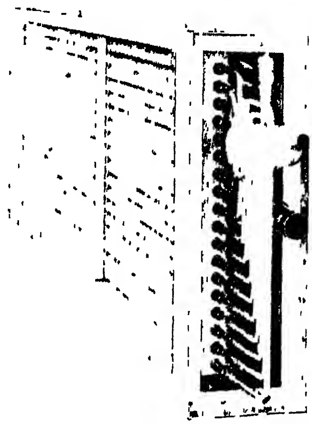


Fig 9

Aerofin Direct Expansion Units: (Fig 9) Centrifugal Header Type—Recommended where control of rows in direction of air flow is not required

Advantages: Weighs but 9 to 16 per cent of same equivalent cast iron surface and occupies one-third of the space Eliminates expensive foundations and building re-inforcement Can be suspended from roof beams or trusses if necessary

AEROFIN Sizes

Flexitube: 13 standard lengths, three widths, one and two rows deep

Narrow: same as Flexitube

Universal: 17 standard lengths, two widths, one and two rows deep

Continuous Tube: 13 standard lengths, three widths, 2-3-4-5 and 6 rows deep

Cleanable Tube: 17 standard lengths, one width, 2 and 4 rows deep

Direct Expansion: Row Control—11 standard lengths, 3 widths, 1-2-3 rows deep Face Control—11 standard lengths, 3 widths, 2-3-4-5-6 rows deep Centrifugal Header—11 standard lengths, three widths, 2-3-4-5-6 rows deep

Steel Supporting Legs: 18 in and 24 in high Punched same bolt hole centers as standard casings Quickly attached No other foundation required

Sale: AEROFIN is sold only by manufacturers of nationally advertised Fan System Apparatus List upon request

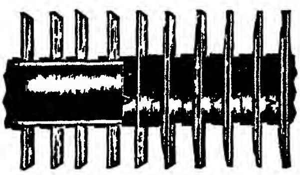
Write Newark for Heating Bulletin G-32, Direct Expansion Bulletin DE-34 on refrigeration type units, Continuous Tube Bulletin C T 34 for Water Cooling Coils, or pamphlet on Cleanable Type AEROFIN for cooling

The G & O Manufacturing Company

138 Winchester Avenue **New Haven, Connecticut**

G&O INDIVIDUAL FIN TUBING

RADIATING ELEMENTS FOR ALL HEAT TRANSFER PURPOSES



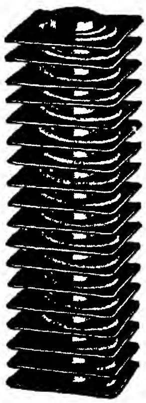
INDIVIDUAL G & O FINS—The use of individual fins results in high efficiency in heat transfer from primary tube surface to secondary fin surface, because all G & O fins have ample collars around the tube opening insuring liberal contact with tubing

ANY SIZE OR SHAPE—Fins of any size or shape may be obtained giving any desired proportion of primary and secondary radiating surface

SQUARE FINS—A square fin has about 30% greater surface than a round fin of a diameter equal to a side of the square

INDIVIDUAL FINS—Individual fins permit of any fin spacing also, of using fins in groups at intervals along tubes. Fins placed at practically any angle on tubes

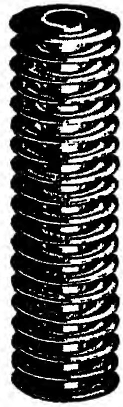
VARIOUS SHAPES—G & O individual fin tubing is furnished in straight lengths in U bends with short radii in continuous return bend coils, or other shapes. Ends of tubes may be left free of alloy for mechanical joints



G & O Individual Fin Tube Data

Standard Sizes

O D of Tube	Fin Size	Fin Spacing per Inch	Surface per Linear Foot
3/8"	3/4" sq	6	0.55 sq ft
3/8"	7/8" sq	6	0.80 sq ft
3/8"	7/8" r'd	6	0.60 sq ft
3/4"	1 1/2" r'd	6	1.55 sq ft
3/4"	1 5/8" sq	6	2.40 sq ft
1"	2 1/8" sq	6	4.00 sq ft



GRINNELL COMPANY INC

Heating, Industrial and Power Plant Piping, Fittings, Hangers, Valves, Pipe Bending, Welding, Piping Supplies, Etc.

Executive Offices: **Providence, R. I.**

Offices Plants and Branches

Albany N Y
Atlanta Ga (Plant and Foundry)
Auburn R I (Plant and Foundry)
Baltimore Md
Boston Mass
Buffalo N Y
Charlotte N C (Branch)
Chicago Ill (Branch)
Cincinnati, Ohio
Cleveland Ohio (Branch)
Columbia Penna (Plant)
Dallas Texas
Detroit Mich
Kearny N J (Branch)
Milwaukee Wis
Minneapolis Minn (Branch)
Newark N J
New Orleans La
New York N Y
Philadelphia Penna (Branch)
Pittsburgh Penna
Providence R I (Plant and Foundry)
Rochester N Y
St Louis Mo (Branch)
St Paul Minn (Branch)
Warren Ohio (Plant and Foundry)

GRINNELL COMPANY OF THE PACIFIC

Los Angeles Cal (Branch) Oakland Cal (Branch) San Francisco Cal (Branch) Seattle Wash (Branch)

GRINNELL COMPANY OF CANADA, LTD

Montreal, Que (Branch) Vancouver B C (Branch) Toronto Ont (Plant and Foundry) Winnipeg, Man
Oshawa Ont (Foundry)

PRODUCTS AND SERVICES—

Complete Service on materials to Specification on Power Plant Piping, Industrial Piping, and Industrial Heating Systems; Prefabricated Piping including Pipe Cutting and Threading, Pipe Bends, Welded Headers, Welded and Welding Fittings, Lap Joints and the Grinnell line of products for Super Power.

Grinnell Equiflo Valves for forced hot water heating systems; Grinnell Adjustable Pipe Hangers and Supports; Grinnell Cast Iron and Malleable Iron Pipe Fittings; Grinnell Malleable Iron Unions; Grinnell Welding Fittings; Grinnell Thermoliers (Unit Heaters); Grinnell Thermofin (Convectors); Thermoflex Traps and Heating Specialties.

Also Humidifying Systems; Constant Level Size Circulating Systems; Piping for acids and other special materials.

Malleable Iron, Brass, Bronze and other Castings; Brass, Cast Iron, Wrought Iron and Steel Pipe; Seamless Steel Tubing in Iron Pipe Sizes.

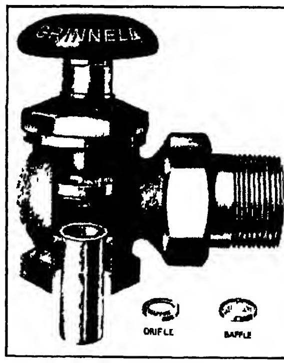
Valves: Check, Globe, Pressure Reducing and Regulating, Quick Opening, Safety and Y.

Automatic Sprinkler Systems; Stand Pipes; Underground Supply Mains; Hydrants; Fire Pumps; Pressure and Gravity Tanks.

Grinnell "Junior" Automatic Sprinkler Systems for Basements and other hazardous areas of Dwellings, Small Apartment Buildings, Schools, Churches, Stores, etc.

Grinnell Equiflo Valves

For Forced Hot Water Heating



Equiflo Valve

The designing of forced circulation hot water heating systems is so simplified by the Grinnell Equiflo Valve that they can be laid out and installed as easily as vapor or steam systems This valve consists of a regular type packless radiator valve with a cartridge or tube made up of a series of orifices and baffles capable of setting up any required frictional resistance This method of establishing any desired resistance does away with elaborate calculation of pipe sizes Grinnell guarantees perfectly balanced circulation to each and every radiator where these valves are installed throughout the system

Equiflo Data Book sent to interested parties

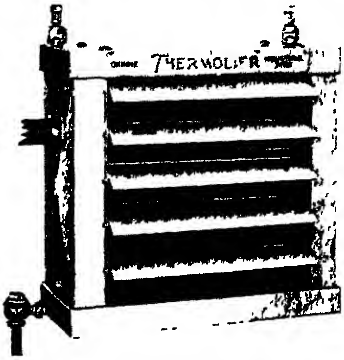
For Data on Thermoflex Traps and Heating Specialties, see page 1078

THERMOLIER

Patented

THE GRINNELL UNIT HEATER

Industrial and Factory Types—125 Lbs. W.S.P.



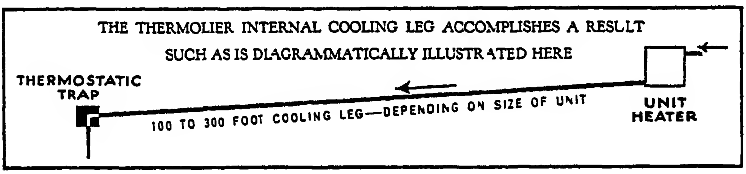
Industrial Type

Thermolier is a ruggedly built unit heater whose efficiency and dependability have been proved by actual performance in field service Thousands of them are installed in industrial buildings and commercial structures of all types of occupancy

Thermolier has 14 points of superiority, the most outstanding of which is the internal cooling leg built right into the unit, an exclusive Thermolier feature See drawing below

Radiation is from brass-finned seamless copper U-tubes rolled into a cast-iron tube sheet No solder is used for strengthening joints and there are no flat horizontal surfaces to catch dirt.

Units may be controlled manually or automatically, singly or in groups Installation and piping are extremely simple and inexpensive, hence the unit may be moved from one



location to another at small cost if found desirable on account of changes in building or occupancy The complete line includes 28 Models in Two Types

Thermoliers provide maximum distribution of heat without objectionable drafts

Specifications

Fan—Grinnell special of rugged construction **Motor**—heavy duty, oversize, enclosed, moisture-proof **Housing**—Copper on Industrial Type with rubbed lacquer finish, steel on Factory Type finished in gray duco **Frame**—Heavy pressed steel, providing rugged support for motor and fan **Special Features**—Adjustable swivel hanger rod couplings, louvers rigid, but easily adjustable integral cooling leg insuring perfect drainage through one thermostatic trap for pressures up to 25 lb

For pressures not exceeding 125 lb, a thermostatic trap of proper construction can be used and should be attached directly to the unit

CAPACITIES

60° F. Entering Air Temperature—2 lbs Steam Pressure

Model Nos	Btu per Hour	Model Nos	Btu per Hour	Model Nos	Btu per Hour
20	35,000	50	90,700	100	234,000
20L	26 900	50L	67,100	100L	196 000
25	40,500	60	104,800	110	259 000
25L	30,900	60L	77,700	110L	211,600
30	47,800	70	142,000	140	320,000
30L	35,200	70L	117,000	140L	271,000
40	69,400	80	164,600	180	368,000
40L	53,300	80L	139 300	180L	294,000
45	81,200	90	189,200		
45L	62,600	90L	151,600		

Data Book covering other pressures and temperatures dimensions and complete installation information on application Address GRINNELL COMPANY, INC, 277 West Exchange Street, Providence R I

GRINNELL ADJUSTABLE PIPE HANGERS AND SUPPORTS

One of the chief advantages of Grinnell Adjustable Hangers is that they permit adjustment of pipe lines after installation, thus obviating the necessity of turnbuckles or the removal of hangers Their time and trouble-saving qualities during installation are equally exceptional Below are shown a few Grinnell Hangers and Supports of particular interest to heating engineers Send for Hanger Catalogue showing complete line

Adjustable Swivel Rings (Patented)

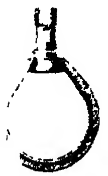


Fig No 101
Solid Ring

These Malleable Iron Adjustable Swivel Rings can be used with Coach Screw Rod or Machine Threaded Rod in connection with practically any type of Ceiling Flange, Expansion Case, Insert, etc

Adjustment of at least 1½ in is secured by turning Swivel Shank Swivel Shank automatically locks, preventing loosening due to vibration in the pipe line

The Split Ring permits adjustment either before or after Ring is closed A wedge type pin is loosely but inseparably cast into the hinged section for fastening this section after pipe is in place

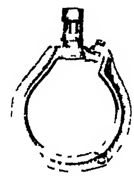


Fig No 104
Split Ring

Adjustable Swivel Pipe Rolls (Patented)

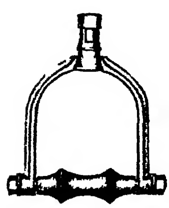


Fig No 174
Swivel Pipe Roll

An adjustable type of pipe roll using a single hanger rod Swivel Shank allows vertical adjustment and automatically locks, preventing loosening from vibration

CB-Universal Concrete Inserts
(Patented)

Made of air furnace malleable iron, in one body size, to take a special removable nut, tapped for ⅜ in., ½ in , ⅝ in or ¾ in rod as required Nuts automatically lock by means of V-type teeth on both insert and nuts

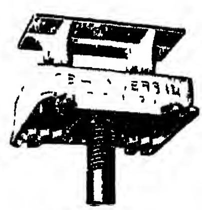


Fig No 282
CB-Universal Insert

GRINNELL WELDING FITTINGS

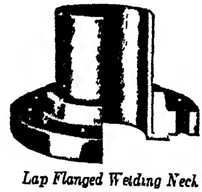
90° Elbow, Long Turn

Grinnell Welding Fittings are made from Seamless Steel Pipe and possess the same physical characteristics as standard, extra strong and o d steel pipe or seamless steel pipe of comparable size They can be used under the same conditions, pressures and temperatures as the pipe itself.

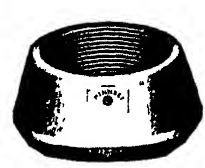
Welding faces for all plain circumferential Butt Welds are scarfed or beveled to the regulation 45 deg angle with 1/16 in square end on inside of fitting Angles of bevel other than 45 deg can be furnished on special orders

Welding Outlet

Welding Tee



Lap Flanged Welding Neck



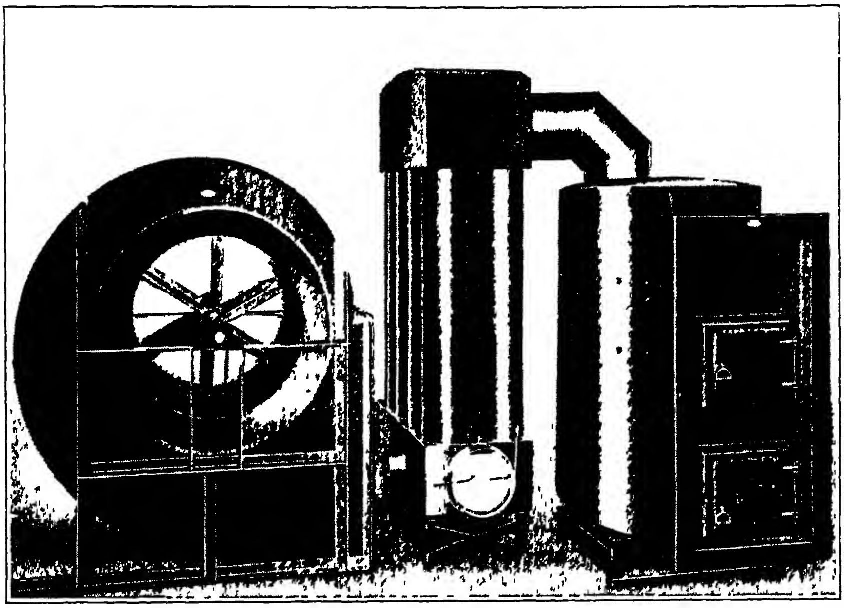
Threaded Outlet

E. K. Campbell Heating Co.

2441-3-5 Charlotte St Kansas City, Mo.

Factory Branch 4485 Olive St, St. Louis, Mo

THERMIDAIRE HEATING, VENTILATING AND AIR CONDITIONING EQUIPMENT



E. K. CAMPBELL FURNACE FAN SYSTEM OF HEATING AND COOLING—Heavy locomotive firebox steel furnace, acid resistant economizer, large combustion space, large vertical gas passages for low resistance and constant efficiency, great amount of surface for heat extraction from gases, counter flow air and gases

THERMIDAIRE FANS AND BLOWERS for heating, ventilating and cooling are sturdy, efficient and quiet operating

THERMIDAIRE STEAM UNIT HEATERS are built in all types The heating element is all copper, with a special brazing which has equal mechanical strength to the copper, insuring permanent tightness A wide range of types and capacities

THERMIDAIRE BLAST COILS have the same heating element as the unit heaters, free from leaks or expansion troubles Encased in heavy steel, they are easily installed, and made a permanent installation

THERMIDAIRE GAS FIRED UNIT HEATERS have the vertical tube, plunge draft design, with large gas passages, which make the units self-cleaning and maintain high and constant efficiency

THERMIDAIRE DIFFUSERS AND GRILLES are made in various designs Neat, inconspicuous, finished in baked enamel with choice of several colors

Further information gladly given on request

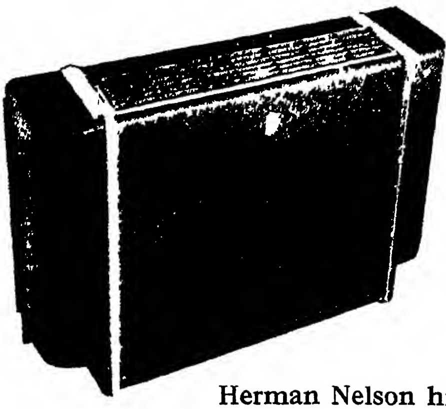
The Herman Nelson Corporation

General Offices and Factories at **Moline, Illinois**

Sales and Service Offices in all Principal Cities

Heating and Ventilating Equipment for Schools and Industrial Plants

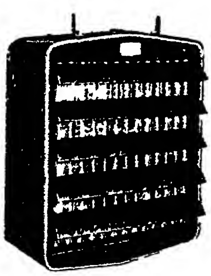
Herman Nelson Unit Ventilator



Herman Nelson Unit Ventilators for maintaining comfortable and healthful conditions in school classrooms consist of heating element, filter, cabinet, air intake, fan and motor unit and two sets of dampers—all designed and arranged to produce efficient heating and ventilating at low cost One Herman Nelson Unit Ventilator is all that is normally required in a classroom Other Herman Nelson equipment especially designed for large rooms, auditoriums and gymnasiums is available

Herman Nelson hiJet Heaters

The Herman Nelson Corporation manufactures a complete line of hiJet Unit Heaters in two types propeller fan and blower hiJet Heaters provide low cost heat and ideal working temperature for all types of industrial and commercial buildings Dependability and efficiency proved by actual performance in thousands of installations



Suspended Type

Available in eight sizes, each with single speed two speed or three speed operation on single phase 25, 50 and 60 cycle, 110 or 220 Volt current Also for single or multi-speed operation on 115-130 or 230-260 Volt direct current Three phase motors for larger models for 25, 50 or 60 cycle, 220 or 440 Volt current

Type D

Available in ten models Each model in two depths, 21 and 32 in The 21 in unit can be furnished with two, three or four fans, and the 32 in unit with two or three fans High or low capacity heating element available Furnished normally with direct connected motors, but equipped with V belt drives if desired

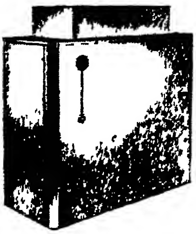
CATALOGUES AVAILABLE on both Herman Nelson Unit Ventilators and Unit Heaters For Complete Information and Specifications Write to THE HERMAN NELSON CORPORATION, MOLINE, ILLINOIS

The Herman Nelson Corporation

General Offices and Factories at **Moline, Illinois**

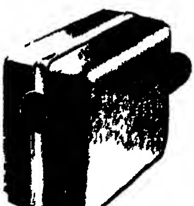
Sales and Service Offices in all Principal Cities

Complete Line of Automatic Heat and Air-Conditioning Equipment



Oil Burning Air-Conditioning Furnace

A simple, compact unit built from the ground up to deliver Automatic Heat and Air-Conditioning efficiently and economically Stack, electric and water connections are made from the rear, leaving the front ends free from obstructions Attractive in appearance, it occupies comparatively little space

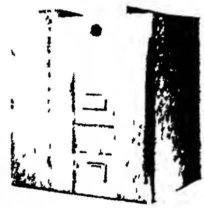


Conversion Oil Burner

Backed by 30 years of experience in the heating field, The Herman Nelson Conversion Oil Burner is stripped of all unnecessary gadgets Of the Pressure Atomizing Type, burner is fully enclosed Exclusive Herman Nelson Streamlined Fan Housing results in combustion efficiency Comes in four sizes

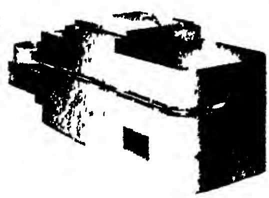
Oil Burning Boiler

Oil Burning mechanism, boiler and domestic hot water combined in one simplified unit, giving high heating efficiency and low cost operation Specially constructed Combustion Chamber induces rapid circulation Gas flow divided into fine, intensive heat streams Cabinet heavily insulated Removable panels permit ready access to all parts



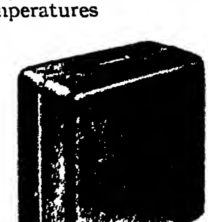
Coal Burning Air-Conditioning Furnace

Built to give trouble-free, low cost heating and air-conditioning Air-conditioning fans of exclusive design mounted on rubber in specially designed housing, give quiet operation and necessary air delivery Draft dampers open and close and circulating fans start and stop automatically Heat loss eliminated by heavy, cellular asbestos insulation



Automatic Stoker

The Herman Nelson Automatic Stoker burns low-priced coal with complete combustion Gives maximum heat from fuel used Fractional horsepower motor mounted in rubber, assures quiet operation and low power consumption All moving parts operate in constant bath of oil Tuyeres, retort and throat extra heavy Entire construction gives lasting protection against high temperatures



Self-Contained Summer Air Conditioner

Cleanses and humidifies as well as cools and circulates air Only 30 in high, 15½ in deep and 29½ in wide, it has a cooling capacity of 8,000 Btu per hour Air discharge is 375 cfm at high velocity Sound in every detail of engineering, material and workmanship.

CATALOGUES AND FOLDERS Available For Complete Information and Specifications Write to THE HERMAN NELSON CORPORATION, MOLINE, ILLINOIS

John J. Nesbitt, Inc.

AND

Buckeye Blower Company

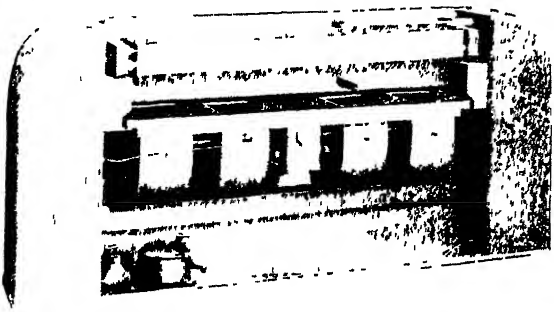
Manufacturers of Heating, Ventilating and Air Conditioning Equipment

EXECUTIVE OFFICES
State Road and Rhawn Street
Holmesburg, Philadelphia, Pa.

FACTORIES
Holmesburg, Philadelphia, Pa.
Columbus, Ohio

Buckeye Sales and Service Offices in Principal Cities of U S A

Sales and Service on Nesbitt School Room Unit Ventilators through Offices of American Blower Corp



Below — Cross-section view showing the Nesbitt Syncretizer mixing outdoor and room air

Nesbitt Series 400 Syncretizer

Called TOMORROW'S Heating and Ventilating Unit TODAY, because of its easy adjustment to present or future ventilation requirements Made in three types or cycles of control—F, A, and O All three types recirculate all room air during the heating-up period, before room occupancy **Type F,** during room occupancy, introduces all outdoor air **Type A,** during room occupancy, introduces any desired minimum quantity of outdoor air when heating is required, and an increasing quantity, up to all outdoor air, when cooling is required Type F may be readily converted to Type A cycle of control at any time by a simple adjustment within the unit

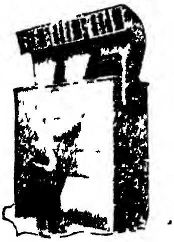
Type O, during room occupancy, introduces a variable quantity of outdoor air without a fixed minimum The quantity supplied is governed by the indoor and outdoor temperatures, and is that volume required to maintain a minimum air-stream temperature—usually 60 deg—at the fan discharge, before the air passes through the unit radiator

All three types are made in the same size and style of casing for a given capacity, and all have the same arrangement of component parts The differences are in the cycles of control All three have the Nesbitt Airstream Minimum Temperature Control to prevent cold drafts Ask for Publication 225 (engineering data) and Publication 226 (The Story of Syncretized Air)

Nesbitt Series B Thermovent—for Large Interiors

A large-volume unit ventilator incorporating three distinct features which have been proved essential to the proper heating and ventilating of auditoriums, gymnasiums, assembly halls, and like gathering places (1) *Quietness of operation,* due, in part, to large-diameter, forward-curved fans operated at low speeds, with V-belt drives, (2) *Modulated Steam Valve Control,* with uniform discharge temperatures over the entire radiator assured by Nesbitt internal steam-distributing tubes, and (3) *Air-Stream Minimum Temperature Control,* the Nesbitt development which prevents cold drafts

Furnished, like the Nesbitt Series 400 Syncretizer, in three types or cycles of control—F, A, and O—to deliver all outdoor air, or to partially recirculate, with or without a fixed minimum quantity of outdoor air Available in capacities ranging from 2000 to 6000 cfm, and in two types of casing, for exposed or hidden location Ask for Publication 227



Buckeye Giant Unit Heater

A blower, draw-through type unit heater for the economical heating of large areas in industrial plants, garages, airplane hangars, etc Made in a variety of capacities, ranging from 1880 cfm 139 500 Btu to 19 600 cfm, 1,290 000 Btu, with 2 lb steam, 60 deg entering air

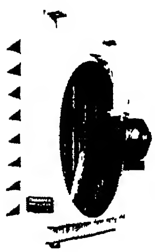
Giant Heaters are equipped with Nesbitt Copper Finned Tube Radiating Surface for operating steam pressures up to 150 lb gage Also applicable for use on Forced Circulation Hot Water Systems

Giant Heaters are built for Floor Mounting—Wall Mounting (both Standard Vertical and Inverted Vertical, and Horizontal Suspended

Wall boxes and outside air dampers can be furnished with all types

Giant Heaters are furnished with or without the Thermadjust Damper This efficient bypass damper with automatic or hand control prevents overheating and stratification and effects substantial fuel savings by controlling heat output in definite relation to demand Giant Heaters may be had with either V-belt drive or direct-connected motor Ask for Buckeye Publication 145

Giant Unit Heaters are tested and rated in accordance with the Standard Test Code of the American Society of Heating and Ventilating Engineers *and the Industrial Unit Heater Association*



Buckeye Unit Heater

A rugged, compact, suspended type disc fan unit heater, built in capacities ranging from 810 cfm, 35,800 Btu, to 5030 cfm, 313,500 Btu, with 2 lb steam 60 deg entering air, and for operating steam pressures up to 150 lb Also adaptable for use on Forced Circulation Hot Water Systems Available in either single- or multi-speed.

Buckeye Unit Heaters may be conveniently located to supply heat exactly where it is needed, and the louvers are individually adjustable to direct the air flow to the proper level Heavy brackets are provided at the top of each heater for attachment to hanger rods

Recirculating ducts or fresh air intake ducts, together with the necessary wall boxes and louvers, will be furnished with these heaters when specified The propeller fans used have been specially developed for these heaters and they combine lightness and durability with exceptionally high aerodynamic efficiency They are perfectly balanced and are securely attached to the motor shaft Ask for Buckeye Publication 144

Buckeye Unit Heaters are tested and rated in accordance with the Standard Test Code of the American Society of Heating and Ventilating Engineers *and the Industrial Unit Heater Association*

Nesbitt Heating Surface with Steam-Distributing Tubes

Sold by Leading Manufacturers of Fan System Apparatus

The increasing application of automatic temperature control to heating, ventilating and air conditioning systems has created the need for this new type of blast steam heating surface Nesbitt Copper Heating Surface, through the use of all-copper, fin-and-tube construction, retains the rapid steam condensing capacity and the permanence proved for light-weight copper radiation, and gains unique *controllability* by inserting, within the usual steam tubes, additional or steam-distributing tubes These carry the steam equally to all parts of the radiator section, regardless of the quantity supplied Thus, whether the section is operating at full capacity or merely tempering the incoming air, a uniform discharge temperature is assured Possibility of freezing is eliminated under normal operating conditions. Nesbitt Copper Heating Surface is made in a wide range of capacities, to meet all conditions Fins are made fast to tubes through mechanical construction processes, thus eliminating the use of solder or other bonding materials Ask for Publication 229

AMERICAN RADIATOR COMPANY

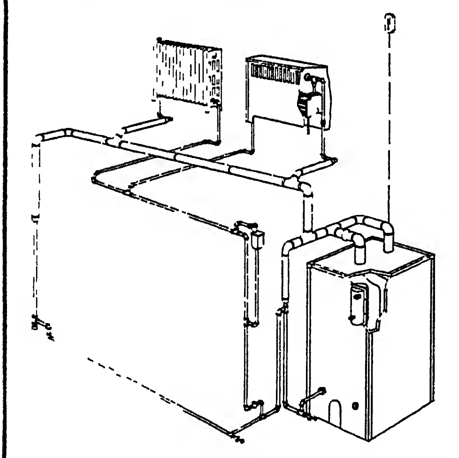
DIVISION OF AMERICAN RADIATOR & STANDARD SANITARY CORPORATION

40 West 40th Street, New York, N. Y.

CONTROLLED WARMTH SYSTEMS

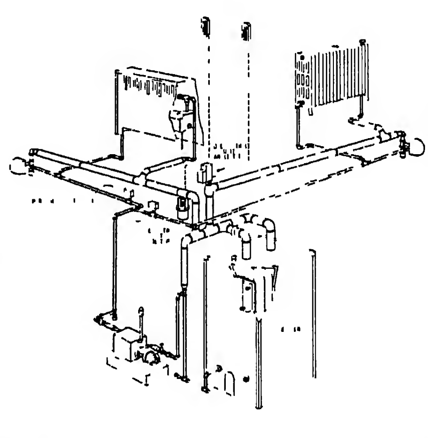
The Arco Model K Vapor-Orifice System is an engineered and coordinated unit which operates to insure equal distribution of warmth. On each radiator there is installed a special inlet valve with an adjustable orifice, by means of which flow of steam to each radiator can be accurately calibrated in accordance with the capacity of the radiator. The adjustment can be made while the system is in operation.

Pressure is so controlled that no more steam is admitted to the radiators than they are capable of condensing. Therefore no thermostatic traps are required on the return connections of the radiators. A thermostatic air eliminator equipped with a vacuum check is provided. If the pressure through inadvertence should exceed the design pressure, steam will enter the return lines, close the vent port of the air eliminator, and the system will automatically equalize itself, insuring the return of condensate to the boiler.



Schematic Piping Arrangement of Simple Model K System

When a zone system is used, the zone valves prevent the self-equalizing action and it is necessary to install an Arco Condensation Pump to insure the return of condensate to the boiler.



Schematic Piping Arrangement of Zoned Model K System

Simplicity of design and moderate cost make possible the installation of this system for little more than the cost of a One Pipe Steam System.

"Standard Specifications" and "Design and Installation Guide" including all details are available at American Radiator Company branch offices.

USED WITH MODEL K SYSTEM

Arco No. 900 Adjustable Orifice Valve—Orifice may be adjusted permanently to "meter" the correct amount of vapor to each radiator according to its capacity, to effect balanced heating.

See also Pages 843, 898-901

Davis Engineering Corporation

HEAT TRANSFER SPECIALISTS SINCE 1915

Branch Office
90 West Street
New York City

Paracoil

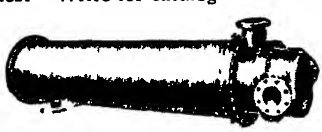
General Office and Factory
Elizabeth, N. J.

PRODUCTS—Davis Automatically Fired Boilers, Paracoil Storage Water Heaters, Instantaneous Water Heaters, Fuel Oil Heaters, Oil and Water Coolers, Distillers, Feed Water Heaters, Feed Water Filters and Grease Extractors, Steam Traps, Evaporators, Converters, Preheaters, Hot Water Generators, Condensers, Heat Exchangers

Years of trouble-free service without excessive maintenance charges have been the aim in the design of Paracoil apparatus Paracoil Heat Exchange Products are preferred by many leading industrial, naval and building engineers Many large ships and famous buildings have Paracoil heat transfer equipment Special heat transfer problems invited Write for bulletin on products in which you are interested

PARACOIL STORAGE WATER HEATER

For heating and storing hot water for apartment houses, office buildings, hospitals, laundries, etc Operates with boiler water, exhaust or live steam as heating medium Steel plate storage water shell with removable seamless drawn copper heating coils Any pressures specified Capacity 40 to 25,000 gal per hour Capacity tables, calculating data, etc on request Write for catalog



TUBULAR HEAT EXCHANGERS

For cooling lubricating oil or water and similar heat exchange uses Removable or non-removable tube bundle as required Transverse or Longitudinal Baffling as required Floating tube sheet and head construction or shell expansion joint compensates for expansion and contraction strains Built in single and multi-pass types Small space requirements Write for information.

PARACOIL INSTANTANEOUS U-TUBE TYPE WATER HEATER

Ideal for apartment houses, hotels etc Connected below water line of boiler, heats water supply when fires are banked Cast-iron or steel shell, heating elements of seamless drawn copper

Capacities 300 gal and up (below water line installation) or 20 gal and up (live steam installation)

Catalog on request

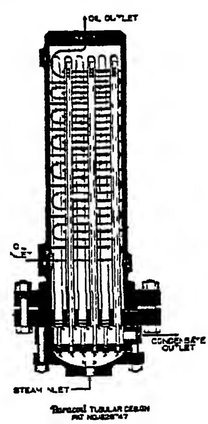
PARACOIL FUEL OIL HEATER, TUBULAR AND COIL TYPES

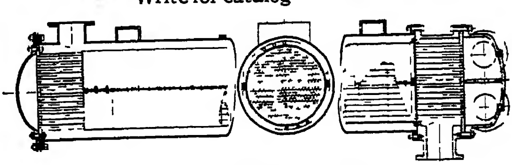
RanD System of Fuel Oil Preheating in Storage Tanks

Efficient and compact, removable heating element, freedom from expansion and contraction strains Designs permitting easy cleaning

Turbulent flow of oil assures high heat transfer Unusually low pressure drop or friction loss Variable baffle spacing gives variable velocity in proportion to change in viscosity Horizontal or vertical installation

Write for catalog





Section of Paracoil Tubular Heat Exchanger

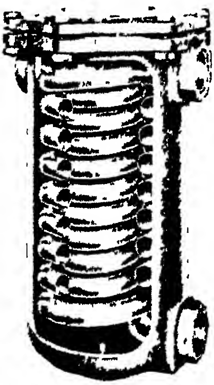
Bell and Gossett Company

3000 Wallace Street **Chicago, Ill.**

HOT WATER SYSTEMS AND SPECIALTIES

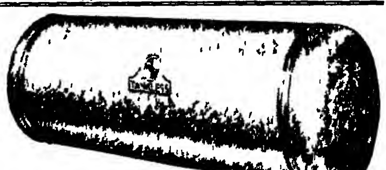
B & G INDIRECT WATER HEATERS FOR STEAM AND VAPOR SYSTEMS

Ratings below are based on 100 degree rise in three hours with boiler water temperature of 180 degrees or more



Description	No	Capacity Gallons	Max Length Inches	Max Width Inches	Shell Openings Inches	Coil Openings Inches	Shipping Weights Pounds
SINGLE COIL For residences of all sizes Duplex apartments and small buildings	30	30	11¾	5 1/16	1	¾	12
	40	40	12¾	5 1/16	1	¾	13
	60	60	14¾	7⅛	1	¾	15
	70	70	15⅛	7⅛	1½	1	29
	90	90	18¾	7⅛	1½	1	39
	100	100	20¾	7⅛	1½	1	40
	120	120	22¾	7⅛	1½	1	42
	150	150	25¼	7⅛	1½	1	46
DOUBLE COIL For larger apartments, garages, medium sized factories and office buildings	160	160	12	11½	2	1½	53
	200	200	14¾	11½	2	1½	59
	300	300	19⅝	11½	2	1½	78
	400	400	23⅜	11½	2	1½	86
TRIPLE COIL For heavier requirements	600	600	21⅜	15¾	3	2½	230
	800	800	25⅜	15¾	3	2½	250
	1000	1000	29¼	15¾	3	2½	290
DOUBLE TRIPLE COILS	1200	1200	23¾	29	4	3½	568
	1600	1600	27½	29	4	3½	642
	2000	2000	30¾	29	4	3½	716

B & G TANKLESS HEATER



This B & G Tankless Type Heater has been designed to fulfill a need for a heater of unusual capacity that could be installed in boiler rooms lacking space for storage tanks

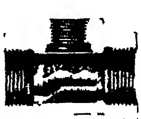
For all Steam, Vapor and Hot Water Heating Boilers

Series Number	Coil Openings	Shell Openings	Boiler Water Capacities 180 F	200 F	212 F	Length	Diameter	Shipping Weight Pounds
12	¾"	2"	110	175	225	26"	12"	80
16	¾"	2"	145	246	305	35"	12"	100
20	¾"	2½"	175	300	375	34"	12"	130
30	¾"	3"	250	430	516	51"	12"	180

B & G THERMOCHEK

Controls automatically the temperature of hot water storage tanks heated by indirect heaters A simple, completely automatic device that prevents lime and sediment formation in the heater coils Sizes 1½ in and 2 in

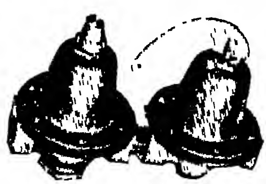
B & G MONOFLO FITTING



This fitting is used on mechanically - circulated one-pipe hot water systems When installed in the main at each radiator connection, it assures a balanced system with short circuits eliminated and all radiators receiving their full quota of hot water

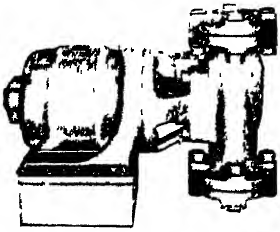
Sizes 1 in , 1¼ in , 1½ in , 2 in

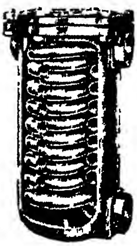
B & G No. 3 DUAL UNIT



A better combination Relief-Reducing Valve at low cost Body and all parts exposed to water are bronze Extra-large chemically treated composition diaphragm Relief Valve opens at 28 lbs pressure Reducing Valve easily adjusted for various building heights

B & G TRIPLE DUTY SYSTEMS FOR HOT WATER SYSTEMS†

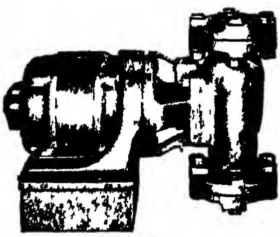




Size No	Recommended Tank Sizes	Capacities Gals per Hr	*Capacities Sq Ft Net Rad	Shipping Wt Lbs
11	30 gals	30	750	111
C-11	30 gals	30	750	97
12	45 gals	35	750	115
C-12	45 gals	35	750	102
13	66 gals	40	750	118
C-13	66 gals	40	750	114
14	88 gals	50	750	125
C-14	88 gals	50	750	121
15	30 gals	30	1,500	150
C-15	30 gals	30	1,500	133
16	45 gals	35	1,500	154
C-16	45 gals	35	1,500	137
17	66 gals	40	1,500	157
C-17	66 gals	40	1,500	140
18	88 gals	50	1,500	164
C-18	88 gals	50	1,500	131
19	70 gals	30	3,000	173
XC-19	120 gals	70	1,500	168
20	66 gals	40	3,000	180
XC-20	180 gals	100	1,500	184
21	88 gals	50	3,000	187
XC-21	300 gals	140	1,500	188
22	120 gals	70	3,000	207
C-22	120 gals	70	3,000	181
23	180 gals	100	3,000	223
C-23	180 gals	100	3,000	197
24	300 gals	140	3,000	227
C-24	300 gals	140	3,000	201

*Capacities in square foot of direct radiation are based on 200 Btu radiator emission with a 20° F drop

†NOTE —Bell and Gossett Company have always recommended the use of a Flo-Control Valve in both feed and return main to prevent any possibility of an air-bound boiler or of radiators heating by back circulation in summer If the system, however, has a D & T Air-Cushioned Tank installed in exact accordance with our instructions, a Type "C" System, using only one Flo-Control, will work satisfactorily

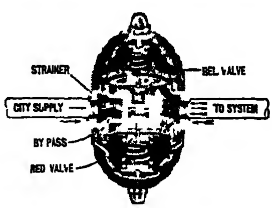


B & G BOOSTER

Size No	Face to Face Dimensions	Flange Size	Motor 110 Volt 60 Cycle	Normal Delivery Gallons per Minute	Shipping Weight
1"	7¼"	1" Screwed	1/6 hp	15 at 2 ft head	62 lbs
1¼"	8½"	1¼" Flanged	1/6 hp	23 at 2½ ft head	68 lbs
1½"	8½"	1½" Flanged	1/6 hp	30 at 2½ ft head	75 lbs.
2"	8½"	2" Flanged	1/6 hp	60 at 2½ ft head	79 lbs
3"	12"	3" Flanged	1/3 hp	100 at 5½ ft head	136 lbs

D & T SELF-FILLING, AIR-CUSHIONED PRESSURE SYSTEMS

Self-Filling Air-Cushioned Tank Equipment consists of a D & T Air Tank and a D & T No 500 "All In One" Unit which is a combination relief and pressure reducing valve, performing the double duty of relieving excess pressure and keeping the system filled with water



System No	Capacity Sq Ft Radiation	System No	Capacity Sq Ft Radiation
518	Up to 500 ft	540	Up to 2000 ft
524	Up to 800 ft	560	Up to 3000 ft
530	Up to 1200 ft	580	Up to 4000 ft
535	Up to 1600 ft		

D & T SIMPLEX TANK-IN-BASEMENT PRESSURE SYSTEM

This system consists of an air-tight D & T Compression Tank, D & T Simplex Relief Valve (illustrated), and D & T Vacuum Breaking Valve Made in 8 different sizes with capacities up to 5000 sq ft of radiation

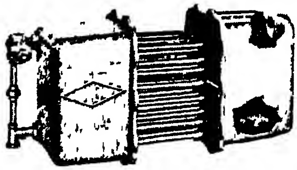
Taco Heaters, Incorporated

342 Madison Avenue, New York

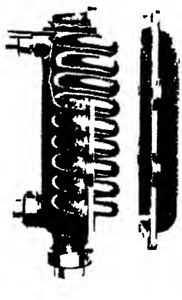
TACO HEATERS, Inc offers a complete line of water heating specialties for domestic hot water supply on steam and vapor (*Taco-Abbott System*) and hot water boilers (*Taco-Trol System and Equipment*)

TACO-ABBOTT SYSTEM

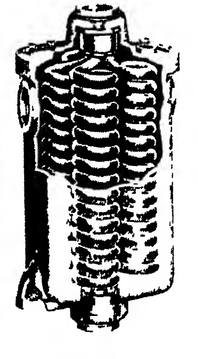
For Domestic Hot Water Supply on Steam and Vapor Systems THE TACO-ABBOTT SYSTEM makes possible the efficient year 'round operation of a steam or vapor heating boiler to supply Domestic Hot Water This *patented* System provides for automatic control of the heating boiler in summer so that when the room temperature reaches the desired point, the control of the firing device is transferred from the room thermostat to the aquastat, which is operated by the water in the boiler This aquastat controls the boiler water at a sufficiently high temperature to heat the domestic water without producing heat in the radiators



Tankless Taco No 14



Domestic Taco



Multi-Coil Taco

CAPACITIES, DIMENSIONS AND SHIPPING WEIGHTS OF TACO HEATERS

Size of Taco	Gallons in 3 Hours Heated from 40° F to 140° F: Below Water Line: Boiler Water at 212° F	†Boiler Water from 160° to 190° F	On Steam of 0 Lbs Gauge Pressure	Sq Ft Hot Water Radiation: Below* Water Line Boiler at 212° F	On** Steam of 0 Lbs Gauge Pressure	Size of Taco	Pipe Connections: Boiler Conn., In	Tank Conn In	Height In	Diameter In	Shipping Weight Lbs
A	B	C	D	E	F	G	H	I	J	K	L
				DOMESTIC TACO							
00	30					00	1	3/4	8½	5½	10
0	30		40	15	20	0	1	3/4	8¾	5½	10
30	30		50	20	30	30	1	3/4	10	5½	12
1	40		75	30	50	1	1	3/4	12	5½	14
1A	52	30	100	40	65	1A	1	3/4	15	5½	18
1B	66	30 to 40	125	50	85	1B	1¼	1	14	7½	22
2	82	40 to 52	150	60	100	2	1¼	1	16½	7½	27
2A	100	52 to 66	175	75	125	2A	1¼	1	18¾	7½	31
2B	120	66 to 82	225	90	150	2B	1¼	1	21¼	7½	34
2C	144	82 to 100	275	105	175	2C	1½	1	19½	9	50
3	160	100 to 120	300	120	200	3	2	1¼	21¾	9	65
3A	200	120 to 144	375	150	250	3A	2	1¼	26	9	80
				MULTI-COIL TACO							
M 60	250	120 to 185	450	210	300	M 60	2	1½	11	9⅞	75
M 75	300	150 to 220	600	250	400	M 75	2	1½	13½	9⅞	84
M 100	400	200 to 300	750	320	500	M 100	2½	2	13½	12	120
M 125	500	250 to 360	900	420	600	M 125	2½	2	15¾	12	140
M 150	600	300 to 430	1200	500	800	M 150	3	2½	16	14⅞	205
M 200	800	400 to 570	1500	640	1000	M 200	3	2½	20⅜	14⅞	255

Multi-Coil height represents measurement between supply and return connections to the Boiler

*Based on 120° F inlet temp to heater and 160° F outlet temp

**Based on 140° F inlet temp to heater and 180° F outlet temp

†Years of experience affirm our conviction that for best results a liberal size Taco should be selected Installing a slightly larger Taco than is absolutely necessary makes possible lower aquastat setting and consequent fuel saving This more than pays for the slightly additional initial cost

TANKLESS TACO Nos. 14 AND 16

Tankless Taco No 14 recommended for 1 bath or small 2 bath residences **Tankless Taco** No 16 recommended for 1 to 3 bath residences Write for information on New No 15 **Tankless**

Taco Tempering Valve is available in following sizes ½, ¾, 1, 1¼, 1½ and 2-in

TACO TANKS

Tank					Unit		Capacities, Gallons per Hour			
Capacity, Gal	Size In	Conn In	Gauge Metal Galvanized		Heating Surface Sq Ft	Conn, In	Heated from 50 to 150° F Below Water Line		Heated 50° to 180° F Steam 0 Lb Gauge Pressure	Shipping Weight Lb Galvanized
			Shell	Head			Boiler Water at 212° F	Boiler Water at 190° F		
40	16 x 48	6-1	10 ga - [illegible]/64	7 ga - 3/16	6	2	80	40	120	202
66	18 x 60	6-1	10 ga - [illegible]/64	7 ga - 3/16	6	2	90	40	120	263
82	20 x 60	6-1½	10 ga - [illegible]/64	7 ga - [illegible]/16	7½	2	100	50	150	296
100	22 x 60	6-1½	9 ga - [illegible]	7 ga - [illegible]	7½	2	100	50	150	318
120	24 x 60	6-1½	7 ga - [illegible]/16	3 ga - 1/4	7½	2	100	50	150	450
144	24 x 72	6-1½	7 ga - [illegible]/16	3 ga - 1/4	9	2	120	60	180	601
180	30 x 60	6-1½	7 ga - [illegible]/16	3 ga - 1/4	13½	3	180	90	270	800

Galvanized (copper bearing steel) 300 lb test pressure, 127½ lb working pressure Copper Monel Metal or Everdur 250 lb test pressure, 106¼ lb working pressure

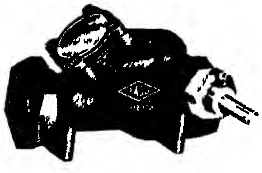
TACO-TROL SYSTEM Heating Specialties for Hot Water Systems

Advantages of the Taco-Trol System—All radiators heated uniformly—Rapid heating of radiators—Efficient heat control—Dependable hot water supply—Lower fuel consumption—Less piping—Smaller radiators—A saving in time and labor.

Operation—Taco-Trol System is a forced circulating one-pipe hot water heating system which performs double duty of heating the home to uniform temperature during the winter months and provides an all-year-round domestic hot water supply

The heat is supplied, when needed, rapidly and automatically through the employment of a *Taco Circulator* and a *Taco Check* When no heat is required, the Circulator stops, the Check Valve closes, and an Aquastat maintains a temperature in the boiler just sufficient for domestic hot water purposes A transformer relay is necessary with low-voltage thermostat

Taco Hot Water Heating Specialties can be used on either one-pipe or two-pipe hot water heating systems but Taco-Trol fittings make possible the use of a one-pipe system with considerable reduction of labor and material costs.

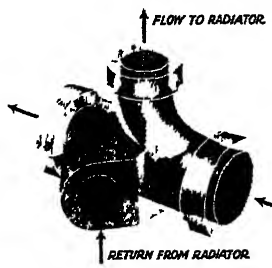


Taco Check Valve

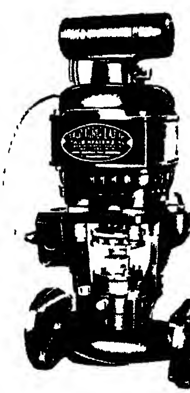
TACO CHECK

A high grade carefully balanced automatic flow valve for horizontal installation A stainless steel shaft protruding to outside facilitates hand operation for cleaning

Taco Check Valves are made in sizes corresponding to Taco Circulators



Taco-Trol Fitting



Taco Circulator

TACO CIRCULATOR

Made in five sizes 1, 1½ and 2 in direct drive, 3 in belt driven with either 3 or 4 in impeller Regularly furnished 110 volt, 60 cycle capacitor motors

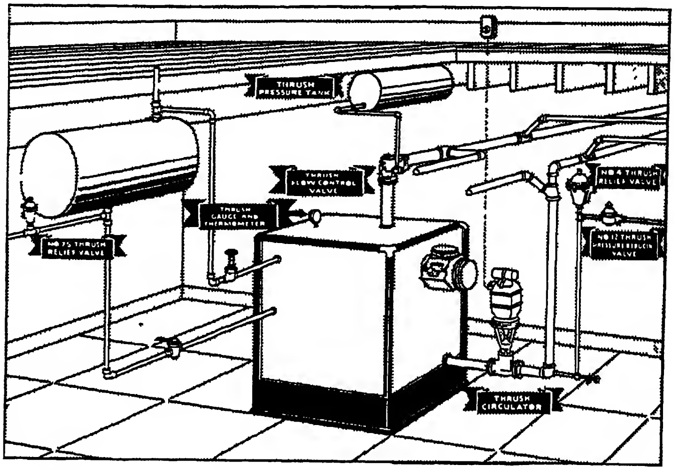
Circulator Size In	Capacities Radiation Square Feet		Gals Circulated per Minute	Domestic Hot Water Storage Tanks, Gals	Dimensions, In		Motor Hp
	Taco-Trol One Pipe System	Two Pipe System			Face to Face of Flanges	Overall Height Approx	
1	750	500	15	1,000	8¼	18½	1/8
1½	1000	800	30	1,500	8¼	18½	1/8
2	2000	2000	60	2,000	9	19½	1/6
3A	4000	4000	100	3,000	13¼	18¼	1/3
3B	5000	5000	130	4,000	13¼	18¾	1/2

H. A. Thrush & Co.

Peru, Indiana

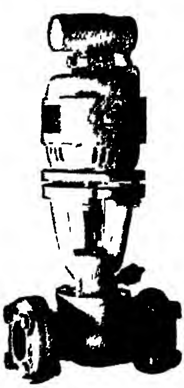
Service and Sales Offices in Principal Cities

THRUSH FLOW CONTROL SYSTEM



Patent Reissue No 19873

THERE are many advantages to be gained from the addition of Thrush Flow Control System to any hot water heating plant, old or new It makes the job heat quickly, prevents overheating, heats every radiator in every room uniformly, furnishes domestic hot water supply economically, as a by-product, and permits the year 'round use of the heating boiler and automatic firing device to supply domestic hot water It also increases the efficiency of the firing device and reduces fuel consumption On new work it permits the use of smaller pipes and valves which completely offset the cost of Thrush Equipment



Thrush Water Circulator

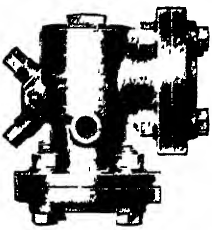
FORCED CIRCULATION

Thrush Flow Control System consists of a Thrush Water Circulator, a Thrush Flow Control Valve, a No 201 Thrush Radiant Heat Control, a No. 12 Thrush Pressure Reducing Valve, a No 4 Thrush Differential Pressure Relief Valve, and a special airtight Thrush Pressure Tank This converts an open gravity job to a closed pressure system with automatic filling and maintenance of the proper water supply, automatic relief of excess pressures, adequate provision for expansion when the water is heating and complete mechanical control of circulation It also anticipates outside temperature changes The heating plant becomes completely automatic, flexible and quick to respond to the slightest need for heat

H. A. Thrush & Co.

Peru, Indiana

Service and Sales Offices in Principal Cities



Thrush Flow Control Valve

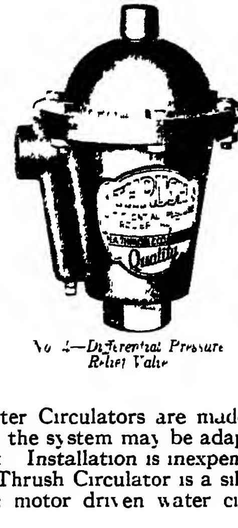
SIMPLE CONTROL

Provides a simple method of controlling circulation The water in the boiler is maintained at a constant temperature and cannot pass through the Flow Control Valve to the radiators until the Thrush Radiant Heat Control calls for heat

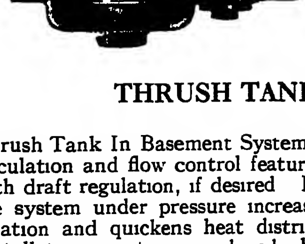
NO OVERHEATING

When the Thrush Radiant Heat Control operates it actuates the Thrush Circulator which quickly forces water through the Flow Control Valve to every radiator in the house, restoring room temperature almost instantly Then the Flow Control Valve shuts off all further circulation, preventing overheating and attendant waste of fuel Running time of firing device is shortened

Air and gases are vented and expansion provided for beneath the valve seat, an exclusive Thrush feature, so the valve closes tightly Thrush Flow Control Valves and Thrush Water Circulators are made in five sizes so that the system may be adapted to any size plant Installation is inexpensive and easy. The Thrush Circulator is a silent, efficient, electric motor driven water circulating pump



No 4—Differential Pressure Relief Valve

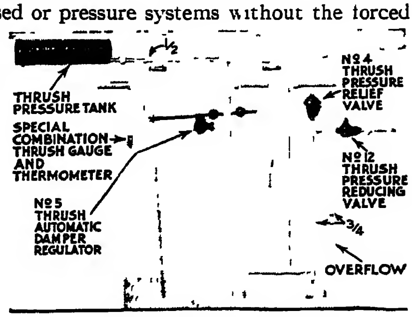


No 12 Pressure Reducing Valve

THRUSH TANK IN BASEMENT SYSTEMS

Thrush Tank In Basement Systems are closed or pressure systems without the forced circulation and flow control features, but with draft regulation, if desired Putting the system under pressure increases circulation and quickens heat distribution Installation cost is reduced because smaller pipes and valves may be used

Thrush Tank In Basement Systems are made in different types to care for hot water heating jobs of every conceivable kind, all providing the advantages of the closed system with increased circulation Each system is made in sizes to care for the needs of any job



Class A 4 Tank In Basement System

The Bristol Company

Pioneers in Process Control Since 1889

Waterbury, Connecticut

Branch Offices

			PITTSBURGH, PA
AKRON OHIO	CHICAGO, ILL	LOS ANGELES, CALIF	ST LOUIS, MO
BIRMINGHAM, ALA	DENVER COLO	NEW YORK, N Y	SEATTLE, WASH
BOSTON, MASS	DETROIT, MICH	PHILADELPHIA, PA	SAN FRANCISCO, CALIF

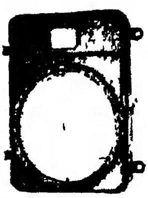
THE BRISTOL CO OF CANADA, LTD, 64 Princess Street, TORONTO, ONT, CANADA
BRISTOL S INSTRUMENT CO, LTD, North Circular Road, LONDON, N W 10, ENGLAND

TRADE-MARK

BRISTOL'S

REG U S PAT OFFICE

Recording Pressure Gages



Recording Pressure Gage Model 40M

For securing continuous day and night records of pressure or vacuum for steam, air, gas and liquids Furnished with charts reading in pounds, ounces, feet, inches, metric or any other desired units for ranges from full vacuum to 12,000 lb per sq in

Recording, Indicating Thermometers

For all commercial ranges from 60° F below to 1000° F above zero.

Electric Indicating Thermometer indicates temperature at any number of distant points

Metameter for Telemetering

Consisting of (1) a recording receiver in your office or other central place, (2) a transmitter at the far point where the measurement is made, and (3) a simple circuit of two small gage wires (or any existing telephone line) for carrying the electric impulses automatically sent out by the transmitter, the Metameter tells accurately and instantly how pressure, temperature, or flow is fluctuating at any point a hundred feet or several thousand miles away.

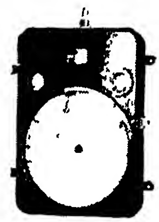
No interference with or from power, light or telephone lines Easy to read 12 in diameter chart, with wide open scale. D C or A C , 25 or 60 cycles, self-checking

Automatic Temperature Control

Metameter Transmitter, Model M35T

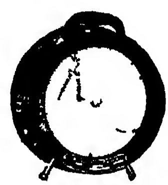
A complete line of Automatic Control Instruments is available for temperatures up to 3000° F. These are suitable for application to steam, oil, gas and electric heated equipment Both electric and air operated models are furnished

Since Bristol makes both control instruments and automatic valves, it is able to develop the complete control system as a unit Bristol has the experience and facilities correctly to engineer the system and to assume individual responsibility for the success of the results secured



Free Vane Temperature Recorder Controller, Model 5840M

Direct Reading Relative Humidity Recorder



Thermo-Humidigraph, Model 4069

Chart record shows at a glance the trend of humidity condition Novel vapor-sensitive hygroscopic element Special aging process assures sustained accuracy Eliminates calculation, and use of humidity tables No water required No fan used Accurate below freezing temperature Light portable corrosion-resisting case

Flow Meter for Steam, Liquids, Gas

Employs orifice and mercury manometer system of measurement Meter body is of forged steel, with stainless steel parts inside the mercury chambers All connections are welded or mechanical The stuffing box is of hardened stainless steel. Grease-packed and leak-tight

For working pressures up to 1000 lbs Special bodies available up to 3000 lbs

Industrial Stem Thermometers

With plain or red background Fixed thread union connection or socket design Straight form, or rear oblique, right side and left side angle forms

Wet- and Dry-Bulb Psychrometer

Accurate and dependable instrument by which relative humidity or atmospheric moisture may be determined Available in self-contained and distance types

Brown Instrument Division

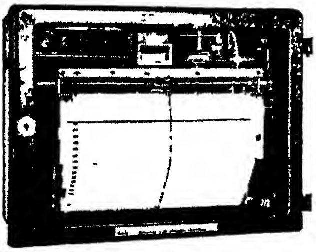
of Minneapolis-Honeywell Regulator Co

Main Office and Plant

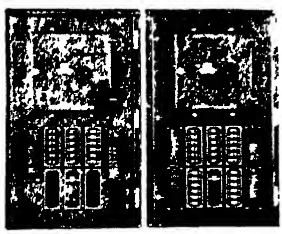
Philadelphia, Pa.

Equipment available through more than sixty Minneapolis-Honeywell Branch Offices

See list of offices under Temperature Control [illegible] *Page 111*[illegible]



Brown Recording Resistance Thermometer



Brown Balanced Bridge Type Resistance Thermometer and Multiple Key Switch Panel

Brown Portable Thermometer

Brown Instruments

The extent to which air conditioning equipment is being used in office buildings, theatres stores, industrial buildings, etc, has opened up a wide demand for indicating and recording resistance thermometers because the temperatures throughout these air conditioning systems should be checked periodically to get the best results at minimum operating cost

Indicating

To obtain uniform conditions from modern equipment, it is necessary that the one in charge have a visual picture of actual conditions If the individual units of the system are controlled automatically, the operator will know when this control is adequate and when it is not Whether automatic or manual, the Brown Resistance Thermometer will detect all variations, allowing the operator to maintain the conditions as set forth in the specifications

Recording

Uniformity of temperature and the consequent control of humidity depends on reliable facts obtained concerning wet- and dry-bulb temperatures

Controlling

Because the three-lead Brown Resistance Thermometer is entirely independent of variations in lead resistance, variation in bulb location lengths, number of bulbs, etc, it affords an excellent means of obtaining reliable results Bulbs may be placed at will or changed from one location to another, instruments may be moved about, and readings or records taken without any delay due to recalibration of leads, instruments or bulbs.

In addition to Resistance Thermometers, the Brown Instrument Division manufactures

Thermometers	Flow Meters
Hygrometers	CO_2 Meters
Pressure and Vacuum Gauges	Tachometers
Potentiometer Pyrometers	Liquid Level Gauges

Write for catalog on "Brown Instruments for Heating, Ventilating and Air Conditioning" which includes a complete set of sample specifications

See also Pages 1114-1115

Consolidated Ashcroft Hancock Co., Inc.

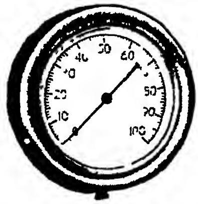
Bridgeport, Conn.

BRANCHES IN PRINCIPAL CITIES

Makers of AMERICAN INDUSTRIAL INSTRUMENTS—Since 1851

Manufacturers of Indicating and Recording Gauges, Gauge Testers; "U" Gauges, Draft Gauges; Indicating and Recording Thermometers, Tachometers; Dial Thermometers, Pressure and Temperature Controllers, Electric Temperature Controllers, Pop Safety and Water Relief Valves, Steam Traps, Engine Indicators, Counters; Absolute Pressure Gauges. Also manufacturers of Bronze, Cast Steel and Forged Steel Valves, Locomotive and Engine Room Clocks, Barometers; Mercury Column Gauges, Steam Whistles, Hydraulagraphs, Gauge Boards

Ashcroft American Gauges—Ashcroft American Gauges are made in all sizes from 2½ to 12 in, for pressures from 8 oz to 25,000 lb and also for vacuum Cases are cast-iron or cast brass The movements are Heavy Duty and all bearings are Monel Metal. Write for Catalog No A-59



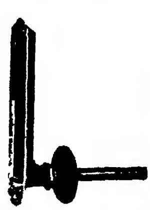
For Mercury Pressure and Vacuum Gauges, "U" Gauges, Draft Gauges and Mercurial Barometers, write for Catalog B-59.

American Recording Gauges—American Recording Gauges are made for all pressures from 15 in of water to 10,000 lb and for vacuum. They are made in one size only to accommodate a 10 in. chart, having an effective scale width of 3⅝ in. The case is Die Cast with a dull black hard-rubber finish and with either bottom or back connection. The pen-arm is made of non-corrosive Monel Metal and is of the inverted type Operating instructions are lithographed on the chart plate so that they cannot be lost.

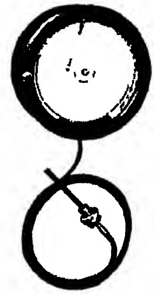
Especially designed Seth Thomas clocks are used, and all customary time periods can be furnished

American Recording Gauges are equipped with the Time Punch which virtually makes each instrument a time clock, since a hole is punched in the chart whenever a reading is taken. Write for Catalog E-59

American Air Duct Thermometer—Designed especially for both warm and cold air ducts. Fitted with chromium plated frame, glass front Furnished with 9-in or 12-in. scale graduated 0-160 F Write for Catalog F-59

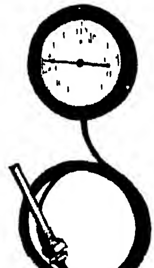


American Recording Thermometers—Made for recording temperatures from minus 40 to plus 1000 F. or equivalent C. Very flexible connecting tubing up to 200 ft One size to accommodate 10 in chart, with an effective scale width of 3⅝ in



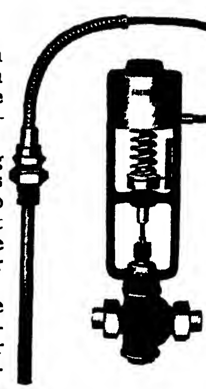
Same case as for the American Recording Gauge, so that all instruments are uniform in appearance when mounted on Gauge Boards Write for Catalog H-59

American Dial Thermometers—American Dial (mercury-filled) Thermometer has the accuracy of the standard glass tube thermometer and the reading convenience of a dial face Entire working mechanism is made of steel, meaning long life



Six sizes, ranging from 4½ in to 12 in diameter dials Furnished with rigid connection or flexible capillary tubing up to 200 ft For temperature ranges from minus 40 to plus 1000 F Write for Catalog G-59.

American Precision Temperature Controllers—Self-operated For regulating temperatures from 20 to 475 F For hot water service tanks, water heaters, etc Size of valve must be specified Write for No 2200 Bulletin



H-B Instrument Co., Inc.

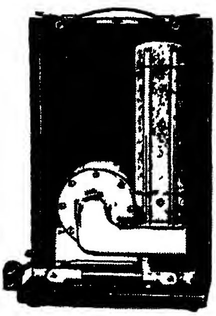
2523 North Broad Street — Philadelphia, Penna.

Accurate Testing Instruments For Air Conditioning Engineers

What Instruments Mean To Industry

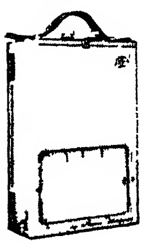
The accurate measurement of temperatures and humidity is essential in practically all processes of industry. Only with the knowledge supplied by such measurements can established processes be duplicated or better methods be developed. Only by such knowledge can costs be determined and economies be effected.

Motor Operated Aspirating Psychrometer



The H-B aspirating psychrometer is offered in two portable types the self-contained battery type and the plug-in-type, operating on 110 Volts, 60 cycles, A C. Both models employ the proven aspirating principle of drawing air at proper velocity by motor driven fan, across the sensitive mercury bulbs of two thermometers calibrated in single degrees. Request Bulletin 650

Combination Recorders Humidity and Temperature



Sensitive Elements—The Hygroscopic or Humidity Element is special multiple human hair. Temperature element is spiral special bimetal. Portable with strap handle, self-contained with week's supply of charts, extra pen and ink.

Compact—8½ x 5½ x 2¼ in, weight 52 ounces.

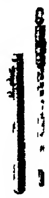
Charts are standard index cards, 3 x 5 in with reverse side ready to fill in date and record.

Case is die cast aluminum, satin silver finish with square grid back for exposing elements to temperature and humidity to be recorded. Request Bulletin 650

Sling Psychrometer

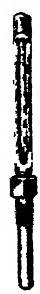
Aluminum frame and pivot handle partition between bulbs to prevent moisture from striking dry bulb during swing. 9 in thermometers range 10 to 110 F, ½ deg sub-divisions. Request Bulletin 650

Pocket-Handle Type



Brass Armor-chromium plated designed so that armor becomes handle when slipped over ring top and bayonet is engaged, dependable pocket clip, "stays put" overall 7 in x 7/16 in thermometer 4¾ in. Stock ranges minus 30 to 220 deg up to 30 to 1000 F. Request Bulletin 650

Glass Front Industrial Thermometers

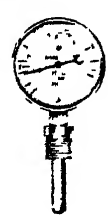


Easy to read Red Reading Mercury Tubing. Heavy engraved scale plates. Nickel plated brass parts. All standard ranges highly finished and accuracy guaranteed. Request Blue Book Part 3

Midget Thermometers

Brand new equipped with easy to read canary lens front tubing. Chromium plated brass armor. Angle and straight types. Request Blue Book Part 3

Dial Thermometers



Improved Bourdon Tube Vapor Tension Type. Rigid and Capillary Type. Chromium Plated Brass cases 2½ and 4½ in diameter dial standard ranges. Request Blue Book Part 3

H. B. Catalogs Available

Write for Catalogs—

Blue Book Part One. On Engraved Stem Laboratory Type Thermometers

Blue Book Part Two—Hydrometers

Blue Book Part Three—Industrial Type Indicating Thermometers Dial and Recording Types

Bulletin 650—Accurate testing instruments for air conditioning engineers

Illinois Testing Laboratories, Inc.

422 N. LaSalle Street, Chicago, Illinois

TESTING ENGINEERS AND MANUFACTURERS

"Alnor" Indicating Pyrometers—Portable and Stationary, "Alnor" Surface Temperature Pyrometers, "Alnor" Distant Reading Resistance Thermometers, "Alnor" Velometers (Boyle Type)—Air Velocity Meters.

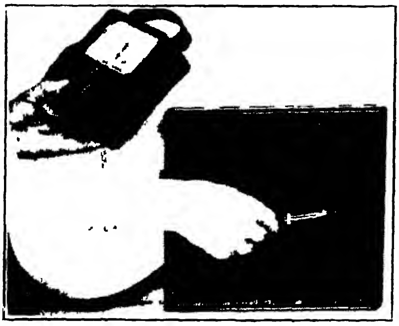


Fig 1—Showing Tube Type Velometer with No 5 Angle Jet taking air velocity of grille

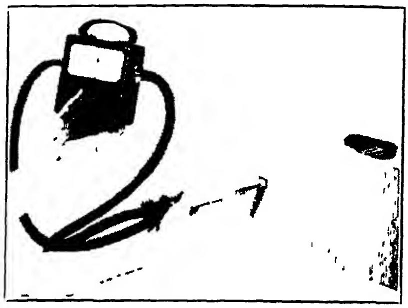


Fig 2—Showing Tube Type Velometer with Duct Jet taking readings directly in the duct Duct Jet furnished 18 in or 36 in long

"ALNOR" (BOYLE TYPE) VELOMETER

The Instantaneous Direct Reading Air Velocity Meter

The Velometer is a direct reading instrument which gives accurate and instantaneous readings of the speed of air measured in feet per minute

The indicator or meter is housed in an attractive black Bakelite case $5\frac{1}{4}$ in x $5\frac{3}{4}$ in x $2\frac{3}{8}$ in and weighs approximately 2 lb

The air which enters the meter actuates a movement comprising a swinging vane, control hair springs, pointer and a magnetic damping system

No mathematical calculations or stop watches are necessary to determine the velocity reading as the pointer indications are a direct measurement of air speed

While the Velometer is offered in three types, the Tube Type is by far the most popular This type uses attachments called jets for most of the velocity readings The style of jet to be used depends on the requirements or purpose for which it is used

No. 2215 Averaging Jet—This jet is offered for obtaining velocities of ordinary duct openings or grilles With this jet, average velocity readings over a 4 in circle can be quickly obtained

No. 2220 and No. 2240 Jets—These jets, angle and straight have small orifices to give spot velocities over a ½ in diameter The Velometer with these jets provides accurate spot velocity readings of irregular shaped or slotted openings or other small velocity streams, locates leaks in duct systems and other similar service that would be difficult by other means

No. 2425 Duct Jets 18 In. and 36 In. Long—These jets are offered for direct velocity readings in ducts The Duct Jet is, in effect, a double jet using two tubes and two meter fittings With this jet, static pressure is balanced out or subtracted from total pressure and the net result or velocity pressure is read directly on the scale of the Velometer in feet per minute Thus no calculations are necessary Inch marks are provided on the jet for the convenience of the user

No. 2460 Static Pressure Jet—This jet provides direct readings of static pressures of ducts, plenum chambers or pipes in inches of water

No. 2485 Total Pressure Jet—This jet gives direct readings of total pressure, in inches of water Inch marks are provided on this jet to enable the user to place the jet at the desired distance in the duct.

Other Jets—Other varieties of jets are available for special applications, but in general, the jets described above will enable the user to check and balance a complete air duct system of any size

The Velometer will be found indispensable by any heating, ventilating or air conditioning engineer or contractor or manufacturer of equipment

Write for Complete Folder

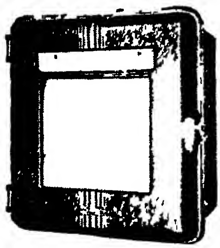
Leeds & Northrup Company

General Office and Works 4941 Stenton Avenue. **Philadelphia, Pa.**

Branch Offices

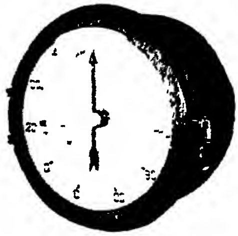
Chicago St Louis Boston Los Angeles San Francisco Cleveland Detroit Pittsburgh Houston Tulsa

RUGGED, NULL-TYPE INSTRUMENTS THAT ARE RELIABLE



Model S Micromax Recorder

Records from 1 to 16 points on a single strip-chart Extremely open record Can operate signals (About 1/15 size)



Model R Micromax Recorder

Records 1 or 2 points on a round-chart Has extremely readable dial Can operate signals (About 1 15th size)

Switchboard Indicator

Hand-operated Can be connected through selector switches to any number of points About 1 15th size

Electrical Thermometers for Air Conditioning

No method for measuring temperatures fits the specific needs of air conditioning as does the three-lead null-type resistance thermometer method It is independent of distance and disregards all temperatures except those right at detector locations The detectors (resistance thermometer bulbs called Thermohms), can be placed anywhere—in rooms, air ducts or water lines They are connected by simple electrical wiring to instruments at a central location Instruments may be Micromax Recorders, Model S for up to sixteen Thermohms, Micromax Model R, for related pairs such as wet and dry bulb, indicators with switches for any number of Thermohms; or indicating and recording combinations

This equipment is fundamentally reliable Instruments and Thermohms are highly responsive, yet rugged in construction A complete system is easy and economical to install, regardless of distances. It is easy to operate and demands minimum maintenance Thermohms and instruments are interchangeable, and can be replaced without disturbing wiring or returning anything to the factory

L&N Resistance Thermometers make it possible to operate efficiently, to maintain comfort or correct process atmosphere constantly . so that maximum return is realized on the entire conditioning investment

Electrical Instruments for the Heating Plant

The facts needed to operate a modern heating plant so as to save fuel, to protect equipment, and to operate efficiently at varying loads are provided reliably by rugged L&N instruments Readings can be indicated or recorded or both Recorders can be equipped to operate signals or alarms that warn the operator of extreme conditions In some cases the instruments control automatically

Micromax Model S provides a permanent record of conditions at from 1 to 16 points on one wide-scale chart. Micromax Model R concentrates on conditions at one point, provides a permanent record, and has a giant indicating dial that can be read at a glance. The switchboard indicator provides intermittent checks on conditions at one or several points.

In the heating plant, L&N measuring signalling or controlling equipment is used for

Metermax Combustion Control
Furnace Pressure Control.
Smoke Density Analysis
Flue Gas Analysis (Percent CO_2).
Flue Gas Temperatures
Steam and Water Temperatures
Boiler-Furnace Temperatures.
Electrolytic Conductivity of Water
Percent Leakage of Cooling Water

Taylor Instrument Companies

Rochester, N. Y., U. S. A.

IN CANADA—Taylor Instrument Companies of Canada, Ltd, Toronto

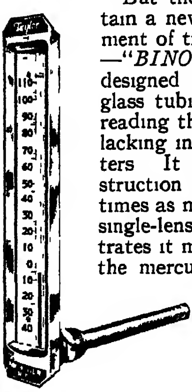
NEW YORK, CHICAGO, BOSTON, PHILADELPHIA, PITTSBURGH, CLEVELAND, LOS ANGELES, INDIANAPOLIS, SAN FRANCISCO, ST LOUIS, CINCINNATI, TULSA, DETROIT, ATLANTA, MINNEAPOLIS

Manufacturing Distributors in Great Britain, Short & Mason, Ltd, London

Manufacturers of Taylor Instruments for Indicating, Recording and Controlling Temperature, Pressure and Humidity

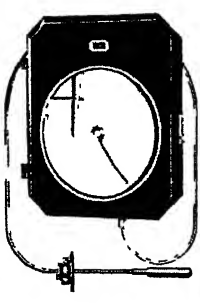
Taylor

Taylor Industrial Thermometers—with new "BINOC" Tubing—This line of thermometers includes many styles and scale ranges with bulbs for every application Suitable for air ducts, kiln temperatures and oven temperatures



But these thermometers contain a new and radical development of tremendous importance—"*BINOC" Tubing* This newly designed and optically correct glass tubing assures an ease of reading that has been generally lacking in industrial thermometers It has a triple-lens construction that gathers three times as much light as the usual single-lens tubing and concentrates it much more strongly on the mercury column A much wider background behind the mercury column eliminates bore reflection "*BINOC" Tubing* more than doubles the angle of vision within which readings can be made Its broad, contrasting mercury column can be read easily and accurately with both eyes at close range and also at greater than normal distances

Taylor Recording Thermometers—Temperature ranges and time requirements vary greatly in heating and ventilating work Taylor Recorders are made in needed scale ranges and time periods

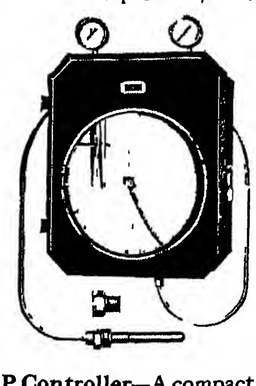


These instruments are beautiful and efficient, particularly adapted for heating and air conditioning applications They may be had for surface or flush mounting When set in panel boards, the polished flanges make an effective installation

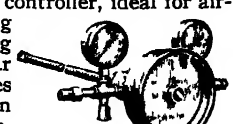
Taylor Electric Contact Temperature Control--These instruments combine in the same case an electrically-operated temperature controller with an indicating thermometer One tube system operates both units

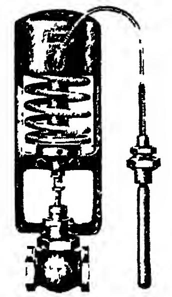
The New Taylor "Fulscope" Recording Controller An air-operated controller so versatile that practically any character of process control can be obtained, regardless of *time lag* in apparatus, by a simple screw driver adjustment

Available for controlling temperature, pressure, temperature and pressure, rate of flow, liquid level Where extreme load changes or badly balanced operating conditions exist, the *Taylor "Dubl-Response Control Unit"* is the only positive means of maintaining control-point *Write for literature*



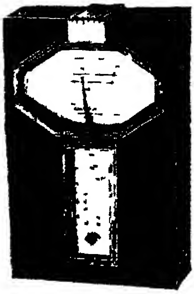
Taylor Type-P Controller—A compact and very sensitive controller, ideal for air-ducts, air-washing machines, cooling rooms and similar applications Uses compressed air as an actuating medium





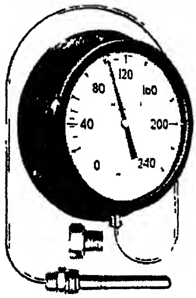
Taylor Self-Acting Temperature Controller—Adapted for use on hot-water storage tanks, etc It requires no auxiliary motive power, such as compressed air, to open and close the steam valve

The valve can be closed at any desired temperature, or a throttling action can be obtained Not practicable on pipe lines having steam pressure over 125 lbs

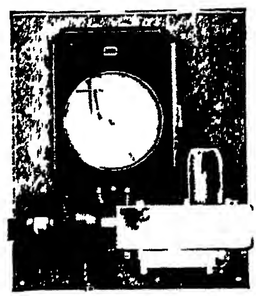


Taylor Hampton-Model Humidiguide (Direct-Reading)—A hygrometer giving direct humidity percentages, in a smart modern case suitable for home, office or public buildings Finish is satin black with chrome trim

The Permacolor Thermometer is filled with non-fading, easy-reading red liquid



Taylor Dial Thermometers for air ducts or any application where it is desirable to have temperature readings at some distance from the thermometer bulb, as in a central control room Can be read at a glance as easily as a steam gage.

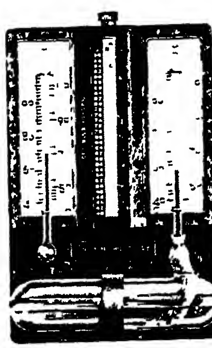


Taylor Recording Hygrometer—This instrument records both wet- and dry-bulb temperatures on the same chart in different colored inks, making comparison very easy.

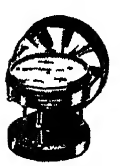
Type shown above with motor-driven fan for conditioned rooms or passages in which circulation is poor Can be supplied without fan for installations where circulation across bulb is good

Taylor Sling Psychrometer—The advantage of this form of Wet- and Dry-Bulb Hygrometer over the stationary form is the facility with which tests can be made and the accuracy of the readings obtainable, as in whirling the bulbs they are subjected to perfect circulation Consists of two accurate etched stem thermometers mounted on a die-cast frame, with the bulb of one covered with a wick to be moistened.

These thermometers have scales of 0 to 100° F, graduated in ½ degree divisions A copper case protects the tubes when not in use.



Taylor Humidiguide—A handsome small hygrometer for the wall of the home, office, school or other building where a neat, easy-reading and inexpensive instrument is desired It is self-contained, requiring no charts or separate tables Frame is Mahogany Bakelite.



Taylor Anemometer—This instrument is ideal for measuring air velocities with the fan revolutions indicated on the dial. Available in various models for a wide range of air speeds and registration limits

For complete information on above instruments and others designed for heating, ventilating and air conditioning, send for new Taylor Catalog Number Five to any of the offices listed on the previous page

The Palmer Company

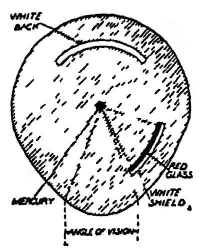
Main Plant 426 Clay Street, **Cincinnati (St. Bernard), Ohio**

Branch Factory THE PALMER THERMOMETER CO., LTD., 93 CHURCH STREET, TORONTO

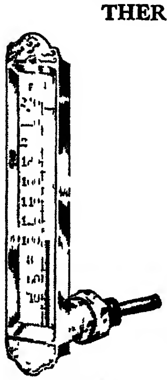
Manufacturers and Originators—"Red-Reading-Mercury" Thermometers

PRODUCTS—Indicating Thermometers with the RED column.

"RED-READING-MERCURY" was originated by Palmer We hold the basic patents on the reflected RED color See illustration A strip of red glass is drawn into the tube and when the mercury column rises, in the tube, the RED color is reflected upon it Once you have seen a thermometer with the RED column, you will not go back to plain mercury



INDUSTRIAL STYLE THERMOMETERS



No. 327

Furnished in all sizes and with STRAIGHT or various ANGLES styles. With Fixed taper thread, Union connection, Separable Socket or flange fittings

With the RED column, these thermometers are very easy to read

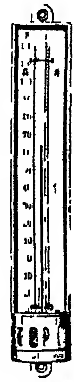
ACCURACY GUARANTEED; MADE STRONG AND DURABLE

OUTDOOR THERMOMETERS

Here is an ideal thermometer for use outdoors or inside The metal case is of bronze which will not corrode Graduated scale, giving very accurate temperature White Duco finish. Nickel-plate or Chromium finish also furnished. With the RED column, it can be seen a great distance

Other styles of Wall Thermometers as well as Laboratory styles furnished with the RED column

Write for FREE catalog No 200-C and Bulletin No 500



No 2120 Thermometer

Instruments for Testing AIR CONDITIONING EQUIPMENT

When air-conditioning equipment is installed, the contractor must have reliable and accurate instruments to make tests Only with such instruments can he give satisfactory results

No 14200

SLING PSYCHROMETER

This pocket style instrument can be carried anywhere and quick tests made Wet- and dry-bulb tubes guaranteed accurate Furnished with metal case, leather covered

ASPIRATING PSYCHROMETER

(This is not illustrated) The Aspirating Psychrometer for testing humidity is recommended where readings of greatest precision are required Contains small motor and blower so the air is blown over the wet and dry bulb thermometers. Two styles·

Battery operated for use where there is no electric connection

Electrically operated, to plug in on light circuit

With carrying case, neat, compact, light weight Very handy instrument.

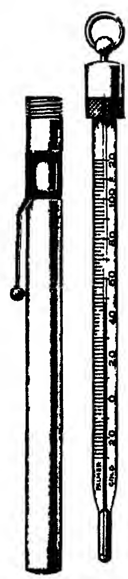
POCKET THERMOMETERS

It is helpful to carry a pocket test thermometer, so that it is available at any time

This is a reliable, guaranteed accurate thermometer Easy to read with the RED column

Repairs—We can repair all makes of mercury thermometers and furnish "Red-Reading-Mercury" at no extra cost. The type of repair work we do will add years to the life of the thermometer.

A trial Order will convince you.



No. 13240

United States Gauge Co.

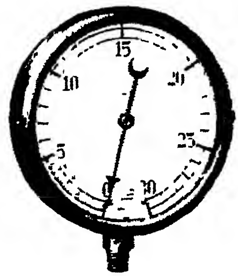
44 Beaver Street, New York, N. Y.

Factory at Sellersville, Pa

Makers of Quality

INDICATING AND RECORDING PRESSURE GAUGES

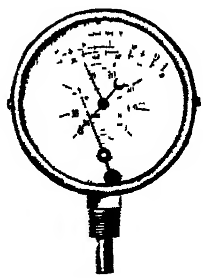
All Sizes and Types for Every Purpose



U. S. GAUGES—U S Gauges are made in all sizes from 2 to 12 in inclusive for pressures from 1 lb up to 50 000 lb, and for vacuum Cases may be cast-iron cast brass, drawn steel and drawn brass for wall mounting or flush mounting For severe service long wearing hardened steel or bushed movements may be supplied

For service on Steam Heating Systems—

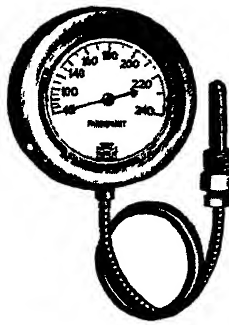
Steam Gauges Compound Pressure and Vacuum Gauges Retard Gauges Compound Retard Gauges Steam Gauges with Internal Siphons



For Hot Water Heating Systems—

Altitude Gauges . . . Tank-in-Basement Gauges . Altitude and Pressure Gauges . Combination Altitude Gauges, and (a) Bimetal Thermometers, (b) Glass Tube Thermometers, (c) Vapor Tension Distance Type Thermometers . Glass Tube Hot Water Thermometers

U. S. RECORDING GAUGES—U S Recording Gauges are made in 8½, 10 and 12 in sizes for pressures from 1 lb up to 50,000 lb, and for vacuum. Cases may be cast-iron or cast brass for wall mounting or flush mounting Pen arms are made of non-corrosive metal Especially designed clock movements are used Charts can be furnished for customary time periods



U. S. DIAL THERMOMETERS—U S Dial Thermometers are of the vapor tension type with open scale reading in the central and upper portion of the scale Cases may be cast-iron, cast brass, drawn steel or drawn brass for wall mounting or flush mounting Supplied in all sizes from 2 to 12 in. inclusive, for temperature ranges from 40 F to 800 F. Furnished with rigid connection bulb or with flexible capillary tubing up to 100 ft. long

Alfol Insulation Company

Incorporated

Chrysler Building :-: New York, N. Y.

Agents in Principal Cities

INSULATION *for*

Fans, Blowers, Pumps, *Ducts*, Houses, Buildings, etc., Turbines, Dehumidifiers, Air Conditioners, Boilers, Pipes, etc.

At temperatures up to 1250° F.

ALFOL

PAT IN 34 COUNTRIES

INSULATION *for*

Refrigerators, Refrigerator Cars, Refrigerator Trucks, Refrigerator Boxes, Refrigerated Rooms etc, Ships, Ovens, Ranges, Tanks, Stills

ALFOL consists of single or multiple sheets of polished pure aluminum foil installed between wall studs, furring strips, ceiling joists or roof rafters. When heat tends to flow through such insulated structures, 95 per cent of radiant heat is reflected back toward its source In summer, stifling outdoor heat is driven off. In winter, practically all supplied heat is retained in the house where it is needed

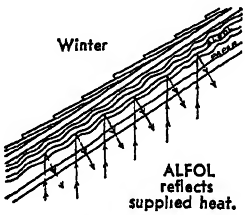
The reflection of heat is a revolutionary principle in insulation engineering which has won widespread recognition and acceptance by engineers and scientists throughout the world ALFOL is now extensively used by many of the largest manufacturing and industrial plants, U S Navy, and others, as well as in houses and buildings of every kind and size

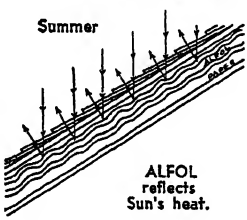
ALFOL will save up to 75 per cent or 80 per cent of the heat loss that would occur through an uninsulated house

It consequently costs less to heat an ALFOL insulated home.

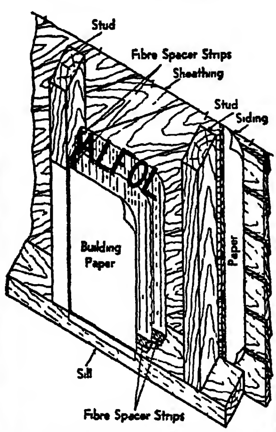
Many owners report over one-third saving in fuel costs

ALFOL is installed only by Authorized Applicators All work is done in strict accordance with ALFOL Standard Specifications





Trained mechanics unreel sheets of Alfol as needed These compact rolls cause no dust, dirt or waste



Alfol is flanged on edges and nailed to face of studs joists or rafters, through fibre spacer strips

This Alfol-insulated ceiling is equivalent in insulating effect to a solid concrete ceiling slab 8 ft thick

Aluminum Foil Does Not Tarnish

Data on Heat Savings Effected by Alfol

Construction	Heat Transmission BTU Hr Sq Ft °F			Heat Transfer Stopped %	
	Not Insulated	1 Layer ALFOL	2 Layer ALFOL	1 Layer ALFOL	2 Layer ALFOL
Wood Stud Wall—4-in studs	0 25	0 11	0 09	56%	64%
8-in Brick Wall—Furred	30	12	09	60	70
Open Attic Floor	62	13	10	79	84
Wood Shingle Roof (no lath and plaster)	46	13	09	72	81
Concrete Roof—6-in slab—suspended ceiling	35	13	10	63	72
Slate, Tile or Composition Shingle Roof (no lath and plaster)	56	15	11	73	81

These Coefficients have been determined in accordance with the method employed in THE A S H V E GUIDE, and are based on new insulating value of aluminum foil tabulated in National Bureau of Standards Letter Circular No 465, June 4, 1936, which contains the latest available data on the insulating value of aluminum foil

ALFOL ADVANTAGES

UNAFFECTED BY DAMPNESS

ALFOL is non-porous, has no capillarity, cannot absorb moisture Moisture infiltration in non-metallic insulations increases conductivity frequently as much as 50 per cent Such variations cannot occur in ALFOL

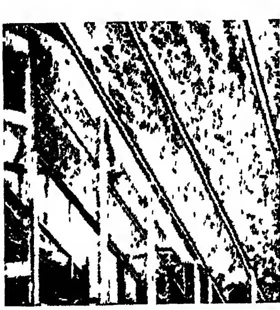
INSULATING EFFECT

Transmittance coefficients (see table above) are determined from tests made on actual wall sections insulated with ALFOL They accurately measure ALFOL's insulating effect including the heat loss through and around studs, joists, rafters, etc Where published coefficients of materials are based on 12-in by 12-in panel tests materials preheated in "bone dry" condition and at specified densities they do not include the heat loss at studs, nor the effect of moisture infiltration, variation in density, thickness, etc Many noted authorities on heat transfer recognize that the performance of ALFOL in actual use is difficult to surpass

FIREPROOF, CLEAN, AND SANITARY

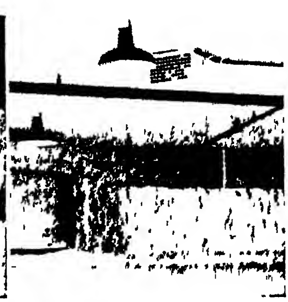
No heat-storage capacity ALFOL preheats or cools in approximately one-fourth the time required for other insulations Being metal, ALFOL is clean It repels vermin, prevents bacterial growth.

Alfol metal-jacketed pipe lines



Alfol applied between roof rafters reduces heat loss over 80 per cent

Recessed radiators give off about 15° more heat when insulated with Alfol



Alfol applied to overhead air duct of an Air Conditioning installation

Alfol Actively Reflects Radiant Heat

Armstrong Cork Products Company

Building Materials Division

Lancaster, Pennsylvania

District

Albany	Cincinnati	Houston	Minneapolis
Atlanta	Cleveland	Indianapolis	New York
Boston	Columbus	Jacksonville	Omaha
Buffalo	Dallas	Kansas City	Pittsburgh
Charlotte	Des Moines	Louisville	Rochester
Chicago	Detroit	Milwaukee	St Louis

Representatives

Baltimore	John R Livezey	Richmond	John R Livezey
Denver	Stearns-Roger Mfg Co	San Francisco	Van Fleet-Freear Co
Los Angeles	Van Fleet-Freear Co	Seattle	Asbestos Supply Co
New Orleans	H T Steffee	Spokane	Asbestos Supply Co
Philadelphia	John R Livezey	Tacoma	Asbestos Supply Co
Portland, Oregon	Asbestos Supply Co	Washington, D C	John R Livezey

For detailed technical information, samples, and descriptive literature, ask any office or representative

PRODUCTS—Armstrong's Corkboard, Armstrong's Cork Covering, Armstrong's Vibracork, Armstrong's Corkoustic, Armstrong's Temlok, Armstrong's Temcoustic, Armstrong's Insulation Sundries.

Corkboard

Sizes and Thicknesses

Armstrong's Corkboard is furnished in rigid boards 12 in by 36 in., 12 in. x 32 in , 18 in. x 36 in., 24 in x 36 in , and 36 in x 36 in., in several thicknesses 1 in , 1½ in , 2 in , 3 in , 4 in , and 6 in

Insulating Efficiency

The thermal conductivity of Armstrong's Corkboard, depending on the density, is 0 27 to 0 29 B.t u per hour per inch thickness at 90 deg F mean temperature (U S. Bureau of Standards)

The value of adequate and efficient insulation is covered in the text section of this book (Chapter 5) and the tables on pages 114 to 126 indicate the savings which can be effected by using 1½ in or 2 in of corkboard in standard wall and roof construction. The reduction in heat loss amounts to from 50 per cent to 75 per cent This means that an adequate thickness of corkboard on walls and roofs reduces the heat wasted, and, therefore, the heat requirements of the house by 25 per cent to 40 per cent.

Air Conditioning

An adequate thickness of corkboard insulation is essential for any air conditioned room or structure Insulation reduces the amount of heating or refrigerating equipment required to produce desired temperatures and is highly important to the satisfactory and economical control of humidity conditions

The insulation of air conditioning equipment assures economical and efficient operation by minimizing refrigeration losses De-humidifying chambers and ducts as well as pumps and brine storage tanks may be insulated with Armstrong's Corkboard or Cork Lagging

Cork Covering

Armstrong's Cork Covering is made of pure cork in sizes to fit all standard pipe sizes The inside surfaces of each piece are machined to assure an accurate fit, free from moisture-catching air pockets Cork covering is rigid and will not sag Thicknesses are Ice Water (1 20 in to 1 93 in), Brine (1 70 in to 3 00 in), and Special Thick Brine (2 63 in to 4 00 in)

Armstrong's Fitting Covers are designed to fit accurately standard ammonia and extra heavy fittings, both screwed and flanged, of all types The economies resulting from the application of Armstrong's Cork Covering to all cold lines repay the cost of the insulation and assure additional refrigeration savings

Vibracork

The elimination of noise and vibration transmission is of primary importance in air conditioning work Armstrong's Vibra-

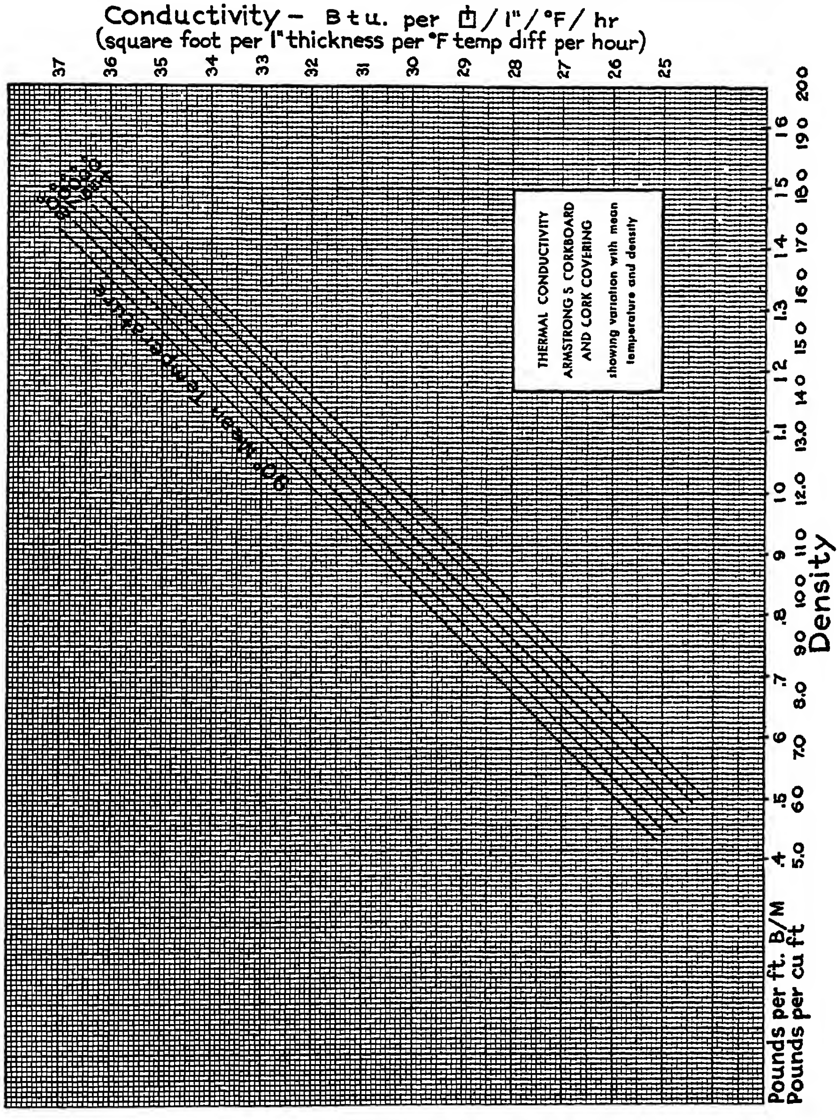
cork, made in three densities, is ideal for this purpose It does not take a set, is not affected by atmospheric moisture, and will not deteriorate in service

Products—Information

Additional Armstrong insulating and acoustical products especially suited to heating, ventilating, and air conditioning work include

Armstrong's Corkoustic
Armstrong's Temlok
Armstrong's Temcoustic
Armstrong's Insulation Sundries

For aid in the solution of any technical problems involving insulation, isolation, or acoustical treatment and for literature and prices, get in touch with an Armstrong district office or representative or the Armstrong Cork Products Company, Building Materials Division, Lancaster, Pa



The Agasote Millboard Company

Trenton, New Jersey

Manufacturers of Quality Products Since 1909

HOMASOTE
TRADE MARK

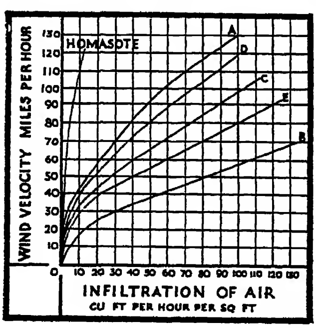
THE WEATHERPROOF INSULATING AND BUILDING BOARD

Homasote at Home—Protection from the Elements Unnecessary

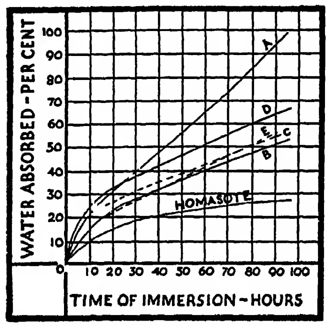
Homasote can be used as a combination insulating and structural material Important facts in the selection of an insulating material are its efficiency as an insulator, its structural strength, its moisture resistance, its plaster adhesion, its resistance to air infiltration, whether it is fire-retardent and the cost of installation and upkeep

Homasote is manufactured in BIG sheets, 8 ft x 14 ft, 8 ft x 12 ft, 6 ft x 12 ft, and even multiples thereof, making it possible to cover the average size wall in one piece It may be used on the exterior of a building even without painting

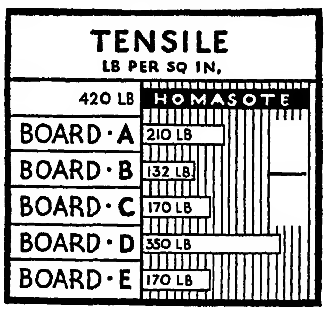
Comparative tests on air infiltration are shown below These tests—made by the Lewis Institute, Chicago, Ill—show that at a wind velocity up to 40 miles per hour, 18 to 112 times more air passed through the other boards tested than through



Homasote and at higher velocities, the infiltration through these other boards increased in a much greater proportion



Homasote has very high moisture resistance, according to tests made by the Robert W Hunt Co, Chicago, showing that the other boards took up 101 per cent to 274 per cent more moisture



In house construction, a material may be employed solely for its insulating value But, because of its rigidity, **Homasote** is often used as a combination insulating and structural material—such as sheathing on the outside of framing members, and as a wall covering within The chart above shows that **Homasote** has from 20 per cent to 217 per cent greater tensile strength than other boards tested Write direct for samples and descriptive literature

Cork Insulation Company, Inc.

155 East 44th Street, New York, N. Y.

Factory Wilmington Delaware

Branches

Boston, Mass	Philadelphia, Pa	Seattle, Wash
*Chicago, Ill	San Francisco, Calif	St Louis, Mo
Los Angeles, Calif		Washington, D C

*Represented by Corinco Insulation Co, Inc

PRODUCTS

Corinco Corkboard.
Corinco Cork Pipe Covering.
Corinco Cork Tiling.
Corinco Machinery Isolation.
Corinco Acoustical Corkboard.
Corinco Cork Lagging.

Corinco Corkboard

Corinco Corkboard is manufactured in sheets 12 by 36 in , in thicknesses of 1, 1 1 2, 2, 3 and 4 in Details in regard to weights freight classifications, shipping crates, etc will be furnished upon request Technical requirements conform in detail to the United States Government Master Specifications

Corinco Cork Pipe Covering

There are three standard thicknesses of Corinco Cork Pipe Covering, Brine, Ice-Water and Special Thick All thicknesses are supplied in 36 in lengths with a groove just sufficient to accommodate the specified diameter of pipe Properly applied this covering insures a snug fit precluding any possibility of frost forming and affecting its efficiency after installation

Machinery Isolation

The present day use of machinery for almost every industrial purpose has tended greatly to increase noise and vibration Vibration, aside from the fact that it is a direct cause of many noises is, in itself a menace to machines and to the structures surrounding them

Corinco Cork Machinery Isolation is available in standard boards, 36 in long, 12 in wide and 1, 1½, 2, 3 and 4 in thick The boards are manufactured in several densities to meet various requirements These densities are classified according to the number of pounds of Cork Compressed into one board foot.

Insulation

Proper and efficient insulation demands the employment of a medium which will insure a low degree of thermal conductivity Heating and refrigerating costs can be greatly affected, reduced or increased, by the use or omission of satisfactory insulating materials Cold storage equipment, particularly, is vitally dependent upon efficient insulation Freezers, Vaults and Cold Pipe Lines all must be protected in such a manner as to avoid any possibility of refrigeration loss Corinco Corkboard and Corinco Cork Pipe Covering give maximum protection wherever applied

Air Conditioning

The widespread use of Air Conditioning equipment necessitates that highly efficient insulating materials be employed when installation of ducts, pumps and brine storage tanks is being made Use of Corinco Corkboard will insure a satisfactory and economical control of humidity conditions Properly applied, this product will materially reduce the amount of heating or refrigerating equipment necessary to maintain a desired temperature

Additional Products

Corinco Acoustical Corkboard and Corinco Cork Lagging are also widely popular, the former for acoustical correction and decoration, the latter for insulating large pipe lines and boiler surfaces Any detailed information you may require in regard to Corinco products can be obtained promptly by addressing your inquiries directly to us, Cork Insulation Company, Inc , 155 East 44th Street, New York City

THE CELOTEX CORPORATION

919 N Michigan Ave, **Chicago, Illinois**

Mill NEW ORLEANS LOUISIANA

Boston, Mass
Minneapolis, Minn
Philadelphia, Pa
Denver, Colo
New York, N Y
Cleveland, Ohio

Sydney Australia
Paris, France
London England

CELOTEX
BRAND
INSULATING CANE BOARD
REG. U S PAT OFF

Los Angeles, Calif
St Louis, Mo
Seattle, Wash
Portland, Ore
Spokane, Wash
Tacoma, Wash

Buenos Aires, Argentina
Durban, South Africa

Builds - Protects - Insulates - Decorates - Subdues Noise

Building Board
Lath
Sheathing Board
Finish Plank
Tile Board
Adhesives
Batten Strips
Hard Board
Tempered Hard Board
Black Tempered Hard Board
Tempered Concrete Form Board
Panel Board
Studio Board
Hardboard Tile (Tempered)
Roof Insulation
Vaporproofed Low Temperature Insulation
Ferox Insulating Protection Course
Insulation Blocks
Ornaments and Mouldings
C-X Utility Board
C-X Texbord
C-X Wallboards
C-X Rock Wool Products

Celotex Cane Fibre Insulation

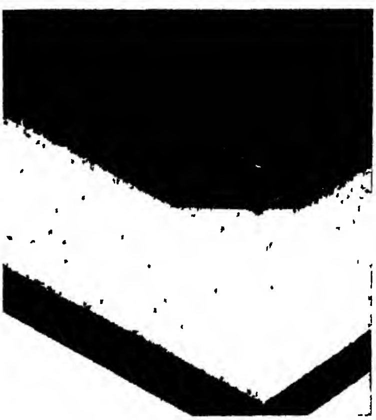
In the manufacture of Celotex, long tough fibres of bagasse (cane) are properly refined thoroughly sterilized, effectively waterproofed, firmly felted and securely interwoven to produce large boards of maximum strength consistent with the light weight necessary to assure high heat retarding value Careful technical control assures uniform high quality

The thermal conductivity of Celotex is 0 33 Btu per hour per square foot per 1 degree Fahrenheit per inch thickness (based on a mean temperature of 70 F) Tests conducted at Armour Institute of Technology and at recognized laboratories confirm this figure

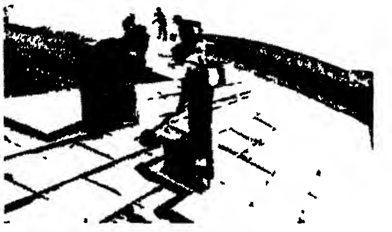
The Celotex Corporation maintains an Engineering and Research staff which is available for all types of insulation investigations Engineers are invited to address their problems to The Celotex Corporation, Chicago, Ill

Celotex Vaporseal Insulating Sheathing

A new sheathing for use on outside walls under any type of exterior. Designed to meet the advancement of modern building construction All surfaces and edges moisture proofed with a coating of special asphalt One side additionally treated with a bright aluminum compound as a vaporseal—this side is applied facing the studs and interior Coated on the surfaces—not integral—no impregnation—thus maintaining full insulating efficiency . Dry Rot and Termite Proofed by the exclusive Ferox Process (patented) . . . Greater rigidity and bracing strength than ever before Backed by Celotex written 10 POINT Life-of-Building Guarantee Same thickness as the wood sheathing it replaces—1 in —S2S to 25/32 4 ft wide; 7 ft, 8 ft, 8½ ft, 9 ft, 9½ ft, 10 ft and 12 ft long.



Celotex Roof Insulation



Selected to insulate wood concrete, steel, unit tile or poured gypsum roof decks under average conditions Size 22 in x 47 in Full ½ in thick Also furnished laminated in thicknesses up to 4 in Used to prevent ceiling condensation and conserve fuel also to prevent excessive expansion and contraction of concrete roof decks Reduces size of heating plant required

Celotex Vaporproofed Low Temperature Insulation

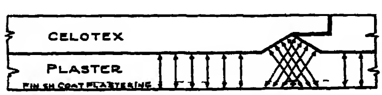
Waterproofed and vaporproofed low density insulation for low temperature requirements Coolers (beer, meat, creamery, etc.) Fruit and Vegetable Storage Rooms, Packing Plants, Fur Storage, Air-conditioned Spaces General Cold Storage Rooms and Freezers Each block enrobed in a sealed odorless membrane Ferox Treated Conductivity 0 30 Btu per inch Sizes 18 in x 18 in 18 in x 36 in, 9 in x 36 in Thicknesses 1 in, 1½ in, 2 in or any multiple of ½ in

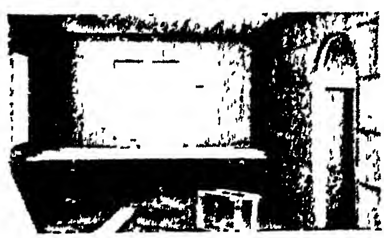
Celotex Building Board

The original cane fibre insulation Suitable for use as sheathing or interior finishing Can be beveled, paneled, grooved or painted to provide attractive walls that insulate Neutral tan color Double surfaced—one side smooth sanded, the other a tapestry-like texture ½ in and 1 in thick Sizes 4 ft wide and 4 ft, 5 ft, 6 ft, 7 ft, 8 ft, 8½ ft, 9 ft, 9½ ft, 10 ft and 12 ft long Backed by Celotex written 10 point Life-of-Building Guarantee

Celotex Insulating Lath

A natural bond for plaster a continuous surface eliminates lath marks and reduces cracking beveled edges to reinforce plaster (patented) tests prove a bonding power of 1,000 lb per square foot Shiplapped joints (see diagram) Size 18 in. x 48 in, ½ in, ¾ in and 1 in thicknesses Backed by Celotex written 10 point Life-of-Building Guarantee





Celotex Insulating Tile and Finish Plank

Tile Board—For attractive wall and ceiling treatments . may be applied over existing plaster or as a new finish Neutral tan color or several pleasing tints on reverse side that simplify decoration Type Double A joint permits alternated surfaces if desired Type A, furnished ½ in, ¾ in and 1 in thick, has beveled square edges, otherwise the same as Type Double A Type Double A joint ½ in thick only, ¾ in. and 1 in thick on special order Sizes range from 6 in x 6 in to 24 in x 48 in Backed by Celotex written 10 point Life-of-Building Guarantee

C-X Rock Wool Products

Effective wall-thick insulation made from molten rock Incombustible, vermin proof and permanent Batts—15 in x 23 in, 1⅓ lb per square foot Loose and Granulated to spread between ceiling joists

The Ferox Process

All Celotex Cane Fibre Products are manufactured under the exclusive Ferox Process (patented) and therefore effectively resist damage by Fungus Growth, Dry Rot and Termites (White Ants) It is not a surface treatment—it is integral—insoluble in water—non-volatile—odorless—permanent.

Ehret Magnesia Manufacturing Co.

Valley Forge, Pa.

DURANT PRE-SEALED INSULATED PIPE SYSTEM

The Modern Way to Insulate and Protect Underground and Outdoor Piping. (Hot and Cold)

Explanation

Durant Pre-Sealed Insulated Pipe Systems have been in successful use in the far west for fifteen years A great many installations have been operating under a variety of conditions which have proven beyond any doubt that the system is thoroughly reliable The absolute protection of the pipe and insulation against moisture and soil conditions has solved the problem which has bothered engineers

The Ehret Magnesia Mfg Co obtained an exclusive control of the manufacturing and sales rights A new department was built at Valley Forge and equipped with

the most modern equipment to properly produce Durant Pre-Sealed Insulated Pipe This department is now in active production and orders of any size can be efficiently and expeditiously handled

The most carefully designed and constructed systems for protecting underground piping have frequently caused serious trouble due to leakages with not only great impairment to the insulation and the pipe but to loss of efficiency in the operation of the lines The same has often been the case with outdoor pipe lines exposed to the elements Here the old method of wrapping on roofing materials has usually been ineffective, for no matter how securely applied the action of wind and weather eventually causes cracks and looseness with resulting damage to the covering and pipe

The Durant Pre-Sealed Insulated Pipe system completely overcomes these difficulties in an effective and permanent manner Not only did the Durant Pre-Sealed Insulated Pipe system overcome the engineering difficulties of protecting the pipe lines but it greatly simplified the installation as the sections of treated pipe are delivered to the job as entirely complete units, only joint and fitting connections being necessary in the field

Description

Solid Asphalt Shell—A special grade of pure high melting point asphalt is melted and cast around the pipe, or insulation, which results in a seamless and jointless coating which will be forever a permanent resistance to the entrance of moisture or infiltering air Jackets accurately spaced around the pipe, or insulation, insure uniform thickness of the asphalt layer These jackets are left in place and protect the surface from mechanical damage in shipping and handling

Thickness of Asphalt—The layer of asphalt can be applied in any thickness. For insulated piping in underground conditions a 1-in thick coat of the asphalt is recommended, and ordinarily supplied For insulated piping with outdoor conditions a thickness of ½-in is usually sufficient Where uninsulated piping is to be protected from the corrosive action of the soil and from electrolysis there should be a surface seal consisting of a ½-in thickness of the asphalt

Perfectly Sealed Connections—The method of treating joint and fitting connections, as described on the next page, actually extends the asphalt shell along the entire pipe system in an absolutely seamless manner

Any Kind, Size and Length of Pipe—There is no limit in size or length of pipe which can be given the Durant treatment and any specified type of pipe can be used However, ordinary stock lengths will be supplied unless special lengths are ordered Curved or bent pipe can be treated in the same manner as straight pipe

Insulation Material—Engineers recognize the durability of 85 per cent Magnesia and that it is a yardstick of thermal efficiency because it is practical both for low temperature piping and for piping having temperatures up to 600 F Therefore Ehret has selected 85 per cent Magnesia as the ideal product to incorporate in the Durant Pre-Sealed Insulated Pipe System Thickness recommendations of the 85 per cent Magnesia will depend on the service and the conditions

Discussion

The reasons for the effectiveness of the Durant Pre-Sealed Insulated Pipe method in protecting pipe and insulation are obvious Asphalt is well known to be absolutely impervious to the passage of moisture or air through it The dense, jointless casing of asphalt thoroughly and permanently prevents water and air from penetrating through This not only insures against deterioration of the pipe and insulation, thus preventing losses in the line, but the perfect surface air seal actually increases the overall efficiency

The asphalt protection will not crack from the pressure of the soil and it is sufficiently ductile to follow any possible movements of the pipe line

There are no cemented joints or seams which might crack open and leak

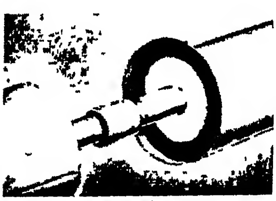
The insulation and the protective coating are applied at the factory under most favorable conditions This is in great contrast to the difficulty experienced in separately installing the pipe insulation and surrounding protection in the field This is especially true for underground installations where the work must be done at the bottoms of narrow, deep and often muddy trenches with questionable results

Durant Pre-Sealed Insulated Pipe is delivered to the job in complete pipe lengths (approximately 20-ft sections The only work to be done in the field is to make the joint and fitting connections

For underground work the trench can be narrower and shallower than for a conduit or tunnel for the same size pipe The running length of the trench need only be wide enough to accommodate the treated pipe Only at connections need any greater width be used to allow for a man to work in The Durant Pre-Sealed Insulated Pipe rests directly on the bottom of the trench and no greater depth is necessary to allow for crushed stone and underdrain tile The back filling can be accomplished in one operation as the soil can be settled into place by flooding with water This eliminates the necessity of the men returning at a later time to add earth to the sunken surface

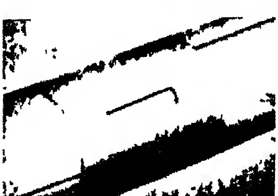
Installation

After the connections are made and tested, insulation is applied of the same type and thickness as on the adjacent pipe after which the necessary jackets are placed in position and the melted asphalt poured in The following illustrations indicate the simple manner in which this is done



STEP ONE
Field joint ready for inspection

STEP TWO
Joint covered with standard pipe insulation



STEP THREE
Special Durant joint casing in place ready for Asphalt

STEP FOUR
Asphalt poured in slot—a perfect seal

The Eagle-Picher Lead Company

General Offices: Temple Bar Building, **Cincinnati, Ohio**

Offices in All Large Cities

EAGLE INDUSTRIAL INSULATION

The Eagle-Picher Lead Company manufactures a complete line of industrial insulation materials, effective for a complete range of temperatures Representative products are

Eagle Super "66" Plastic Insulation

For application on practically all forms of heat producing and heat transferring equipment "Springy ball" construction provides remarkable heat-saving efficiency up to 1800 F Easily applied with a trowel May be applied on any clean surfaces Great coverage—60 sq. ft 1 in thick per 100 lb. 100 per cent reclaimable.

Eagle Blanket Insulation

Eagle "66" quickly applied with a trowel on hot and cold surfaces

Eagle Insulating Wool felted and secured between metal fabrics For large surfaces where temperatures reach as high as 1200 F. Available in flexible or rigid form Easy to cut and fit. Sizes—2 ft x 4 ft and 2 ft x 8 ft. Thicknesses range between 1 in and 8 in.

Eagle Asbestos Cements

Inexpensive asbestos fibre cements of good mixing, trowelling and adhesive properties For temperatures up to 1000 F

Eagle Insulseal (Waterproofing Cement)

Durable coating designed to protect all kinds of insulation from moisture, water, air infiltration, fumes, from vibration, abrasion Ready mixed, easily and quickly applied

Other Products

Eagle "99" Finishing Cement, Hair Felt, Pipe Covering and Blocks (all types), Insulating Wool, Boiler Setting Cement.

Data and Specifications

For complete specifications and technical data on Eagle Industrial Insulation, see Sweet's Engineering or Power Plant Catalog

EAGLE HOME INSULATION

The Eagle-Picher Lead Company manufactures two types of "mineral wool" insulation for homes (1) in granulated form for pneumatic application in new and existing construction (2) in bat form for new construction

Both types are wall-thick, non-structural, extremely light-weight, non-corrosive, fire-proof—with thermal conductance (in applied thickness of 3⅝ in) of only 0 074 Btu (The over-all conductance would be considerably lower)

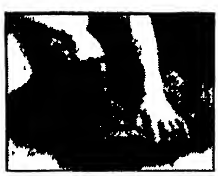
Eagle Insulating Wool easily blown into spaces between wall studdings

Eagle Home Insulation keeps homes cooler in summer (12 to 15 degrees cooler than outdoor temperatures), and warmer in winter (fuel saving ranges between 10 and 40 per cent)

For Pneumatic Application

In granulated form, Eagle Insulation is blown into hollow spaces between wall studdings and between joists in the attic floor by a special pneumatic process No building alterations are necessary, whether the house is of frame, brick or stucco construction. No mussing up inside. Work is done by skilled contractors, licensed by Eagle-Picher

Bat Form



Eagle wall-thick bats quickly installed in new construction

Eagle Insulating Bats are rectangular pads 15 in x 18 in or 23 in x 3⅝ in , designed to fit snugly between studding and joists Irregular spaces around doors and windows filled by cutting bats to exact size

Data and Specifications

For complete specifications and technical data on both types of Eagle Home Insulation, see Sweet's Architectural Catalog

International Fibre Board Limited

Sales Offices
OTTAWA—MONTREAL—TORONTO—WINNIPEG
Administrative Offices and Mills GATINEAU, QUE

London Office
THE TENTEST FIBRE BOARD CO LTD
ASTOR HOUSE, ALDWYCH, LONDON, W. C 2, ENGLAND

TEN/TEST
INSULATING WALL BOARD

TEN/TEST is a manufactured lumber made from spruce fibres, solidly pressed under hydraulic pressure into a strong, homogeneous board The fibres are chemically treated and water-proofed during process of manufacture, until the insulation is non-hygroscopic, free from capillary attraction and moisture-resisting in service commensurate with the maximum degree of insulation obtainable

Official Tests

Conductivity. TEN/TEST has a conductivity of 0 33 B t u. per hour per square foot per degree fahr per 1 in thick Authority. Professor E A Allcut, M Sc M I Mech E Mem A S M E Professor of Applied Mechanics, University of Toronto Tests performed by Hot-Plate method. Mean temperature 47 8 deg

Tensile Strength 228 lb. per sq in. Tests made on 7/16 in board cut to strips 1 in. wide and tested in a Riehle Tensile Testing Machine, the grips being 2 in. apart 228 lb is the mean average of seven series of tests.

Transverse Strength (equal deflection) is 28.4 lb. Test made on 7/16 in. board, 6 in. wide, 18 in. long, on 12 in. centers, and load being applied to breaking point.

Plaster Bonding Strength 2163 lb per sq. ft. Brown and scratch plaster coats were applied to standard 7/16 in board, and the pull registered in an Olsen Testing Machine Authority: Columbia University Testing Laboratories, New York.

Moisture Resisting. TEN/TEST, after complete immersion in water for 24 hours, registered 37 5% increase in weight

NOTE.—Authority for tensile strength, transverse and moisture tests; J T. Donald & Co., Ltd., Chemical Analysts and Engineers, Montreal, Que.

TEN/TEST Products

TEN/TEST Insulating Building Board. Standard insulation for use as exterior sheathing, interior finish, between walls and under floors for sound deadening Standard Industrial Insulation for refrigeration and the prevention of condensation Manufactured in convenient sizes: 4 ft wide and up to 17 ft long, 1/2 in. to 2 in thick

TEN/TEST Notch Board Plaster Base. Insulating plaster base having tongue and groove interlocking joints. Provides an effective bond with plaster without use of metal lath at joints. Sizes. 16 in wide, 32 in. and 47 3/4 in. long Thicknesses from 1/2 in to 2 in

TEN/TEST Roof Board. An effective roof insulation Manufactured in two sizes 1 x 4 ft. and 2 x 4 ft Thicknesses from 1/2 in to 2 in.

TEN/TEST Ashlar Block (Acousti "A") For interior decoration and acoustical correction. Absorbs 35 per cent of incident sound at a frequency of 512. Can be supplied in a variety of designs and sizes to harmonize with any decorative treatment, allowing the architect much freedom in design and finish of churches, auditoriums, theatres, etc. Ashlar Blocks have bevelled edges, standard or to suit, can be left in the natural color or tinted as desired.

TEN/TEST Moulded and Shiplap Edge Wall Panels. Conceals joints and provides excellent decorative treatment. Featured in widths of 11 in. to 47 3/8 in, lengths up to 12 ft.

TEN/TEST Mouldings. An effective trim and finish for joints, corners, etc Available in widths of 3/4 in to 10 in and lengths up to 12 ft

HYDRO/TEST. Water proof, insulating building board, designed particularly for low temperature requirements

The Insulite Company

Executive Offices· **Minneapolis, Minnesota**

Factories·
International Falls
Minnesota
Kymi, Finland

INSULITE
The Original Wood-Fiber Insulating Board

Stocked by Dealers in All Principal Cities Throughout The World

Insulite Wood Fiber Insulation Board Products

Insulite, the original wood fiber insulation board has been specified by engineers and architects for 23 years—used for exterior sheathing or interior finish, duct lining, and for other thermal insulation and sound control work Surface textures fine screen one side and burlap one side

Ins-Light Building Board—This board is of a natural light color with high light-reflecting value Thermal Conductivity 0 33 Btu/hour/square foot/inch/° F based on a density of 16 lb per cubic foot Thickness ½ in, ¾ in, 1 in, sizes up to 4 ft x 12 ft

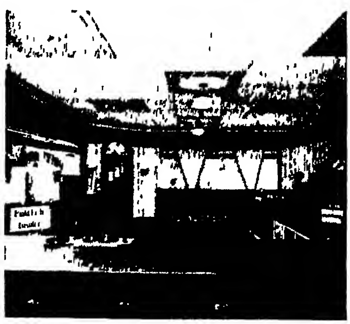
Graylite Building Board—Made from the same wood fibers as Ins-Light, but during the manufacturing process each fiber is coated with asphalt which adds to its strength and moisture resisting qualities Thermal conductivity 0 35 Btu/hour/square foot/inch/°F

Ins-Light Lok-Joint Lath—An insulating plaster base, fabricated from Ins-Light Building Board Has patented "Lok" that firmly locks the sheets together between supporting members, thus providing a rigid, level base for the plaster Thicknesses ½ in, ¾ in, 1 in Size 18 in x 48 in

Tile—Available in both Ins-Light and Graylite with burlap and fine screen textured surfaces Tile is beveled on all four edges of both surfaces, and is available in two types of joint B-B (Butt joint), V-W (interlocking) Both types are reversible, making it possible to reverse the tile and expose either surface Thicknesses ½ in, ¾ in, 1 in, in sizes 6 in x 6 in to 24 in x 48 in

Plank—Available in both Ins-Light and Graylite with burlap and fine screen textured surfaces Plank has the interlocking V-W joint and is beveled both long edges on each side Available either with or without beaded edges Plank is reversible so that either surface may be exposed Thicknesses ½ in, ¾ in, 1 in, in sizes 6 in to 16 in wide and 6 to 12 ft long

ABOVE *Insulite Building Board, applied with metal washers and screws being used to insulate air ducts in air conditioning installations*



BELOW *Ins-Light and Graylite Tile and Plank serves as the interior finish in a modern sales room*

Bildrite Sheathing

Bildrite Sheathing, although made from the same wood fiber stock used in other Insulite products, is an outstanding, new building material development Treated by a patented integral asphalt treatment during the manufacturing process, it provides several times the bracing strength, greater resistance to moisture, and more insulation than that afforded by ordinary wood sheathing, and it builds a knot and crack free wall at less cost To assure rapid and proper application, each large sheet 25/32 in thick, and in sizes up to 4 ft x 12 ft, is plainly marked for proper alignment and spacing of nails Thermal conductivity 0 36 Btu/hour/square foot/inch/° F

Ins-Light Roof Insulation

Ins-Light Insulation is made from the same wood fiber as standard Insulite It can be used over any roof construction The standard size is 22 in x 47 in , for convenient handling, and with either offset or square edges Furnished in thicknesses of ½ in , 1 in , 1½ in , and 2 in Conductivity 0 32 Btu/hour/square foot inch/° F

Graylite Roof Insulation

Graylite Roof Insulation is a new and improved product, exclusive with Insulite Treated with an asphalt emulsion during the process of manufacture, it provides those qualities of greater resistance to moisture and greater durability so desirable in roof insulation, combined with an ideal base for bonding to the roof deck and to the roofing Furnished in full ½ in thickness and in multiples of ½ in up to 2 in with either square or offset edges Size 22 in x 47 in

Insulite Hardboard Products

Tough, durable grainless boards with hard, smooth, surfaces in various densities, and sizes up to 4 ft x 12 ft

DualBoard—¼ in thick, golden oak brown, has lowest density of all HardBoard products

DeLuxe DualBoard—¼ in thick with smoother surface and greater density than DualBoard

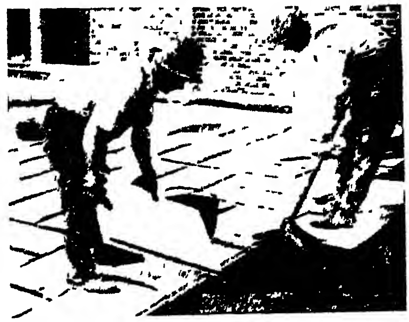
HardBoard—1/10 in , ⅛ in , 3/16 in , ¼ in , 5/16 in , golden oak color, much greater density than DualBoard

Tempered HardBoard—Same thicknesses as HardBoard, burl walnut color, extremely high density

PanelTile—Same as Tempered HardBoard except that it is scored in 4 in squares to simulate tile

Ins-Light Cold Storage Insulation

Cold Storage Insulation is a low density Insulite product which has found a wide acceptance for use in refrigerator cabinets, refrigerator cars, ice houses, storage plants, breweries and other places where low temperatures are maintained The thermal conductivity is 0 29 Btu/hour/square foot/inch/° F and is available in eleven standard sizes and in thicknesses of 1 in , 1½ in , 2 in , 3 in , and 4 in



Applying Insulite Roof Insulation

Insulite Sealdslab

Fabricated from special low density board each block of Sealdslab is "sealed dry" with a 3/32 in impregnation of a specially developed asphalt which insures an effective seal against moisture absorption after Sealdslab is asphalt coated or dipped during application on the job The factory primed surfaces also insure an excellent bond for subsequent asphalt coatings Available in same sizes and thicknesses as Ins-Light Cold Storage Insulation Sealdslab is especially adapted for use in freezers,meat, beer and creamery coolers, milk cooling tanks, fruit and vegetable storage rooms and general cold storage rooms

Insulite Fiberock

Insulite Fiberock insulation is a rock wool product, treated for moisture resistance, with a low percentage of shot and with sufficient resiliency to prevent it from settling Conductivity 0 26 Btu hour/square foot/inch/°F Available in three forms.

Loose Fiberock—A fluffed form of rock wool insulation for hand packing in walls Furnished in 35 lb bags

Granulated Fiberock—A granular form of rock wool which can be conveniently poured into place Usually used over ceilings Supplied in 35 lb bags

Fiberock Bats—Rock wool in bats 15 in x 23 in to fit snugly between studs and joists The bats are wall thick and require no special fastening Packed in cartons of 9 bats each

Johns-Manville

Executive Offices

22 East 40th Street, New York, N. Y.

Offices in All Large Cities

Johns-Manville Home Insulation

J-M Home Insulation consists of special grades of rock wool for retarding heat flow through walls, ceilings and floors of frame houses and all types of buildings.

J-M Home Insulation is remarkably effective in providing year-around comfort—keeping homes up to 15 deg cooler in summer—reducing fuel bills up to 30 per cent in winter. It is permanent, fireproof, and will not support vermin. Furnished in three forms: Type A for existing construction; Types B and C for new homes.

Blown Method—Type A

Type A Rock Wool is blown pneumatically into the spaces between studs in outer walls and between rafters or joists in roofs or attic floors. Insulation thickness in walls corresponds to stud depth, approximately 3⅝ in., density does not exceed 10 lb per cubic foot. This type of insulation is installed by J-M Approved Home Insulation Contractors, who are equipped with the necessary apparatus and trained crews.

Home Insulation—Types B and C

Type B Home Insulation is furnished in the form of resilient prefabricated batts, of uniform density and full stud thickness, in sizes 15 in. x 23 in. and 15 in. x 48 in., designed to fill completely the width between studs, joists or rafters spaced 16 in. center to center.

Applying J-M Home Insulation batts in new home

Heavy waterproof paper, affixed to the batt surface toward the occupied portion of the building, provides a barrier to infiltration of moisture-laden air into the Rock Wool. The side-flanges, fastened to the supporting framework, hold the batts firmly in place in ceiling or roof construction.

Insulation in odd-shaped spaces is secured either by application of pieces cut from the batt, or by the use of Type C, which is furnished in pieces 8 in. x 15 in., full stud thickness, without the waterproof paper backing.

Write for Details

For complete information on J-M Home Insulation write for Brochure HI-18A.

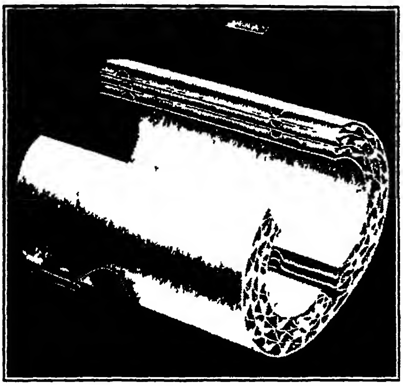
J-M Insulating Board and Insulating Lath

Johns-Manville furnishes a complete line of Insulating Board, Insulating Lath and Roof Insulation Board in standard sizes and thicknesses. These materials are thoroughly efficient, with high insulating value and great structural rigidity.

J-M Air-Acoustic Sheets for Lining Air-Conditioning Ducts

J-M Air-Acoustic Sheets, for duct linings of air conditioning systems, are fireproof, highly sound-absorbent, moisture-resistant, with a surface which will not materially increase friction losses in the duct system. Write for DS Series 275.

Johns-Manville Pipe and Boiler Insulation



J-M Pre-Shrunk Asbestocel Pipe Insulation

J-M Pre-Shrunk Asbestocel Pipe Insulation

J-M Pre-Shrunk Asbestocel is a radically improved insulating material for hot water or low pressure steam piping, which, since it is made of moisture-proofed asbestos paper, offers positive protection against shrinkage troubles

Supplied in canvas asbestos paper or aluminum finishes All types furnished in 3-ft sections in standard thicknesses of 2 to 8 plies, each ply approximately $\frac{1}{4}$ in thick

J-M 85% Magnesia

Recommended as the most efficient insulation of the molded type for temperatures up to 600 F. Pipe insulation is furnished in sectional or segmental form for all commercial pipe sizes, in thicknesses up to 3 in Blocks are 3 in by 18 in and 6 in by 36 in., flat or curved, from ½ in to 4 in thick

J-M Pre-Shrunk Wool Felt

Due to its Dual-Service Liner—an asphalt-saturated felt—J-M Pre-Shrunk Wool Felt is equally effective and durable on either hot or cold water service piping By the use of waterproofed felts shrinkage troubles have been minimized

Supplied in two finishes, the regular canvas and a smooth, dull-coated aluminum In either finish, it is furnished in 3-ft sections in thicknesses of ½ in ¾ in, 1 in, Double ½ in, and Double ¾ in, for pipe sizes from ½ in to 5 in. Can also be supplied in sections to fit straight runs of copper pipe or tubing with the following outside diameters $\frac{3}{8}$ in, $\frac{1}{2}$ in $\frac{5}{8}$ in, $\frac{7}{8}$ in $1\frac{1}{8}$ in $1\frac{3}{8}$ in $1\frac{5}{8}$ in, $2\frac{1}{8}$ in, $2\frac{5}{8}$ in, $3\frac{1}{8}$ in, $3\frac{5}{8}$ in, $4\frac{1}{8}$ in, $5\frac{1}{8}$ in and $6\frac{1}{8}$ in

J-M Asbesto-Sponge Felted

Recommended on all high pressure steam piping at temperatures up to 700 F where insulation may be subjected to rough usage or where maximum efficiency and durability are desired Furnished in 3-ft sections up to 3 in thick

J-M Superex Combination

Superex Combination Insulation (an inner layer of high temperature Superex and an outer layer of 85% Magnesia) is recommended where temperatures exceed 600 F. Superex and Magnesia are both furnished in sectional, segmental, pipe covering and block forms

J-M Improved Asbestocel Sheets and Blocks

Asbestocel Sheets and Blocks are used for insulating warm-air ducts flues, heater casings and fan housings in the ventilating system Temperature limit 300 F Furnished 6, 9, 12, 18 and 36 in wide by 36 and 72 in long, from $\frac{1}{2}$ in to 4 in thick

J-M Rock Cork Sheets and Pipe Insulation

J-M Rock Cork is made of rock wool and a moisture-proof binding ingredient molded into sheets for insulating refrigerated rooms and air conditioning ducts, and into sectional pipe insulation with an integral waterproof jacket for all low temperature service It is strong, durable, and will not support vermin Because of its unusual moisture resistance, its high insulating efficiency is maintained indefinitely

Furnished in sheets 18 in by 36 in, in $1\frac{1}{2}$, 2, 3 and 4 in thicknesses, also 18 in by 18 in by 1 in thick In pipe covering form in ice water, brine and heavy brine thicknesses for all commercial pipe sizes

Details on Request

Write for complete information on any Johns-Manville insulating material

Mundet Cork Corp.

450 Seventh Avenue — New York, N. Y.

Manufacturers of Corkboard, Cork Pipe Covering, Compressed Machinery Isolation Cork, Natural Cork Isolation Mats, Cork Tile, Cork Bulletin Board, and all kinds and varieties of Cork Specialties.

Branches

Atlanta Ga — Chicago, Ill — Cincinnati, Ohio — Dallas, Texas — Des Moines, Iowa — Detroit, Mich — Houston, Texas — Kansas City, Kans — Los Angeles Calif — Memphis, Tenn — New Orleans La — No Cambridge (Boston), Mass — Philadelphia, Pa — St Louis, Mo — San Francisco, Calif

Agents

Baltimore Md	The McCormick Asbestos Co	Oklahoma City, Okla	Standard Roofing & Material Co
Buffalo, N Y	Clayton Asbestos Co	Portland, Oregon	F J Leonard
Charlotte, N C	C R Howard	Portland, Oregon	Pacific Asbestos & Supply Co
Cleveland Ohio	C S Ross	Salt Lake City, Utah	Louis A Roser
Fredericksburg Va	Fredericksburg Insulation Co	Seattle, Wash	Pioneer Sand & Gravel Co
Hartford Conn	The Hartford Cement Co	Tulsa, Okla	Standard Roofing & Material Co
Nashville Tenn	John Bouchard & Sons Co	Utica, N Y	George Weisenberger
Norfolk Va	F H Gaskins Corp		

Engineering and Specification Service

Our engineering department is at the service of Architects and Engineers at all times to assist and advise in the preparation of specifications pertaining to cork This service is also available to any one who has a cold insulation or a vibration isolation problem, and is rendered without obligation. Our complete catalogue is filed in Sweet's Architectural Catalogue, and will be sent on request It is replete with valuable information and data that should always be within reach of every specification writer whose field touches our products

Contract Service

We contract for the erection of our products In this way we may be certain that our material is installed in accordance with best established practice This gives a definite advantage to an owner, in that divided responsibility for a given installation is eliminated No contract involving cork is too large, too small or too far away All materials and workmanship are unqualifiedly guaranteed

Mundet "Jointite" Corkboard

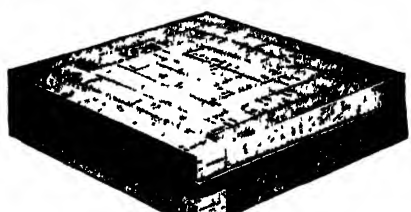
Mundet "Jointite" Corkboard is 100 per cent pure cork, fabricated in accordance with the U. S Government Master Specification, and is unsurpassed in its field It is used for all cold insulation services and for acoustical correction. We manufacture only one grade of corkboard. Mundet "Jointite" Corkboard is sold in the standard 12 in x 36 in. sheet. Standard thicknesses are ½ in., 1 in , 1½ in., 2 in , 3 in , 4 in and 6 in.

Mundet "Jointite" Cork Pipe Covering

Mundet "Jointite" Cork Pipe Covering is the complement of Mundet "Jointite" Corkboard and is used for all types of cold lines. The three thicknesses in which it is manufactured make it suitable for pipes carrying sub-zero to 50 F temperature The pipe covering comes in sections 36 in., long. A complete line of standard fitting covers is available in the three thicknesses

Mundet Cork Vibration Isolation

The transmission of machine vibration can be easily and permanently prevented by the use of Mundet cork isolation The machines commonly associated with the heating and ventilating industry are best isolated with Mundet Natural Cork Isolation Mat. This form of isolation is fabricated from blocks of pure cork These blocks are held together within a rigid steel frame or bound with asphalt paper applied with hot asphalt top and bottom Steel bound isolation mat is



Above is shown a Steel Bound Mundet Natural Cork Isolation Mat Note the natural cork strips within the steel frame

usually used under exposed mounts, and asphalt paper bound isolation mat, under concrete foundations, of the envelope type Mats are constructed to fit under any type of machine foundation

For loads exceeding 2,000 lb per square foot, we manufacture Mundet Machinery Isolation Cork, which is a board form of compressed granulated cork and comes in three densities.

Both types of isolation are furnished in 1 in., 1½ in , 2 in , 3 in , 4 in. and 6 in thicknesses, depending on the class of service

The Pacific Lumber Company

PALCO WOOL INSULATION

100 Bush Street
San Francisco

59 E Van Buren St
Chicago

700 So. Broadway
Los Angeles

122 East 42nd St
New York

WHAT IT IS

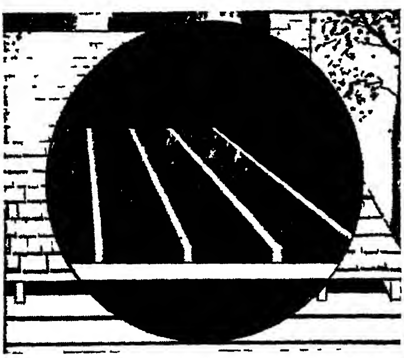
PALCO WOOL is a loose fill insulating material made from the bark of the Redwood tree, the protective covering of the world's oldest living thing It is highly refined into an insulating material of light weight wiry fibres of springy resilience Recent improvements in manufacturing have made it clean, dustless and lighter in weight In practical use PALCO WOOL has proven to be ideal for all types of construction, large or small, where resistance to conduction of heat is required It is continuously efficient and reasonably priced, thus assuring economical performance

USES

PALCO WOOL is suitable for any type of domestic or commercial construction, in fact every place where an insulating material is required to effectively resist the transmission of heat.

INSTALLATION

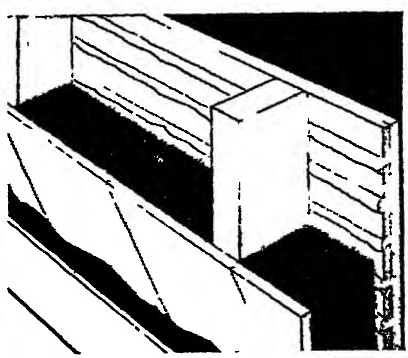
Approximately 5½ lbs of PALCO WOOL are required for each cubic foot of space in walls, floors, ceiling and partitions It is easily put in place by hand Between 150 and 200 lbs can be applied per hour per man It comes in bales weighing approximately 100 lbs Size 22″ x 24″ x 26″



Data Folder and Sample on Request

8 PROPERTIES that make it AN IDEAL INSULATION

1. **Thermal Efficiency:** The established conductivity of PALCO WOOL is 258 B t u per one inch thickness per sq ft per hour per degree F. difference in temperature by the Flat Plate Method
2. **Non-Settling:** The fibres of PALCO WOOL possess such resilience that no settlement in a wall can occur under the most severe conditions of vibration
3. **Moisture Resistant:** The fibres of PALCO WOOL are entirely lacking in capillarity, and have little attraction for moisture, enabling it to remain dry and efficient when in use
4. **Permanent:** The inherent antiseptic qualities of PALCO WOOL make the existence of fungus impossible The fibres retain their resilience indefinitely
5. **Vermin Proof:** PALCO WOOL is distasteful and repellent to rodents and insects
6. **Fire Resistant:** PALCO WOOL will not readily support combustion and is fire resistant
7. **Odor Proof:** PALCO WOOL is odorless itself and does not absorb or give off odors
8. **Economical:** PALCO WOOL is light in weight and low in density, offering exceptional thermal efficiency per dollar invested



Reynolds Corporation

Executive Offices 19 Rector Street, **New York City**

Offices in Principal Cities

REYNOLDS *METALLATION

Description

Metallation consists of bright sheet aluminum cemented to one or both sides of tough kraft paper Available in roll form, in the form of Metallated *Ecod (a reinforced plaster base); and with heavy Sisalkraft paper base These forms facilitate installation according to construction conditions and amount of insulation required

Metallation is primarily insulation material for building construction involving hollow wall, floor or roof sections Data on special applications on request Its insulating effectiveness depends upon its influence on the enclosed air spaces it bounds or forms, not upon its thickness Hence, it must always face or divide an air space to have insulating value Because it is windproof and moisture-proof, it always serves as an effective weatherproofing barrier in any building section where used.

How It Insulates

Metallation insulates by repelling practically all radiant heat which strikes its surface This portion of the total heat loss ordinarily flows unimpeded across air spaces Type A or B Metallation dividing an ordinary air space reduces heat loss by convection, by dividing the air space into two air spaces and stops 95 per cent of the radiant heat loss across such air spaces Metallation has equal effectiveness on either side of an air space, in vertical walls regardless of the direction of the heat flow

Characteristics

Metallation is waterproof, windproof, vermin-proof and non-absorptive It is light in weight and flexible, yet stiff enough to conform to angles and curves when not under tension or suspended Its heat storage capacity is negligible, due to its light weight and limited mass The aluminum used retains its reflectivity under all normal conditions, as the surface is protected by a transparent oxide which forms immediately on exposure Tests recently conducted by Gordon B Wilkes, Massachusetts Institute of Technology, show that visual brightness is not essential to good performance

The value of Metallation as an insulation is unlikely to be impaired by internal condensation within the building section Some external walls and roofs standing between cold exterior air and warm humidified air indoors develop a "dew-point" temperature within them where condensation forms Any insulating material that absorbs this moisture loses a part of its insulating value until it dries out again Metallation being non-absorptive and uneffected by moisture, retains its full value.

Type A—Bright sheet aluminum cemented to both sides of tough kraft paper with 1¼ in plain paper margins exposed for nailing inside framing members Applications curtains between framing members 16 in o c **Type B**—Bright sheet aluminum cemented edge to edge to both sides of tough kraft paper Application curtains in air spaces Also one surface as insulation, the other as moisture-proof membrane **Type C**—Bright sheet aluminum cemented edge to edge to one side of tough kraft paper Application. one side as insulation, the other in contact with building materials **Metallated Ecod**—Bright sheet aluminum cemented to back of reinforcing plaster base

See Sweet's Catalog for complete details. Used by U S Government and leading architects and engineers Stocked by leading dealers throughout U S

Type	Width of Roll	Length of Roll	Weight of Roll	Area of Roll
A	17 in.	176 ft 5 in	12¼ lb net	250 sq ft
B	25 in	120 ft	14 lb net	250 sq ft
B	33 in	90 ft 11 in	14 lb net	250 sq ft
B	36 in	83 ft 4 in	14 lb net	250 sq ft
C	25 in	120 ft	10½ lb net	250 sq ft
C	33 in	90 ft 11 in	10½ lb net	250 sq ft
C	36 in	83 ft 4 in	10½ lb net	250 sq ft

*Metallation, Metallated Ecod and Ecod are trade-marks of Reynolds Corp, duly registered in U S Pat Off

NOTES FOR HEAT TRANSFERENCE TABLE on opposite page

[a]Computed by standard methods from data in THE A S H V E GUIDE, Vol 14, pp 104-111, 1936

[b]All air spaces are at least ¾ in wide

[c]The thickness of 1 in yellow pine sheathing or flooring is approximately 25/32 in

[d]Plaster ⅝ in to ¾ in thick is applied to ECOD or expanded metal plaster base

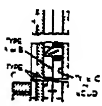
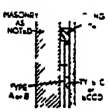
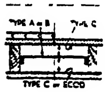
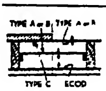
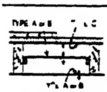
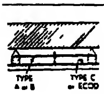
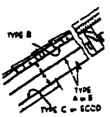
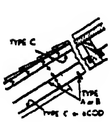
[e]Four inches of face brick and four inches of common brick

[f]Nailing strips measure approximately 25/32 by 3⅝ in and are spaced 2 in apart

[g]Furring wide enough to permit ¾ in air spaces between layers

COEFFICIENTS OF TRANSMISSION OF VARIOUS TYPES OF CONSTRUCTION INSULATED WITH REYNOLDS METALLATED ECOD AND TYPE A OR B METALLATION

Coefficients (U) are expressed in Btu per hour per square foot per degree Fahrenheit difference in temperature between air on the two sides of the construction

Illustration	Construction Nos	Construction	Uninsulated	I Metallated Ecod plaster base ¾ in plaster	II 1 layer Type A or B Metallation dividing air space into 2 air spaces Metallated Ecod plaster base	III 2 layer Type A or B Metallation dividing air spaces into 3 air spaces Metallated Ecod plaster base
			U	U	U	U
		**FRAME OR VENEER WALL**				
	1	Shingles or clapboards, sheathing, stud space, plastered	0.26	0.20	0.13	0.10
	2	1 in Stucco with Ecod base, stud space, plastered	0.50	0.31	0.18	0.12
	3	1 in Stucco with Ecod base, sheathing, stud space, plastered	0.32	0.23	0.15	0.11
	4	4 in Face brick veneer sheathing, stud space, plastered	0.29	0.21	0.14	0.10
		**MASONRY WALLS**				
	5	8 in Solid brick, furred space, plastered	0.32	0.23	0.15a	0.11a
	6	12 in Hollow concrete blocks, furred space, plastered	0.32	0.23	0.15a	0.11a
 		**ATTIC FLOOR (Frame Structure)**				
	7	Plaster ceiling-joist space no flooring	0.69	0.59	0.24	0.15
	8	Plaster ceiling-joist space-rough flooring	0.30	0.22	0.14	0.11
		**MASONRY FLOORS (Suspended Ceiling)**				
	9	4 in Concrete metal lath and plaster ceiling	0.37	0.25	0.16	0.11
	10	6 in Concrete metal lath and plaster ceiling	0.35	0.24	0.15	0.11
		**CONCRETE ROOF DECK** 3/8 in B. U. Roofing Suspended Ceiling Metal lath and plaster				
	11	4 in Concrete	0.40	0.26	0.16	0.12
	12	6 in Concrete	0.37	0.25	0.16	0.11
 		**FRAME ROOFS**				
	13	Wood shingles, nailing strips, rafter space no ceiling, exposed rafters	0.46	0.22	0.14	0.11
	14	Wood shingles, nailing strips, rafter space, metal lath and plaster	0.30	0.22	0.14	0.11
	15	Asphalt shingles, rigid asbestos shingles, composition roofing or slate or tile roofing on wood sheathing no ceiling	0.56	0.24	0.15	0.11
	16	Same as No 15 plus metal lath and plaster ceiling	0.34	0.24	0.15	0.11

The Ruberoid Co.

Executive Offices

500 Fifth Avenue, **New York, N. Y.**

Divisional Offices

NEW YORK CHICAGO BOSTON (MILLIS) ERIE BALTIMORE MOBILE

RUBEROID RESIDENTIAL INSULATING PRODUCTS

Genuine RU-BER-OID Mineral Wool

From the viewpoint of efficiency Mineral Wool is one of the finest insulating materials for residential construction It can be used in varying thicknesses up to 4 in It cannot deteriorate or decompose, and is fire- and vermin-proof This insulation can be expected to save 20 to 35 per cent in fuel bills in the winter, and make the interior of the home 10 to 15 deg cooler in summer

Genuine RU-BER-OID Mineral Wool in the loose or bulk form is clean, silky and long fibred, a quality which overcomes the tendency to pack and settle It is free from harsh brittle particles This wool is offered in three forms—*loose* or bulk for packing, *granulated* for pouring, and in *pre-formed bats* for use between joists, rafters, and studding

The long silky fibres make this wool extremely stable, overcoming the tendency to pack and settle

Loose Mineral Wool

The loose mineral wool is packed in 35 lb bags containing about 3½ cu ft A bag will cover in excess of 21 sq ft gross (including the studs or joists) 3⅝ in thick when packed to a service density of 6 lb per cubic foot

Granulated Mineral Wool

Granulated Mineral Wool is in pellet form averaging from the size of a pea to a marble In this form it is easily poured between studs and joists and for filling in irregular spaces Bags contain 30 lb When applied to a service insulation density of about 6 lb per cubic foot, 4 in thick, it gives a coverage of about 18 sq ft per bag

Mineral Wool Bats

Mineral Wool Bats are packed in cartons containing 7 pieces, 16 75 sq ft The bats are wall thickness (approx 4 in.) measuring 15 in x 23 in which permits easy application between the studding of sidewalls, or between the rafters under the roof

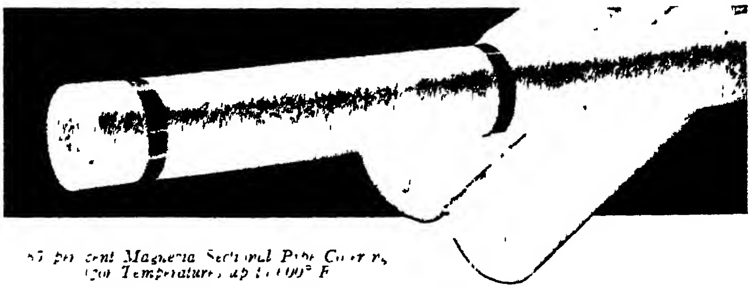
RUBEROID INDUSTRIAL INSULATING PRODUCTS

85 Per Cent Magnesia for Medium and High Pressure Steam Lines

85 per cent Magnesia pipe coverings are made of approximately 85 per cent pure carbonate of magnesia and 15 per cent carded long Asbestos fibre Temperature limit 600° F. Light in weight, fire-proof, extremely high insulating value, yet have maximum mechanical strength consistent with efficient insulation. Supplied in various thicknesses up to 3 in , canvas jacketed

For Higher Temperatures MINERAL WOOL BLANKETS Limit 1800° F

Mineral Wool for flat or curved surfaces is manufactured into blankets by using annealed long fibre wool, felting it and building it up to the required thickness



85 per cent Magnesia Sectional Pipe Coverings, for Temperatures up to 600° F

between metal fabrics of various types These blankets are secured with galvanized wire between the metal fabrics Thickness from 1 in to 6 in Standard size blankets 24 in x 96 in and 24 in x 48 in

High Temperature Pipe Coverings Limit 1600° F

This insulation is recommended for insulating surfaces having temperatures between 600° F and 1600° F It has low thermal conductivity, practically no shrinkage, is light in weight, and has good mechanical strength

Supercell Pipe Covering for Low and Medium Pressure Steam Lines Limit 350° F

Supercell Pipe Insulation—A new development in the low and medium pressure field, has a temperature limit of 350° F It has 14 to 16 laminations of indented Asbestos Felt to the inch thickness It is more durable than the old-fashioned cellular pipe insulation, and, due to its unique construction, provides as much as 35 per cent higher efficiency

Other Insulating Materials for Low and Medium Temperature Steam Lines—Air Cell—Limit 350° F

Made 2, 3 and 4 ply—each ply ¼ in thick Furnished in either the new Pyroxylin finish or with canvas jacket Temperature limit 350° F Ideal for industrial hot water lines.

Watco Cell Limit 350° F

Close corrugation, 6 and 8 ply to the inch, raises efficiency standards Either new Pyroxylin finish or canvas jacketed, as desired Temperature limit 350° F

Woolfelt For Hot and Cold Water Lines

Special surfacing process increases efficiency 20 per cent to 30 per cent Weight also is reduced 20 per cent to 30 per cent Made in three styles of liners Asbestos Felt Paper for hot water lines, Waterproofed Tar for cold water lines, "Twin Purpose" for both hot and cold water lines Furnished in either the new Pyroxylin finish or with canvas jacket

Sheet and Block Insulations

Any of the pipe coverings can be made into sheet and block form Standard sizes 6, 12, 18 or 36 in x 36 in Ideal for insulation of flat or irregular surfaces, such as tanks, boilers, breechings, etc.

Insulating Cements

For the finishing of asbestos sheet and block insulation, or the insulation of valves, fittings, or flanges, or any sort of irregular surfaces, the line of Ruberoid-Watson Insulating Cements is quite complete Will take care of temperature conditions from 100 to 2000 deg F.

Detailed Specifications

Complete catalog giving specification data of any Ruberoid Insulating Product may be had upon request

Silvercote Products, Inc.

161 E. Erie Street, **Chicago, Ill.**

SILVERCOTE
Reflective **INSULATION**

SILVERCOTE INSULATION

Fabric—Manufactured from two external sheets coated with a mineral pigment and polished to a highly reflective surface possessing the essential characteristics hereinafter described for all Silvercote surfaces The sheets are laminated with specially prepared asphaltum and reinforced with an interlining of fabric, imbedded in the lamination, consisting of $\frac{1}{2}$ in mesh of strong jute cords Two silver-like surfaces are exposed The material is strong, flexible, and can be creased, folded, and tucked into the most inaccessible places without injury Weight 80 lb per 1000 sq ft

Coreboard—Laminated of 7-plies of pulp board. 16 point liners coated on the exposed sides and polished to highly reflective surfaces Finished board is $\frac{3}{8}$ in thick with 2 silver-like surfaces exposed Weight 1000 lb per 1000 sq ft

Insulation Board—Manufactured in a manner similar to wood fibre wallboard, to which is applied a top liner coated with pigment and polished to a highly reflective surface. Reverse side has a manila liner, dead level, beater sized and tacky, a perfect surface for painting Finished board is $\frac{3}{16}$ in thick Weight 550 lb per 1000 sq ft

SILVERCOTE SURFACES

The surfaces of all Silvercote products consists of a non-metallic, homogeneous pigment, polished into a silver-like sheen which reflects radiant heat to a marked degree They possess the following essential properties

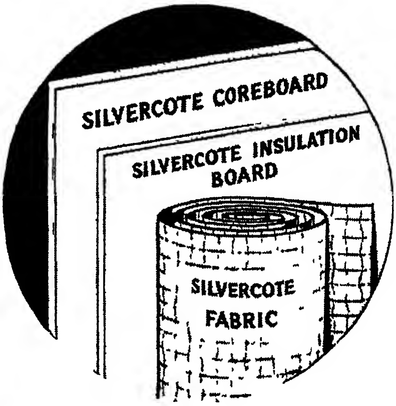
1. The surfaces are not metallic and are consequently free from oxidation to which all metal surfaces (except gold and platinum) are subject.
2. The surfaces are waterproof
3. The surfaces are highly moisture resistant, equal in this respect to sheet lead
4. The surfaces are not affected by acids or gases encountered in the customary uses of thermal insulation, whether in building construction, refrigeration or the cold storage fields.
5. The surfaces are entirely homogeneous and impervious to air infiltration wherever used
6. The surfaces are non-conductors of electric and thermal energy and are designed to establish a difference in surface temperatures on transverse surfaces in the direction of heat flow

Silvercote Insulation Fabric-Conductance
0 33 = Resistance 3 03

Silvercote Insulation Board-Conductance
0 49 = Resistance 2 04

Silvercote Insulation Coreboard-Conductance
0 265 = Resistance 3 77

For complete specifications and technical data see Sweet's Catalog, or address Silvercote Products, Inc, Kalamazoo. Mich



The Standard Lime & Stone Company

First National Bank Building, **Baltimore, Md.**

Manufacturers of Capitol Rock Wool Insulations — **Representatives in all Principal Cities**

Capitol Rock Wool gets its low coefficient of thermal conductivity not alone from the long-fibre method of manufacture, but also due to the fact that it is made only from a specially quarried rock of high silica and low sulphur contents This results in a uniform coefficient of conductivity of only 0 240, (amount of heat in Btu which will flow in one hour through a square foot of the material, if the temperature drop through the material is one degree F per inch of thickness)

Blowing Existing Houses Trained workmen simply remove a few shingles a piece of clapboard or a few bricks for insertion of the hose nozzle into the wall For new construction or unfinished rooms Capitol Rock Wool in Batt form Note how completely the insulation fills the space between the studding

Applications

While applicable for almost all types of insulation Capitol Rock Wool is particularly used for building and home insulation When applied by either the blowing method to finished homes or buildings, or in batt form during construction, it becomes a permanent, integral part of the structure, does not rot or corrode and provides a fire-proof "blanket" of exceptional insulating efficiency

Installations

For new construction or for unfinished wall-spaces, Capitol Rock Wool is available in batts 15 in x 23 in, fitting easily and snugly between 2 in x 4 in studding on 16 in or 24 in centers Weight 6 lb per cubic foot

For insulating side walls and roof spaces of finished construction, the manufacturers have highly trained blowing contractors everywhere Even the best insulating material must be properly installed to insure insulation efficiency and installation economy

Results Obtained

Experience in insulating all types of new or old construction, has demonstrated the value of this insulation material and method In summer, the house insulated with Capital Rock Wool is 8 deg to 15 deg cooler than before insulation, the third floor having but 2 deg or 3 deg variation above the ground floor In winter, temperature variation between different floors and rooms is practically eliminated, doing away with the cause of drafts

Industrial Insulations

Blankets—A reinforced felted Capitol Rock Wool designed primarily for insulating boiler settings, fire walls, boiler breechings, smoke stacks within buildings, tank cars, hot water heaters, boilers, oil stills, bubble towers, large diameter pipes, ducts, etc , and for absorbing sounds set up by machinery

Insulating Blocks—A compressed pliable product for temperatures up to 750 F, for insulating domestic furnaces, boilers, and hot water tanks, and for commercial boilers, tanks, ducts, stills and breechings

Pipe Coverings—With canvas or waterproof cover for temperatures from 60 F below zero to 1250 F.

Insulating Cement—Plastic, quickly and neatly applied, available in both high and low temperature cements.

CATALOGS and Specifications of our complete line of home and industrial insulations will be sent on request.

United States Gypsum Company

300 W Adams Street, Chicago, Ill.

Sales Offices

Atlanta Ga, Boston, Mass, Buffalo, N Y, Cincinnati, Ohio, Cleveland, Ohio, Dallas Texas, Denver, Colo, Detroit, Mich, Indianapolis, Ind, Kansas City, Mo, Los Angeles, Calif, Milwaukee, Wis, Minneapolis Minn, New York N Y, Omaha, Neb, Philadelphia Pa, Pittsburgh Pa, St Louis Mo, San Francisco Calif, Washington, D C

PRODUCTS

Insulating Building Board
Insulating Lath
Metal Reinforced Insulating Lath
Insulating Tile
Insulating Plank
Insulating Mouldings
Roof Insulation
Tongue and Groove Sheathing
Strip Wool
Bat Wool
Junior Bat Wool
Granulated Wool

WEATHERWOOD BUILDING BOARD, SHEATHING LATH, PLANK AND TILE

Weatherwood Insulating Board is a felted wood fiber product that has been treated to make it highly moisture resistant It is formed on a single cylinder which produces a homogeneous board, free from laminations and tendencies to split through the center

Low Thermal Conductivity

The average established in tests is 0 33 Btu's per hour, per square foot per inch thickness, per degree Fahrenheit difference in temperature

Resistance to Moisture—Non-Absorption

Tests show that this board, after being submerged in water for a period of two (2) hours, has an average water absorption of less than 15 per cent

Structural Strength

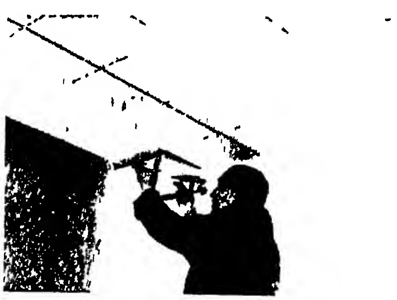
The tensile strength is over 350 lb per square inch, and the modulus of rupture, over 500 lb

Durability

There is nothing in this board that will deteriorate prematurely and impair the original insulation value Since the board is homogeneous, the fibers being interwoven through the sheet, it will not split The Moisture Resistant chemical with which the fibers are treated makes the board distinctly distasteful to rodents and insects

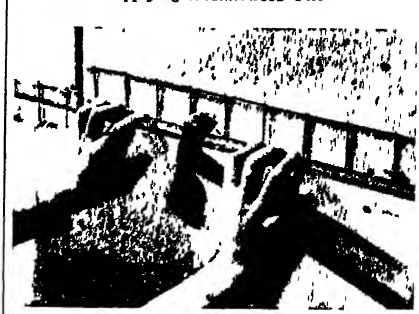
Uniformity

Adequate laboratory control and mill inspection assure uniformity in density and structural strength



Applying Weatherwood Tile

Erecting Weatherwood Insulating Board Easy to handle Applied by nailing



Weatherwood Reinforced Insulating Lath

RED TOP INSULATING WOOL

Description

Red Top Insulating Wool is an extremely light, fluffy wool insulation made of silica—a fireproof material

The nature of the raw materials used, particularly as to purity, permits accurate manufacturing control—the product is uniform and consists of snowy white, long-fiber wool It contains no "shot" or non-insulating materials that add weight Red Top Wool is springy and resilient—it will not mat

Low Thermal Conductivity

The heat conductivity of Red Top Insullating Wool (1-½ lb density) is 0 266 Btu's per inch thickness, per square foot per hour, per degree Fahrenheit difference in temperatures (Tests by Professor Peebles, Armour Institute of Technology)

Unusually high in insulating efficiency, it is outstanding when, as customarily used, it is wall thick (4 in) So used its rating is 0 066

Light Weight (Density)

In its standard density, Red Top Insulating Wool weighs but 1-½ lb per cubic foot

Types

The types of Red Top Insulating Wool are illustrated in this page, showing its adaptability

Applying Red Top Strip Wool between studs Strip Wool has a waterproof paper backing with a ½ in flange for nailing to studding Used as shown it prevents the entrance of moisture into the insulation

Method of applying Red Top Bat Wool between rafters or studding

Applying Red Top Strip Wool between attic joists

Red Top Junior Bats are easily and quickly installed

Coverage of Red Top Insulating Wool

Type of Wool	COVERAGE OF RED TOP INSULATING WOOL		
	Volume per package at standard density	Coverage per package at standard density	
		4 in thick including 2 x 4's	4 in thick excluding 2 x 4's
Strip Wool in cartons	11¼ cu ft	36 sq ft	33¾ sq ft
Bat Wool in cartons	12½ cu ft (20 bats)	40 sq ft	37½ sq ft
Bulk Wool in paper bags	6⅔ cu ft	21½ sq ft	20 sq ft

The Zonolite Company

General Offices 5905 Second Blvd., **Detroit, Mich.**

Plant **Libby, Montana**

ZONOLITE
TRADE MARK REGISTERED

ZONOLITE INSULATION FOR HOMES

Zonolite is a micaceous, non-metallic mineral insulation (aluminum magnesium silicate) expanded many times through a special process by the application of intense heat. It insulates both by its multiple dead-air cell construction and by its bright reflective surfaces. Zonolite offers "three-way" heat control, for it is an efficient, *permanent* barrier to the passage of heat by *conduction*, *convection* or *radiation*.

The density of Zonolite is uniform, tamper-proof. Its insulating value is the same in the walls or on the ceiling as when it left the factory. Zonolite cannot be "fluffed" or stretched when being installed; it flows evenly, filling every nook and corner with a "tailor-made" insulation of high efficiency.

An Efficient Heat Barrier

Believing that the efficiency of any home insulation must be determined under conditions of actual use, with such factors as type of building construction, radiant heat travel, wind movement and the uniformity of insulation density taken into consideration, The Zonolite Company is helping to pioneer the "*U*" factor (overall coefficient of transmission) as the only satisfactory, accurate standard of insulation value.

One such "*U*" factor test, made at the College of the City of New York, showed that Zonolite, installed in a standard wall section, reduced the heat loss through that section by 73.1 per cent. Another significant test was made in Detroit. "Twin" houses, identical in every important particular, were erected; one was left uninsulated, the other insulated with Zonolite. Accurate records through the winter of 1935-36 proved that Zonolite reduced fuel bills 38 per cent ... held temperature fluctuation within 2 deg. as contrasted with a fluctuation of as much as 18 deg. in the uninsulated house.

Zonolite Meets Every Insulation Requirement

Zonolite Insulation is ALL-MINERAL—ROTPROOF and FIREPROOF, VERMIN-PROOF—rats, mice, termites or insect larvae will not eat it or nest in it, SOUND ABSORBING—tests by V. O. Knudsen, acoustical authority, show Zonolite to be high in sound absorption and sound insulation, LIGHT IN WEIGHT—approximately 6 lb. per cubic foot. DIELECTRIC—a test by J. C. Peebles, Armour Institute of Technology, showed that 20,800 volts were required to puncture a one-inch layer of Zonolite between one-inch brass balls, SAFE TO HANDLE—no danger of silicosis, lead poisoning, sores or boils, CHEMICALLY INERT—tests of every description prove Zonolite will not deteriorate in any way or affect any material with which it comes in contact, ODORLESS—will not absorb or give off odors under any conditions. Zonolite forms a THICK insulation, is approved by leading air conditioning firms for use with their units.

Economical to Install

Zonolite is easily installed in attics and walls by blowing or pouring directly from the bag. Whichever method is used Zonolite fills completely, with an unchanging, uniform density. Your local Zonolite representative will furnish accurate estimates of installation costs.

Zonolite is packed in 3⅓ and 4 cu. ft. bags. One 4 cu. ft. bag will cover 17½ sq. ft., 3 in. deep, a 3⅓ cu. ft. bag will cover 14½ sq. ft., 3 in. deep.

ZONOLITE-ASPHALT ROOF BLOCK

Both a durable roofing, weatherproof throughout its thickness, and an efficient insulation (Thermal conductivity 0 38 for the minimum applied thickness of 1½ in) Not penetrated or affected by hot moppings, insulating value is the same in application as in laboratory Elastic will take up expansion, contraction, or uneveness in the roof deck without cracking or forcing out the asphalt between the blocks

Economical to Apply. Three plies of roofing felt are usually ample for any job No special precautions, either before or during application, are necessary to protect the insulation from the weather Rotproof, waterproof, vermin-proof and highly fire resistant, Zonolite-Asphalt Roof Block is an ideal insulator for all types of roof decks It is available in 12 in x24in blocks of any desired thickness from 1½ in up

ZONOLITE AIR DUCTS

(Patented)

Sound Absorbing and Insulated

Entirely new in both concept and design, Zonolite Air Ducts are formed of heavy, specially-designed wire mesh Directly to this mesh is applied Zonolite Insulating Cement—a thermal and sound insulator of great efficiency No inside lining is necessary The result is a duct which prevents condensation, minimizes drag, and absorbs sound at the rate of 1 5 decibels per foot of duct (Duct size 9½ in x 11½ in)

Zonolite Air Ducts (Patented) permit increased air velocity with no increase in noise, thereby allowing the use of a smaller duct to carry a specified volume of air They are absolutely vermin-proof, rotproof, and fireproof, inside and out Will withstand excessive vibration and high air pressures Made exclusively by local sheet metal shops licensed by The Zonolite Company

ZONOLITE ACOUSTICAL PLASTER

A carefully prepared mixture using Zonolite, the all-mineral sound and heat insulation, as a base Can be applied on either old or new walls, at a cost which is extremely low for an efficient acoustical material Extremely light in weight, easy

Zonolite Acoustical and Insulating Plasters used in beautiful new Gesu Chapel, Detroit

to work and apply It has proved its ability to correct the acoustics of the most difficult rooms, and to provide a high measure of thermal insulation at the same time Its color, when dry, is an attractive golden tan May be tinted to harmonize with any decorative scheme

ZONOLITE INSULATING PLASTER

Insulates and Plasters in one Job

An all-mineral plaster, extremely light in weight and uniform Insulates effectively against passage of heat and transmission of sound Fire resistant, easy to handle and spread The elastic nature of the Zonolite ingredient permits expansion and contraction after application, thus adding permanence and strain resistance to the plaster, and reducing cracking By applying Zonolite Insulating Plaster as brown and scratch coats over any plaster base, an unusually fine foundation for finish coats is provided

ZONOLITE INSULATING CEMENT

A 100 per cent mineral product, Zonolite Insulating Cement is ideal for insulating tanks, boilers, etc , where the insulation must stand temperatures up to approximately 1000 deg Its base is Zonolite, which accounts for its light weight and unusual efficiency Has pronounced sound absorbing qualities as well as high thermal insulating properties

Zonolite Insulating Cement adheres well to any clean metal, concrete or tile surface Can be reclaimed simply by adding water

---

Special cements are also available for industrial furnaces and other applications where temperatures run higher than 1000 deg

Write for Full Details, Specifications and Name of Local Representative

H. W. Porter & Co.

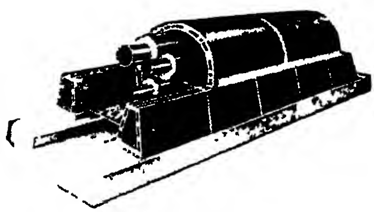
INCORPORATED

Newark, New Jersey

BALTIMORE WASHINGTON RICHMOND CHARLOTTE

Therm-O-Tile REG US PAT OFF STEAM CONDUIT SYSTEMS

Description—Therm-O-Tile is a complete conduit system for the permanent support, protection, and insulation of underground pipe lines The tile is made 6 in to 24 in in diameter and with five different size base tiles, producing twenty-seven different conduit cross sections



Foundation—The base of the system is a thick concrete slab poured directly in the trench bottom, reinforced with steel when installed over a filled or boggy ground

Drainage—The drainage system of the conduit is entirely internal, open to inspection at manholes, and of ample capacity to keep the pipe space dry at all times without any possibility of becoming clogged with silt or vegetation

Pipe Support—All pipes are supported on cast-iron adjustable supports resting directly on the concrete base independent of the tile envelope

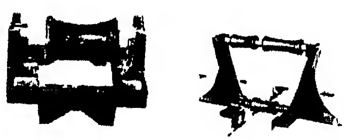
Accessibility—All piping is installed before tile is placed, giving complete accessibility for welding, testing and insulation Pipe fitters work on concrete slab walkway

Strength—Due to immovable concrete base and arch construction of extra heavy tile members, conduit will sustain any roadway traffic load usually encountered without extra reinforcement.

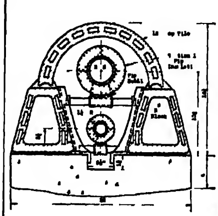
Insulation—In this conduit, either sectional pipe covering or Thermobestos fibre filling may be used for insulation, as the insulation space is kept dry at all times, by the internal drain For single or double pipe lines, sectional insulation of economical thickness is recommended, for multiple pipe lines, a filler type of insulation is usually more economical in first cost

Waterproofing— Under normal soil conditions, this conduit is waterproof If marshy ground or partially submerged conditions are encountered, the conduit may be made completely waterproofed by the use of membrane waterproofing applied under the slab on a sub-base and carried completely over the tile envelope

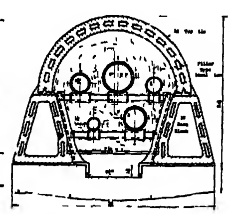
Efficiency—The degree of thermal efficiency secured depends upon the type and thickness of insulation used This conduit, due to its sealed air chambers in the tile and dry insulation space, adds to the normal efficiency of the insulating material on the pipe lines



Pipe Supports



Single or Double Pipe Lines Using Sectional Pipe Insulation



Multiple Pipe Lines Using Filler Type Insulation

Therm-O-Tile is also sold and installed locally by Johns-Manville Construction Units

The Ric-wiL Company

Agents in Principal Cities

RIC-WIL
REG U S PAT OFF

ESTABLISHED IN 1910

CONDUIT SYSTEMS FOR UNDERGROUND STEAM PIPES

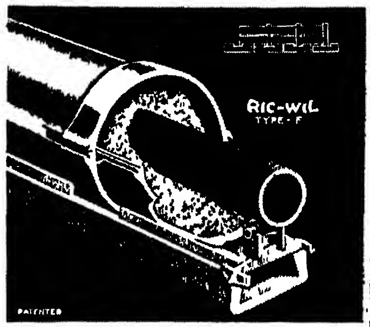
Union Trust Bldg. NEW YORK—CHICAGO—SAN FRANCISCO Cleveland, Ohio

Conduit—Standard conduit is vitrified salt glazed tile or cast-iron with Loc-liP Side Joints, bell and spigot type, lined or unlined Tile in 24 in sections, sizes 4 to 27 in inside diameter A Super-Tile conduit to support any average traffic load is also available For extra heavy duty under railroads or where conduit is subject to extreme loads, Ric-wiL is made of cast-iron in 2 or 4 ft sections Where reduced labor cost is not essential, Ric-wiL Universal Type System is recommended (details on request)

Base Drain—Standard Base Drain is vitrified salt glazed tile for tile conduit and extra heavy tile or cast-iron for the cast-iron conduit, in 24 in lengths Made in three sizes to support and drain **properly** all conduit sizes

Pipe Supports—Standard pipe supports will carry from one to five or more pipes and are ordinarily spaced 12 ft apart They are strong, made of cast-iron, rust-proofed, and interlock with the base drain foundation, imposing no load on the conduit itself No movement of pipes can disturb them Special pipe supports available for different conditions. Information on request

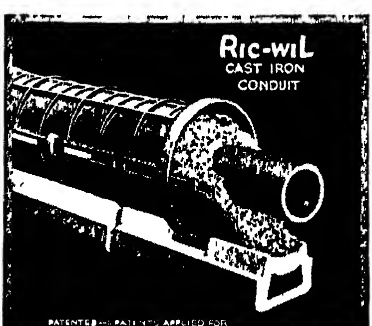
Insulation—Ric-wiL Dry-paC Waterproof Insulation is high-grade long fibre asbestos processed so that it is permanently water repellant Of unusually high efficiency and great natural strength, it will not slump away from pipes and is non-corrosive Sample will be sent gladly for testing Ric-wiL No 11 Insulation (same as Dry-paC except non-waterproof) or sectional pipe covering can also be furnished For lined conduit, diatomaceous earth mixture is molded and keyed inside the tile

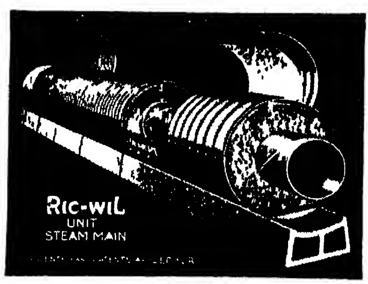


Accessories—Shutter sleeves, filter cloth, asphalt joint cement, water-proofing compound, manhole covers, and other accessories will be furnished as desired

Engineering Service—Full cooperation with architects and engineers Installation supervision if desired

Technical Data—Tabulated Steam Heating Rates, Test Reports, Service Detail Bulletins, Catalog Bulletins, Central and District Heating Bulletins and Architects and Engineers Detail Sheets available upon request





Ric-wiL Unit Steam Main, a prefabricated ready-to-install unit, 18½ ft long, including conduit (Armco Iron), pipe, insulation and accessories Ideal for speed and economy on district heating projects

Underground Steam Construction Co.

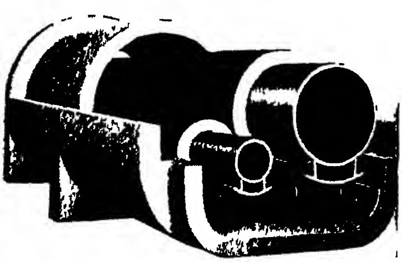
75 Pitts Street, **Boston, Mass.**

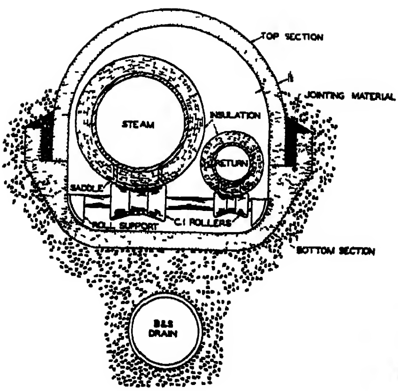
PRODUCTS—Engineering and Contracting of Steam Line Installations. Underground Steam Conduits.

USCCO PRE-CAST CONCRETE CONDUIT

This unit marks a long step forward in the economics and mechanics of laying underground steam lines

Its Outstanding Features Are:





Strength resulting from flat reinforced bell, from the longitudinal joints, and from the fact its sections are 4 feet long, reducing the usual number of joints necessary

Shape which allows 12 per cent greater inside capacity than circular conduit of like diameter. This permits larger pipes in a given size conduit and more room for drainage, pipe supports and rolls

Ease of Installation resulting from the fact that **all** top and bottom halves mate Bottoms may be laid for any distance, the piping installed and then the tops brought up and laid. This speeds up the work and protects idle halves from damage They can be stored away from the job

Materials are high strength cement with suitable aggregate, amply reinforced with wire mesh

Pipe Supports are of cast-iron and quickly installed They may be placed anywhere in the conduit The weight of the pipe holds them in place

Joints may be of any standard accepted brand of jointing compound or they may be of standard Portland cement mortar

Builders Iron Foundry

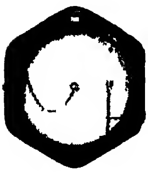
23 Codding Street **Providence, R. I.**

Representatives in Principal Cities

The CHRONOFLO METER for nearby or long distance measurement of flow, level, pressure, temperature, weight, or position.

The SHUNT METER for steam, air and gas

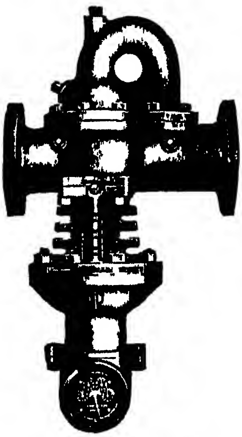
The VENTURI METER for boiler feed water supply, and other main pipe lines.



CHRONOFLO METERS AND CONTROLLERS

"From hundreds of feet to hundreds of miles"

Chronoflo Meters— *Mechanically* measure flow, pressure, temperature, or position in terms of time, *electrically* transmit this time duration equivalent of the quantity to any point—or points—nearby or miles away. There a Receiving Instrument *mechanically* indicates, records, or registers it. If desired, the same "impulse" may be used to operate flow, pressure, temperature, weight, or position controllers. Regular A C current is used and accuracy is not affected by voltage variation. Suitable for nearby or long distance transmission over private wires or public telephone channels. Write for Bulletin 283.



Shunt Steam Meter, Type K S

THE SHUNT METER

A low priced, practical, *mechanical*, meter easily installed and accurate over a wide range. Regularly used by many District Heating Companies for measuring steam sold, also ordered and re-ordered by leading industrial plants to meter steam or air furnished tenants, departments, and processes.

The illustration shows the complete meter. Installation consists simply of bolting to flanges in the flow line, no connecting pipe or electric wiring. A portion of the entering steam is deflected by an orifice through nozzles against the blades of a turbine located in the upper or shunt passageway. The speed of the turbine is kept low by a dampening fan on the vertical turbine shaft, at the bottom end of which is a magnetic drive to the totalizer dials.

The meter is installed as a unit in 2, 3 and 4-in lines, for larger capacities the installation is made in a by-pass around an orifice in the main line. Write for Bulletin 255 *and prices*

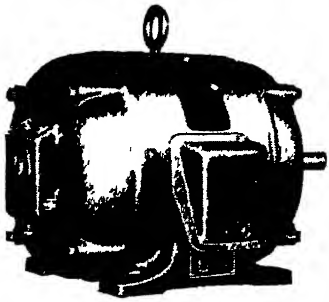
	Steam Pressure Lbs per Sq In Gauge	Rated Capacity of Saturated Steam—Lbs per Hour (Meters with largest orifice)						
		2" Meter	3" Meter	4" Meter	6" Meter	8" Meter	10" Meter	12" Meter
Low Pressure Meters	0	600	1,400	2,530	5,660	9,850	15,200	19,000
	5	800	1,900	3,350	6,500	11,300	17,500	21,800
	15	1200	2,750	4,750	7,860	13,700	21,100	26,300
	30	1650	3,400	5,850	9,550	16,600	25,600	31,900
	50	2000	4,050	7,000	11,400	19,800	30,500	38,000
Standard Meters	50	2650	6,000	10,600	22,200	39,000	60,500	75,500
	100	4150	9,500	16,600	29,000	51,000	79,200	99,000
	150	5000	11,400	20,000	34,400	60,500	94,000	117,000
	200	5700	13,000	22 700	39,000	68,600	107,000	133,000
	250	6300	14,400	25,200	43,200	76,000	118,000	148,000
Extra Heavy Meters	275	6600	15,200	26,500	45,000	79,400	123,000	154,000
	300	6900	15,700	27,500	46,800	82,500	128,000	160 000

Minimum Capacities equal one-tenth tabulated quantities. Many other smaller and intermediate capacities available

Century Electric Company

1806 Pine Street, St. Louis, Mo.

Offices and Stock Points in Principal Cities



MULTISPEED MOTORS 1/8 to 200 Horse Power

Press the Button—Change the Speed .

Especially adapted to meet the variable or constant speed requirements of Fans, Ventilators, Refrigerators and similar apparatus where adjustable speed change is a requirement Built for 2, 3, 4 or more speeds—automatic, push button or manual control Wide or narrow speed ranges, such as 1800/1200 or 1800/600 down to 900/450 rpm (60 cycle) Conventional open or splash proof type Special speed combinations are also available Ball Bearings or phosphor bronze sleeve bearings

CENTURY MOTORS

For Compressors, Pumps, Fans, Blowers, Refrigerators, Stokers, Oil Burners

Type of Motor	Horse Power Range	Starting Duty	Remarks	Applications
SINGLE PHASE MOTORS				
RS—Brush-Lifting	1/8 to 40	Heavy	Low Starting Current, High Starting Torque	Piston or Plunger Pumps, Refrigerators Stokers, Compressors etc
BR—Brush-Riding	1/8 to 3/4	Heavy	Short Annual Service Characteristics	
CPH—Cap Start and Run	1/8 to 10	Heavy	High Starting Torque	
CSH—Cap Start	1/8 to 3/4	Heavy	High Starting Torque	
CSN—Cap Start	1 to 10	Medium	High Starting Torque	Fans (Belted or Direct Connected) Centrifugal Pumps, etc
CPX—Cap Start and Run	1 to 10	Light	Must be Loaded to at Least 50 per cent Capacity	
SP—Split Phase	1/60 to 1/3	Medium	Unrestricted Starting Current, Long or Short Annual Service Characteristics	Oil Burners, Unit Heaters, Blowers, Fans Small Tools, etc
SP—Split Phase	1/60 to 1/3	Light	Restricted Starting Current, Long or Short Annual Service Characteristics	
POLYPHASE MOTORS				
SC—Squirrel Cage	1/8 to 600	Medium	Normal Starting Current Normal Torque	General Purpose Motors
SCN—Squirrel Cage	7 1/2 to 200	Medium	Lower Starting Current than SC Normal Torque	
SCH—Squirrel Cage	3 to 200	Heavy	Low Starting Current High Starting Torque	Refrigerators, Piston or Plunger Pumps, Compressors etc
AS—Automatic Start	1 to 60	Heavy	Lower Starting Current than SCH High Starting Torque	
SR—Slip Ring	1/2 to 200	Heavy	For Frequent Starting and/or Speed Control	Fans, Blowers, Centrifugal Pumps, Compressors etc
DIRECT CURRENT MOTORS				
DM-DN-R Shunt Wound Constant Speed	1/20 to 400		Torque is limited only by Commutation A Direct Current Motor has ample Torque to start any load that it can carry when up to speed Starting Current is limited by Controller to about 150 per cent of full load current for light starting torque requirements with corresponding increases in current for increased starting torque	Fans, Blowers, Centrifugal Pumps, Machine Tools etc
DM-DN-R Compound Wound Varying Speed	1/12 to 400			Reciprocating Pumps, Compressors and Machines with Flywheels, etc
DN-R Shunt Wound Adjustable Speed	1/2 to 100			Fans, Blowers, Machine Tools, etc

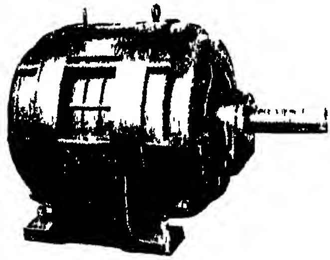
Quiet Starting—Quiet Running—Remarkably Free from Vibration—Keep Themselves Clean Inside—Easy to Keep Clean Outside—Harmonizing Appearance—Century Squirrel Cage Polyphase Motors are Especially Adapted to all types of Air Conditioning Equipment

The Ideal Electric & Mfg. Co.

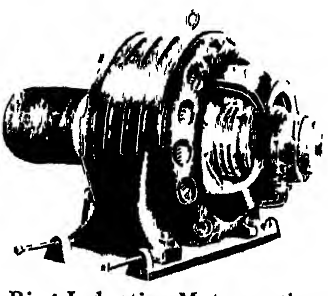
Mansfield, Ohio

Sales Offices in All Principal Cities **167 Service Shops in U S A**

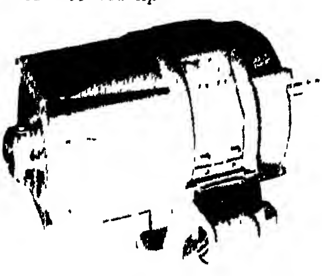
One of the largest independent motor manufacturers, Ideal makes more than 367 different types and styles of motors for all types of service in sizes from 1 to 3000 hp



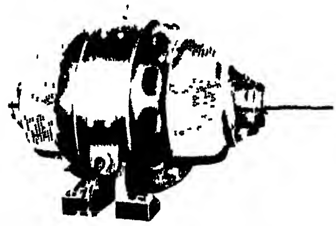
Squirrel Cage Induction Motors suitable for all general purpose requirements Made in all torque and inrush classifications, sizes 1 to 600 hp Available with 1, 2, 3, or 4 speeds, in standard open type, splashproof, drip proof, totally-enclosed fan-cooled, and pipe ventilated designs Also as 100 per cent or 80 per cent leading power factor induction motors for power factor correction



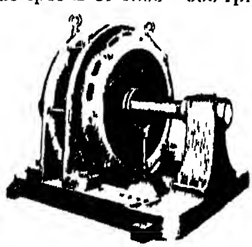
Slip Ring Induction Motors with speed adjustable to 50 per cent full speed values All sizes 1 to 600 hp



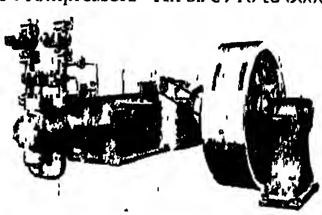
Direct Current Motors of all types and sizes 1 to 600 hp, constant or variable speed



Self-Excited Synchronous Motor for general purpose applications where a constant speed high efficiency A C motor is desired Available in all sizes from 3 to 100 hp at speeds of 1800—600 rpm



Ideal Synchronous Motors, engine and pedestal type for direct connection to low speed compressors All sizes 10 to 3000 hp



Ideal Flywheel Type Synchronous Motors for direct connection to compressors Their compactness enables mounting direct on the compressor in small sizes In large sizes an outboard pedestal bearing supports part of the rotor weight All sizes 15 to 700 hp

Free Engineering Service is available We will gladly make recommendations for the type of motor drive best adapted to any application if you will give the service requirements Literature is available covering all Ideal Equipment Write Customers' Service Division, c/o Ideal Electric & Mfg Company, Mansfield, Ohio

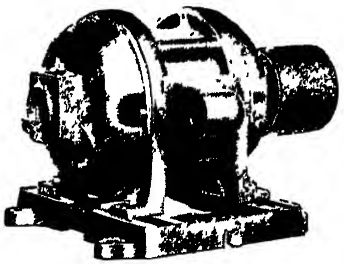
GENERAL ELECTRIC COMPANY

SCHENECTADY, N. Y.

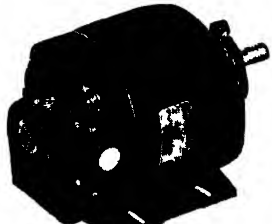
SALES OFFICES WAREHOUSES SERVICE SHOPS AND DISTRIBUTORS IN PRINCIPAL CITIES

For Code Wire, Conduit Products, Wiring Devices, Insulating Materials etc., Address—APPLIANCE AND MERCHANDISE DEPARTMENT, BRIDGEPORT, CONN

HEATING, VENTILATING, AND AIR-CONDITIONING MOTORS



Wound-rotor quiet-operating induction motor on sound-isolating base Type MB



Split-phase induction fractional horsepower motor Type KH

The complete line of motors manufactured by the General Electric Company offers you a motor with electrical and mechanical characteristics best adapted to your compressor, fan, or pump application The most frequently used applications are listed below Complete information on other types of motors, vertical, enclosed, etc , with various electrical and mechanical modifications, may be obtained from our nearest sales office

A complete line of motors, designed and tested especially for quiet operation for use in schools, hospitals, commercial buildings, and also a complete line of special sound-isolating bases for these motors are available

SOME G-E MOTORS AND THEIR USES

Application	Speed	Type Winding	Type	Horsepower Range	Classification
Fans and Centrifugal Pumps	Constant or Adjustable	Shunt	B & CD	1/8–200	Direct Current
Reciprocating Pumps and Compressors		Compound	B & CD	1/8–200	
Small Direct Connected Fans	Constant	Resistance Split Phase	KH	1/40–1/3	Single Phase Alternating Current
		Reactance Split Phase	KX	1/6–1/3	
	Constant or 3-Speed	Low Torque Capacitor	KC	1/50–10	
Belted Fans Centrifugal Pumps	Constant or 2-Speed	High Torque	KC	1/4–10	
Pumps Compressors Fans	Constant or Multispeed	Capacitor	KC	1/8–10	
		Repulsion Induction	SCR	1/8–10	
		Squirrel Cage	K or KB	1/4–1000	Polyphase Alternating Current
		(Low Starting Current)	KF	7½–75	
Reciprocating Pumps and Compressors		(High Starting Torque)	KG	3–100	
Pumps Compressors, Fans	Constant or Adjustable	Wound Rotor	M & MB	½–1000	
	Constant	Synchronous	TS	25–2000	

This Company will gladly assist in the solution of any electrical problems in relation to heating and ventilation

GENERAL ELECTRIC COMPANY

SCHENECTADY, N. Y.

SALES OFFICES, WAREHOUSES, SERVICE SHOPS AND DISTRIBUTORS IN PRINCIPAL CITIES

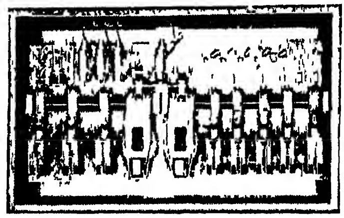
For Code Wire, Conduit Products, Wiring Devices, Insulating Materials, etc ,

Address—APPLIANCE AND MERCHANDISE DEPARTMENT, BRIDGEPORT, CONN

CONTROL FOR HEATING, VENTILATING, AND AIR-CONDITIONING MOTORS

The General Electric line of standard control offers manual or automatic equipment for compressors, fans, or pumps driven by any type motor which you require, providing full protection for your motor, especially those listed on the preceding page

For special applications General Electric controllers can be designed to meet your exact requirements



CR7107 controller (cover removed) for use with Squirrel cage motors

The following is a list of typical control equipment applicable to all motors listed on the preceding page

Full-voltage automatic starters with thermostatic control for fans, or pumps

Automatic reduced- or full-voltage starters for synchronous motors driving compressors

Manual or automatic speed-regulating controllers for wound-rotor motors driving fans

Manual full-voltage starters for pump motors

Manual speed-regulating switches for small capacitor motors driving fans

Capacitors for improving power-factor

ACCESSORIES

Electrically operated valves

Thermostats
Float switches
Pressure switches

Indicating Push buttons

CR 7006 —full voltage magnetic switch for use with induction motors

CR1061 fractional-horsepower - motor starting switch for wall mounting

CR-9507-C electrically operated valve

Motors and control of one manufacture insure perfect operation simplify installation and insure good service for the entire installation

This Company will gladly assist in the solution of any electrical problem in relation to heating and ventilation

See also pages 860 and 861

The American Brass Company

General Offices—Waterbury, Conn.

Manufacturing Plants

Ansonia Conn, Torrington, Conn Waterbury Conn Buffalo, N Y, Detroit, Mich Kenosha, Wis

Offices and Agencies

Boston Mass One Forty Federal St
Syracuse N Y, 207 E Genesee St
Providence, R I, 131 Dorrance St
New York, N Y, 25 Broadway
Newark, N J, 20 Branford Place
Philadelphia, Pa, 117 South 17th St

Washington, D C 1511 K St, N W
Atlanta, Ga 66 Luckie St
Pittsburgh, Pa, 1228 Brighton Road
Cleveland, Ohio, 2906 Chester Ave
Dayton, Ohio 32 North Main St
Cincinnati, Ohio, 101 West 4th St

Chicago, Ill, 1326 W Washington Blvd
St Louis, Mo, 408 Pine St
Houston, Texas, 609 Fannin St
San Francisco, Calif, 235 Montgomery St
Los Angeles, Calif, 411 West Fifth St

CANADIAN PLANT Anaconda American Brass Limited, New Toronto Ontario

PRODUCTS—Anaconda Deoxidized Copper Tubes and Fittings; Anaconda "85" Red-Brass Pipe; Everdur Metal for storage heaters, storage tanks, ducts and air conditioning equipment

ANACONDA
from mine to consumer

ANACONDA COPPER TUBES AND FITTINGS

For Heating, Plumbing and Air Conditioning

Anaconda Deoxidized Copper Water Tubes assembled with Anaconda Fittings offer an unusual combination of advantages in hot water heating systems at a cost only slightly higher than black iron and approximately the same as wrought iron pipe These advantages may briefly be summarized as follows

Complete Immunity to Rust—Copper heating lines can never rust through to leak, nor will they become clogged with rust deposits This means long service at a minimum of upkeep expense

Low Friction Loss—Because the inside surfaces of copper tubes do not become roughened by the formation of rust, these tubes offer a minimum resistance to flow In addition, the long radius turns of Anaconda Elbows and the smooth inside surface of Anaconda Wrought Copper Fittings further reduce friction losses

These factors naturally increase the efficiency of the system, particularly when it includes a forced pressure circulator

Low Heat Loss—The bright copper tubes radiate much less heat than black iron pipe of the same size

Ease of Installation—In many installations, a short section of soft copper tube can be installed between the lateral run and the radiator As these soft tubes can readily be bent by hand, this frequently eliminates several fittings In many other places, also, the flexibility of soft copper tubes simplifies connections that ordinarily would be awkward and expensive to make with rigid pipe and threaded fittings Anaconda Solder Fittings are compact They can be installed in constricted space where the use of a wrench would be impossible

Architects and builders naturally object to large holes and notches cut in the framing members of a building for the passage of piping Anaconda Copper Tubes can be installed with a minimum of cutting in the structure, although holes should be large enough to permit movement of tubes due to expansion and contraction

Appearance — Anaconda Deoxidized Copper Water Tubes assembled with Anaconda Solder Fittings present an attractive appearance It is customary practice to clean the tubes after they are installed and apply a coat of clear lacquer or similar substance This treatment keeps the tubes bright and makes an installation of which both the plumber and owner can be proud

Temper and Gauges—Anaconda Copper Tubes are made in both hard and soft temper and in three types as to wall thickness Designated as types K, L and M, they meet the requirements for these types of tubes in U S Government Specification WW-T-799* and A S T M Specification B-88-33* Type K, the heaviest, is recommended for heating lines and general piping Type L tubes are suitable for interior plumbing Type M tubes are not recommended for heating lines

Accuracy of Dimensions—Anaconda Deoxidized Copper Water Tubes are all finished to the close tolerances required by the A S T M and Federal Specifications,

---

*Specifications for Type M tubes, call for hard drawn tubes only.

The American Brass Company

which have been found essential for efficient assembly with solder fittings

Permanent Identification—For permanent identification, the name "Anaconda" and the letter designating the type of tube is stamped in the metal at intervals of approximately 18 in, throughout every coil or straight length of tube

Availability—Anaconda Copper Tubes, in all standard sizes, are carried in stock by distributors of Anaconda Pipe, located in the principal trading areas of the country These tubes, in sizes up to and including 1¼ in are furnished soft in 30, 45 and 60-ft coils, also hard and soft in 20-ft straight lengths Sizes over 1¼ in are furnished, hard or soft, in straight lengths only

ANACONDA SOLDER FITTINGS

Anaconda Solder Fittings are available in both wrought copper and cast bronze They are made to the exacting dimensions so essential for sound, leak-proof joints Smooth inside surfaces permit quick, thorough cleaning which is necessary for satisfactory soldered connections Deep cups, with adequate shoulders for the tubes to butt against, provide a maximum area for the solder bond

All Anaconda Solder Fittings are tested to 90 lb air pressure under water, which is the equivalent of 400 to 450 lb water pressure They are so designed as to offer a minimum of resistance to the flow of water

Anaconda Wrought Copper Solder Fittings—Anaconda Wrought Copper Solder Fittings provide copper to copper connections They are uniformly true to size, of one piece, seamless construction, and are free from porosity—features which make these fittings ideal not only for heating lines but also for air conditioning and refrigerating installations, where the penetrating power of the commonly used refrigerants demands absolute freedom from porosity

The American Brass Company offers Anaconda Wrought Copper Solder Fittings, including elbows, tees, couplings, unions, and a complete range of reduction and adapter combinations from ¼ in to 4 in inclusive

Anaconda Cast Bronze Solder Fittings—Anaconda Cast Bronze Solder Fittings are extensively used for interior plumbing They are available in all standard sizes from ¼ in to 10 in (sizes up to 4 in are carried in stock), in a complete line of elbows, tees, couplings and unions, including all standard reduction and adapter combinations With such a broad range of Cast Bronze Solder Fittings, any usual type of connection can be made without resorting to the trouble and expense of using combinations of fittings

Literature — Anaconda Deoxidized Copper Tubes, Fittings, Solder and Accessories are discussed at length in Anaconda Publication B-1, Tenth Edition Copies will be mailed to Heating, Piping, Ventilating and Air Conditioning Engineers on request

ANACONDA "85" RED-BRASS PIPE

Anaconda "85" Red-Brass Pipe, in standard pipe sizes, is offered as the highest quality corrosion-resistant pipe commercially obtainable at a moderate price and is recommended for steam return lines

Anaconda "85" Red-Brass Pipe contains 85 per cent copper and conforms to government specifications for Grade "A" water pipe The words "Anaconda 85 Red-Brass" are stamped in the metal at intervals of one foot throughout each length

EVERDUR METAL

***Everdur Tanks**—Nearly all copper, Everdur is a special non-rust alloy which combines high strength and complete immunity to rust with ready weldability It is an ideal material for durable, rustless water tanks of every description—from domestic range boilers to giant storage heaters for hotels, laundries, hospitals, textile plants, schools or breweries

Everdur is made in all commercial shapes, including tank plates which have physical properties as given in A S T M tentative specifications B96-31T For additional data, and names of fabricators, address our nearest office or agency

Everdur for Air Conditioning Equipment—Everdur Metal has been used with marked success for fans and blowers, ducts, humidifiers (air washers) and for various cast and wrought parts of other equipment items subject to corrosive influences

Because of its strength and welding properties, Everdur may be substituted for steel in the same gauges, fabricated by substantially the same methods, and with the same equipment

*"Everdur" is a registered trademark identifying products of The American Brass Company made from alloys of copper, silicon and other elements

Chase Brass & Copper Co.

Incorporated

Subsidiary of Kennecott Copper Corporation

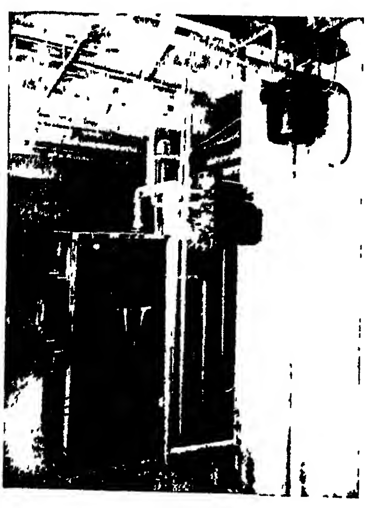
Waterbury **Connecticut**

RUSTLESS HEATING LINES OF CHASE COPPER TUBE AND CHASE SWEAT FITTINGS

Here are a few reasons why copper tube should be used for heating lines **1.** Copper does not rust, therefore the water or steam inside the lines remains clean **2.** Absence of rust and scale formation assures permanent original steam or water carrying capacity. **3.** The resistance to flow in copper tube is low **4.** Delicately adjusted thermostatic valves and traps are not rust-clogged out of commission **5.** Copper is inexpensive and as satisfactory for giving service as any material known

TEMPER—Both hard and soft tubing are available for heating lines In remodelling work, and in replacing old heating systems the soft copper tube will be found particularly useful It can be worked down between walls and around corners, long 60 ft coils eliminate many useless connections For all soft copper tube we recommend the extra heavy gage, (known by United States Government specifications as "Type K"). For the average new installation we recommend the light gage copper tube ("Type M")

PRESSURE—Chase copper tube is adaptable for all low pressure heating systems While we do not recommend the use of copper tube with any steam system having more than 30 lb. pressure, the factor of safety over such pressure is very great The bursting pressure, for example, of ¾ in. Chase copper tube and sweat fittings when connected together is 3,050 lb per square inch



RETURN LINES—While we recommend copper tube for all heating lines, it is in the return lines that the greatest amount of rusting takes place Architects and Engineers have learned by experience that these are the lines that are apt to rust and in some installations need replacement, after a very short period of service

CATALOG—The information on these two pages is necessarily brief If you would like the detailed story write for a copy of our book, "Chase Copper Tube for Heating Lines"

CHASE

This is a typical vapor installation Notice the neatness of the copper tube and sweat fitting heating lines Also the absence of union connections

CHASE COPPER TUBE FOR HEATING LINES

CHASE

Copper Tube and Sweat Fittings are used for mains and risers This minimizes resistance to flow of steam

Steam Carrying Capacities

The steam carrying capacity of copper tube is greater than that of iron pipe of the same nominal size For example, in a steam main, 2 in copper tube is capable of taking care of a load of 444 sq ft of radiation as compared with 386 sq. ft for 2 in iron pipe This is an increase of 15.1 per cent.

For equal steam pressure drop the capacity of copper tube is on the average 10 per cent greater than for the same nominal size of iron pipe Because of this it is frequently possible to use tube of smaller size

Pipe Covering

Copper Water Tube uses the same standard pipe coverings as other heating pipes, but in most cases requires one size smaller covering This is a saving in the cost of covering.

Costs

The slight extra cost of Chase copper tube and sweat fittings is a very small amount in dollars to pay for the advantages of a copper tube installation.

SIZES AND WEIGHTS OF TYPE "M"

Nominal Size	Outside Diameter	Inside Diameter	Wall Thickness	Pound per Lineal Foot
½ Inch	0 625 Inch	0 569 Inch	0 028 Inch	0 203 Lbs
¾ "	0 875 "	0 811 "	0 032 "	0 328 "
1 "	1 125 "	1 055 "	0 035 "	0 464 "
1¼ "	1 375 "	1 291 "	0 042 "	0 681 "
1½ "	1 625 "	1 527 "	0 049 "	0 940 "
2 "	2 125 "	2 009 "	0 058 "	1 46 "
2½ "	2 625 "	2 495 "	0 065 "	2 03 "
3 "	3 125 "	2 981 "	0 072 "	2 68 "
3½ "	3 625 "	3 459 "	0 083 "	3 58 "
4 "	4 125 "	3 935 "	0 095 "	4 66 "

Arthur Harris & Co.

210-218 N Aberdeen (formerly Curtis) Street **Chicago, Ill.**

ENGINEERS — FABRICATORS OF NON-FERROUS METALS AND STAINLESS STEEL

Products

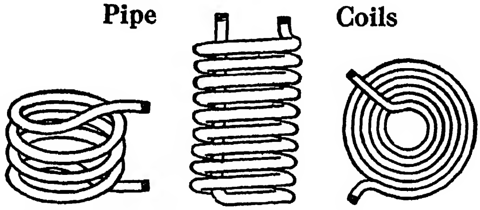
Apparatus for Brewers, Distillers, Paper Mills, Soap Plants, Pharmaceutical Manufacturers, Dyers, and Manufacturers of Grain and Wood Alcohol, Acetic Acid, Lacquer, Vinegar, Tan Liquor Extract, Malted Foods, Milk Products, Sugar, Meat Extracts, Confectionery, Glycerine, Glucose, Preserved Fruits, Cider, Gelatine, Glue, Varnish, Turpentine, etc. Bulletin on request.

Bends

We make bends in every shape from all sizes of pipe and tubing Standard or special connections Copper, brass, aluminum, stainless steel, monel, tin and nickel U-bends for storage water heaters

Also special pipe work for industrial installations, plumbing, heating and brewing Perforated pipe, double pipe coolers, etc

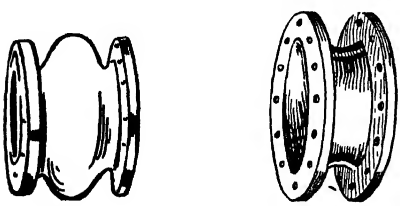
Pipe Coils



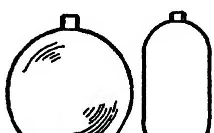
We have special equipment for making coils in all shapes and sizes from pipe or tubing—copper, brass, aluminum, stainless steel, monel, block tin and pure nickel Standard or special fittings Send sketch, blue-print or old coil

Copper Expansion Joints

For low pressure and vacuum Made in two styles—convex and concave Sizes 4 in to 60 in diameter. Cast iron or steel flanges Flanges drilled to American standard unless otherwise ordered



Metal Floats



Made of copper, plain steel, stainless steel, aluminum, brass, monel and pure nickel For open tank and all pressures

Seamless copper ball floats carried in stock in diameters of 4 in , 5 in , 6 in , 7 in , 8 in , 10 in and 12 in and for open tank and pressures of 25, 50, 100 and 150 lbs Special sizes and pressures made to order Float catalog sent on request

Metals Fabricated

Aluminum, Block Tin, Brass, Bronze, Copper, Everdur, Monel, Nickel and Stainless Steel.

Mueller Brass Co.

Port Huron, Mich.

Branch Offices and Representatives in Principal Cities

Albany, N Y
Atlanta, Ga
Birmingham, Ala
Boston, Mass
Buffalo, N Y
Chicago, Ill
Cincinnati, Ohio
Cleveland, Ohio
St Louis Mo
Dallas, Texas
Dayton, Ohio
Denver, Colo
Detroit Mich
Flandreau, S D
Flint, Mich
Harrisburg, Pa
Indianapolis, Ind
Kansas City, Mo
Lansing, Mich
Los Angeles, Calif
Milwaukee, Wis
Minneapolis, Minn
Newark, N J
Philadelphia, Pa
N S Pittsburgh, Pa
Rochester, N Y
San Francisco, Calif
Sarasota, Fla
Seattle, Wash
Washington, D C

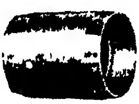
Canadian Sales and Manufacturer

Canada Wire and Cable Co , Ltd , Toronto, Canada

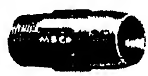
Mexico

George F Gilfrin, Edificio "La Nacional," Mexico, D F

PRODUCTS—STREAMLINE Copper Pipe and Seamless Tubes; STREAMLINE Hard Copper Pipe and Solder Fittings; Valves, Flared and STREAMLINE Solder Fittings for Mechanical Refrigeration; Forgings of Brass, Bronze and Copper; Castings of Brass and Bronze; Rod; Screw Machine Products; Fabricated Parts and Special Nickel and Chromium Plated Parts.



Coupling Copper to Copper



Copper to Outside I P S

45 Deg Elbow

Streamline Copper Pipe and Fittings for heating, plumbing, air conditioning and industrial use are made by the Streamline Pipe and Fittings Division, Mueller Brass Co , Port Huron, Mich

The **Streamline** Solder Fitting is the original solder type fitting, introduced and manufactured by the Mueller Brass Co of Port Huron, Mich It incorporates many advantageous features and has proved to be the revolutionary advance of the age in the development of piping systems for plumbing and heating and for many industrial uses

The **Streamline** Solder Fitting is not connected either by threading or flaring, but by soldering The outside surface of the copper pipe and the inner surface of the **Streamline** fitting are cleaned with sandcloth, and solder flux is then applied to the cleaned surfaces to eliminate oxidation when the assembled joint is heated The joint is then sufficiently heated with a blow or acetylene torch and the soldering operation is performed by feeding wire or stick solder through the feed hole in the fitting

The **Streamline** Solder Fitting alone has the solder feed hole, groove and taper The solder feed hole, through which the solder is introduced, enters directly into an internal feed channel The feed channel is located equidistantly between the internal shoulder against which the pipe rests and the outer edge of the fitting When solder is introduced it is distributed by capillarity from the feed channel and distributed evenly and thoroughly between the bonding surfaces, traveling inward to the shoulder and outward to the edge of the fitting where it appears as a continuous solder ring around the full circumference of the pipe. This ring, and feed hole completely filled with solder, constitute positive proof to the operator that the joint is permanently leak-proof An actual pressure test is not necessary

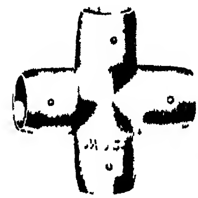
Streamline Pipe and Fittings Division

MUELLER BRASS CO.

Port Huron, Mich.

Patents 1770852, 1776302

Tee



90 Deg Elbow

Tee
Copper to Inside I P. S

Crosses

The tapered ends, since they are the thinner sections of the fitting, hasten the cooling of the solder at these points and facilitate the completion of the joint

The solder may be fed from any position, whether the feed hole is located at the top, side or bottom Owing to the never failing phenomena of capillarity, the solder will flow up, down or laterally with equal facility

Streamline Copper Pipe is a seamless cold drawn copper tubing conforming to *A S T M* B 88-33 For most piping purposes, hard drawn pipe is used though annealed material is supplied where bends are to be made Three weights of **Streamline** Copper Pipe, Govt. Types K, L and M, are made in all sizes, and an additional lighter weight is made in sizes 3 in and larger The latter is used mainly by the paper industry for pressures not over 125 lb

This range of weights permits its use for water or air pressures up to 400 lb

Streamline Solder Fittings are furnished in sizes from 1/4 in to 12 in inclusive with a full range of reducing sizes

Fittings above 6 in are flanged and may be had with either *A S A* or rivetted pipe standard flanges Mating flanges are soldered to the pipe

The **Streamline** Solder Fitting permits the use of thin-walled copper pipe and places a non-rusting, non-clogging piping system within the reach of the ordinary investor Vibration is not localized at the joints, but is harmlessly dissipated throughout the system Copper Pipe has the property of transferring the heated element (steam or hot water) from the point of generation (boiler) to the radiators quickly and with slight temperature drop

During the last five years architects and engineers have used **Streamline** Copper Pipe and Fittings successfully in every type of building construction and in thousands of installations throughout the United States and Canada.

In addition to its rust, clog and vibration-proof qualities and long life, **Streamline** has many other advantages such as the reduction in size of pipe lines and radiator connections from those nominally used, a neat, compact installation requiring a minimum of space and important advantages in industrial and drainage applications There is a **Streamline** product for every piping requirement

Jones & Laughlin Steel Corporation

AMERICAN IRON AND STEEL WORKS

Jones & Laughlin Building, **Pittsburgh, Pa.**

WELDED AND SEAMLESS STEEL TUBULAR PRODUCTS

J & L Welded Pipe

Jones & Laughlin manufactures Standard Weight, Extra Strong, and Double Extra Strong Welded Pipe, Black and Galvanized, for steam, gas, air, water, refrigeration and sprinkler work. Sizes ⅛ in. to 16 in O D inclusive

J & L Copper-bearing Steel Pipe, when specified, can be supplied in standard weight, or extra strong, black or galvanized. Use of this product is recommended for long life, where piping is to be exposed to the atmosphere or other alternate wet and dry conditions

Jones & Laughlin Steel Pipe is made of soft, weldable steel rolled from solid ingots made to a special analysis which, checked over a period of years, has proved to be very uniform in quality The steel pipe produced from this special grade of J & L Steel is soft and ductile, free cutting, strong at the welds, and free from excess scale. J & L Pipe is commercially straight and free from blisters, cracks or other injurious defects and is well within the allowable tolerances as to dimensions and weights, and true to round in the outside diameter

Careful attention is given the threading of the pipe with good clean-cut threads fitted with sound couplings correctly tapped to give a tight joint Soft, ductile steel of free cutting quality enables the contractor to cut clean, sound threads on the job

The Jones & Laughlin process of galvanizing assures a thorough coating and insures against pipe being clogged with spelter The surface of the pipe is carefully cleaned so that when the galvanized coating is applied it adheres strongly and does not tend to flake off

J & L Seamless Pipe

J & L Seamless Pipe is made in three weights; standard, extra strong and double extra strong Sizes· ⅜ in nominal to 14 in O.D inclusive.

J & L Seamless Steel Pipe is pierced from a solid billet—there are no welds The result is dependable and uniform wall strength The method of manufacture, and the use of only specially selected steel, assure exceptional ductility, a quality that is essential to successful coiling and bending, and flanging for Van Stone joints

J & L Seamless Pipe can be used with full satisfaction in either threaded joint or completely welded installations Ductility, strength and safety—highly developed attributes of J & L Seamless—make this product especially adaptable for air, steam, gas and gasoline lines, boilers, refineries, dry kilns, refrigerating systems and other exacting applications

J & L Hot Rolled Seamless Steel Boiler Tubes

J & L Seamless Boiler Tubes are manufactured in accordance with the *A S M E* Boiler Code and comply with the *A S T M* Specifications and the rules and regulations of the Bureau of Navigation and Steamboat Inspection of the U. S. Department of Commerce They are supplied in a full range of standard sizes, from 1 in O D to 6 in O D. inclusive

The process by which Jones & Laughlin manufactures seamless boiler tubes is largely responsible for the unusually high ductility of the product. It is a process in which a forging action is predominant. This forging action gives to the steel the greater density and higher ductility that may be expected of any forging operation It makes it stronger yet more pliable and, therefore, more easily formed in its cold state Forging also effectively eliminates any such imperfections as air holes or blow holes that may be present in the steel

Inspection of this product begins with the careful selection of the steel for the billets and continues without interruption through every stage of manufacture. It is your assurance of receiving only the very finest boiler tubes that can be manufactured

Other J & L Tubular Products

J & L also manufactures Reamed and Drifted Pipe in sizes 1 in to 6 in. inclusive, Dry Kiln Pipe, Pipe for Refrigeration Service, Water Well and Irrigation Casing, Line Pipe and a complete line of Oil Country Tubular Products in welded and seamless

Republic Steel Corporation

General Offices REPUBLIC STEEL **Cleveland, Ohio**

District Offices

ALBANY, N Y
BIRMINGHAM, ALA
BOSTON, MASS
BUFFALO, N Y
CHICAGO, ILL
CINCINNATI, OHIO
CLEVELAND, OHIO
DENVER, COLO
DETROIT, MICH
GRAND RAPIDS, MICH
HOUSTON, TEXAS
INDIANAPOLIS, IND
KANSAS CITY, MO
LOS ANGELES, CALIF
MILWAUKEE, WIS
NEW YORK, N Y
PHILADELPHIA, PA
PITTSBURGH PA
SALT LAKE CITY, UTAH
SAN FRANCISCO, CALIF
SEATTLE, WASH
ST LOUIS, MO
ST PAUL, MINN
TOLEDO, OHIO
TULSA, OKLA
WASHINGTON, D C
YOUNGSTOWN, OHIO

General Export Dept. NEW YORK CITY

TONCAN COPPER MOLYBDENUM IRON

REG. U.S. PAT. OFF.
TONCAN
COPPER
MO-LYB-DEN-UM
IRON

What Is Toncan Iron?

Toncan Iron is a highly refined open-hearth iron with which is alloyed the correct proportion of copper and molybdenum As such, it possesses the maximum rust-resistance of any ferrous material in its price class

Thousands of rigid tests and the performance of untold tons of Toncan Iron in actual service point to the greater economy in the heating and ventilating systems of which the sheet metal and pipe are Toncan Iron

Advantages of Toncan Iron

(1) It resists to a higher degree more of the many and varied types of corrosion than other ferrous material in its price class This resistance is not confined to the surface or "skin" of the metal, but is uniform throughout

(2) It combines the high rust-resistance of an alloy iron with many of the desirable physical qualities of less resistant ferrous materials

(3) It is one of the most workable of materials Sheets form easily Pipe may be handled like any iron

(4) Unlike other ferrous materials, cold working — cutting, bending, punching, threading, etc —does not materially affect the rust-resistance of Toncan Iron

(5) It welds easily by any of the usually accepted modern methods The use of Toncan Iron welding rod insures an installation of equal rust- and corrosion-resistance throughout.

(6) A uniform and tightly adherent galvanized coating can be applied, thus adding the protection of a heavy coating of zinc to the already high rust-resistance of the base metal itself

(7) Through its longer, trouble-free life, it has been found to cost far less per year of service Its use is more than an economy. It is insurance against early sheet and pipe failures and costly replacements

Toncan Iron Sheets

Toncan Iron is available in various sheet forms Plain sheets may be had black or galvanized, in gauge nos 8 to 28, widths from 24 to 50 in, and lengths from 10 to 13 ft, depending upon gauge and width The Toncan Iron trade mark is stenciled in two or three places on every sheet

Toncan Iron Pipe

Toncan Iron Pipe is available in standard and extra heavy weights, black or galvanized, in sizes from ¼-inch to 16-inch O D All Toncan Iron Pipe, 2-inch and larger, is electric resistance welded, and combines the foregoing advantages of Toncan Iron with the advantages of Republic's electric welding process—100 per cent weld, perfect roundness, uniform diameter and wall thickness, and freedom from scale Toncan Iron Pipe, "black" finish, is painted blue, galvanized finish is marked with two blue stripes Couplings are stamped TM

Source of Supply

Toncan Iron Sheets and Pipe are stocked by jobbers in all large cities, leading contractors everywhere use Toncan Iron and are glad to supply it where specified. If, for any reason, you cannot obtain Toncan Iron, write to Republic's nearest sales office

Literature

Two interesting books, "The Path to Permanence" on Toncan Iron Sheets and "Pipe for Permanence" on Toncan Iron Pipe, will bring you the complete story. Write for either or both books.

Other Republic Products

Republic Steel Corporation manufactures hundreds of iron, steel and alloy products, among which of interest to heating and ventilating engineers are Enduro Stainless Steel in sheets and other usual forms, steel pipe, steel sheets, and steel or Toncan Iron boiler tubes

Revere Copper and Brass Incorporated

Executive Office 230 Park Avenue, **New York City**

MILLS—BALTIMORE, MD, TAUNTON, MASS, NEW BEDFORD, MASS
ROME, N Y, DETROIT, MICH, CHICAGO, ILL

SALES OFFICES—BOSTON, MASS, PROVIDENCE, R I, PHILADELPHIA, PA, ATLANTA, GA, NEW ORLEANS, LA, NEW YORK, N Y, PITTSBURGH, PA, CLEVELAND, OHIO, CINCINNATI, OHIO, GRAND RAPIDS, MICH, MILWAUKEE, WISC, ST LOUIS, MO, MINNEAPOLIS, MINN, DALLAS, TEXAS, SEATTLE, WASH, SAN FRANCISCO, CALIF, LOS ANGELES, CALIF.

REVERE COPPER WATER TUBE

Revere Copper Water Tube is recommended for heating lines, refrigerant lines, heat control lines and for other heating and ventilating piping This tube is seamless, 99 9 per cent pure copper, completely deoxidized, with a gun-barrel finish inside.

It is furnished in three types known as "K," "L," and "M," which meet Government and A S T M. specifications. In general the Type "K" is used where corrosive conditions are severe, and types "L" and "M" where these conditions are normal For most heating work, Types "L" and "M" are satisfactory.

Types "K" and "L" are furnished in both hard and soft tempers, Type "M" in hard temper only The hard temper is used for new and exposed work; the soft temper for hidden replacement work and where flexibility is essential

(A 40 page hand book on Revere Copper Water Tube will be sent on request)

Fittings—This tube is joined with Streamline soldered fittings or with any standard make of compression fittings Thus, threading is eliminated and metal in S P.S. pipe used only for cutting threads is saved. The wall thickness of Revere Copper Water Tube is uniform throughout and the joints are leak-proof, vibration-proof and stronger than the tube itself

The Streamline fitting is so made that friction is reduced to a minimum

Advantages of Revere Copper Water Tube—In brief these are as follows·

1. Revere Copper Water Tube has many installation advantages With Streamline Fittings it can be installed in a minimum of space The soft temper tube can be bent where needed, as for example, around spandrel beams This saves installation time and eliminates fittings Furring is greatly reduced Joints may be easily disassembled

2. Revere Copper Water Tube cannot rust, so that it is not necessary to install oversize pipe It is recommended particularly for steam return lines where the greatest amount of corrosion takes place

3. In forced circulation hot water heating systems, where small pipe sizes are used, Copper Water Tube can be installed at prices that are competitive with iron or steel The range of sizes is such that a balanced system can be designed using minimum pipe sizes

4. In general pipe covering can be one size smaller with Copper Water Tube than for the same size I P S pipe because of the smaller outside diameter of the tube It can also be light in weight just heavy enough to prevent convection losses

5. Because of its flexibility and the ingenuity with which Copper Water Tube can be joined with Streamline Soldered Fittings, this tube is recommended for all sorts of special hook-ups

To Heating and Air Conditioning Manufacturers—Revere Engineers are anxious to cooperate with manufacturers of heating and air conditioning equipment and to make recommendations with reference to the use of non-ferrous products

Revere Copper Water Tube
STANDARD DIMENSIONS AND WEIGHTS

Size in In	TYPE K			TYPE L		TYPE M	
	O D in In	Wall Thickness In	Wt. Lbs per Ft	Wall Thickness In.	Wt Lbs per Ft	Wall Thickness In	Wt Lbs per Ft
3/8	500	049	.269	035	198	025	.144
1/2	625	049	.344	040	285	028	203
5/8	.750	049	418	042	362		
3/4	.875	065	641	045	455	032	328
1	1 125	065	839	050	655	035	464
1¼	1 375	065	1 040	055	884	042	681
1½	1 625	072	1 360	060	1 140	049	940
2	2 125	.083	2 060	070	1 750	058	1 460
2½	2 625	095	2 920	.080	2 480	065	2 030
3	3 125	.109	4 000	090	3 330	.072	2 680
3½	3 625	120	5 120	.100	4 290	083	3 580
4	4 125	134	6 510	110	5 380	095	4 660
5	5 125	.160	9 670	125	7 610	109	6 660
6	6 125	192	13 870	140	10 200	122	8.910
8	8 125	271	25 900	200	19 290	170	16 460

Recommended Operating Pressures

Type K—Hard Temper	up to 400 pounds
Type K—Soft Temper	up to 250 pounds
Type L—Hard Temper	up to 250 pounds
Type L—Soft Temper	up to 150 pounds
Type M—Hard Temper	up to 250 pounds

Tempers and Lengths

Type K—Hard and Soft Temper in straight 20 ft lengths
Type L—Soft Temper in 30 ft, 45 ft, and 60 ft coils
Type M—Hard Temper in straight 20 ft lengths

Wolverine Tube Company

SEAMLESS COPPER, BRASS AND ALUMINUM

Main Office and Mill· 1411 Central Avenue, **Detroit, Mich.**

New York Office 420 Lexington Avenue

Sales Offices

Atlanta, Ga	411 Georgia Savings Bank Bldg	Newark, N J	965 Broad St
Baltimore, Md	308 Hearst Tower Bldg	Philadelphia, Pa	231 North 12th St
Boston (Charlestown), Mass	410 Rutherford Ave	Pittsburgh Pa	1228 Brighton Rd , N S
Buffalo, N Y	Court and Wilkeson	Portland Ore	524 N W 14th Ave
Chicago, Ill	129 S Jefferson St	Richmond, Va	Mutual Bldg
Cleveland, Ohio	1740 East 12th St	St Louis, Mo	4067 Park Ave
Dallas, Texas	1916 Young St	San Francisco, Calif	7 Front St
Dayton, Ohio	Route No 9	Seattle Wash	1005 E Pike St
Denver, Colo	1210 California St	Washington, D C	1108 16th St
Long Island City, N Y	23-10 44th Road	Toronto, Ont	20 Beverley St
Los Angeles, Calif	1015 East 16th St	Winnipeg, Man	80 Lombard St
Louisville, Ky	429 W Main St		
Milwaukee, Wis	647 W Virginia St	EXPORT H M Robins Co , 120 Madison Ave ,	
Minneapolis, Minn	529 S Seventh St		Detroit U S A

DEOXIDIZED COPPER WATER TUBE

Copper with a purity of not less than 99 9 per cent is used in the manufacture of Wolverine Copper Water Tubing The use of pure copper and the extreme care followed in the process of manufacture accounts for the uniformity in quality, the resistance to corrosion, and the widespread use of Wolverine Copper Water Tubing

The mirror-like inside surface refuses to collect scale or sediment The ductility of Wolverine Copper Water Tubing permits repeated freezing before it will burst The ease of installation when using Wolverine Copper Water Tubing with compression or sweat fittings is due to the close tolerances to which the tube is held

Resistance to rust and corrosion permits the use of smaller lines when Wolverine Copper Water Tubing is specified instead of metals that rust

Wolverine Deoxidized Copper Water Tubing is made to U S Government WW-T-799 and **A.S.T.M.** B 88-33 Specifications under the regular types K, L, and M

Government Type K, A.S.T.M. Class K

Recommended for Air Conditioning, Refrigeration, Oil Burner, and Plumbing and Heating installations Also for Gas, Steam, Oil Lines, and industrial uses and where water conditions are severe

Wolverine Type K for Air Conditioning and Refrigeration fits any standard sweat fitting on the market

Sizes up to 2 in approved by Underwriters' Laboratory

Furnished in hard or soft temper in straight 20 ft lengths, soft temper in 30, 45 and 60 ft coils

Government Type L, A.S.T.M. Class L

For Oil Burner, Air Conditioning, Refrigeration, and general plumbing uses.

Suitable for normal water conditions

Furnished in hard or soft temper in straight 20 ft lengths, soft temper in 30, 45 and 60 ft coils

Government Type M, A.S.T.M. Class M

Suitable for Air Conditioning and Refrigeration installations and for interior plumbing and heating purposes

Furnished in hard temper in straight 20 ft lengths only

For use with soldered fittings only

FABRICATED TUBING

Wolverine offers a complete and competent fabrication service for tube parts This includes bending, flaring, swaging, brazing, etc. Every operation is held rigidly to size and specification

DEHYDRATED REFRIGERATION TUBING

Wolverine Dehydrated Soft Copper Refrigeration Tubing is guaranteed to meet **A.S.T.M.** Specification B 68-33 The ends of each coil of Dehydrated Refrigeration Tubing are solder-sealed and crimped in the form of Wolverine "W"

Wolverine Refrigeration Tubing, in securely wrapped coils, is now available This spiral crepe paper wrapping protects the coil against atmospheric contact, dirt and other foreign matter It keeps the tubing absolutely clean until ready for use and it gives the service man a compact package to save room in his kit

Your local refrigeration jobber can supply you with any quantity or size of Wolverine Tubing Large stocks carried in the mill at all times for prompt shipment

International Heating & Ventilating Exposition

THE AIR CONDITIONING EXPOSITION

Permanent Address—Grand Central Palace, **New York, N. Y.**

EXPOSITIONS HELD

The first in Philadelphia, 1930
The second in Cleveland, 1932
The third in New York, 1934
The fourth Exposition in Chicago, 1936
The fifth in New York, January 24-28, 1938

Subsequent Expositions will be held on alternate, even numbered years

These are held coincident with the Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS and are directed by the International Exposition Company, under the auspices of the A S H V E

EXHIBITORS

Comprise leading firms in each phase of the industry number has varied from 150 to 304 exhibitors

EXHIBITS

These range from and comprise all the types of articles discussed or advertised in this copy of THE A S H V E GUIDE

1 *The* COMBUSTION *Group* Furnaces, burners (oil and gas), grates, stokers, boilers, radiators (various types), refractories and auxiliaries
2 *The* HYDRAULIC *Group* Water feeders, water heaters, pumps, traps, valves, piping, fittings, expansion joints, pipe hangers, etc
3 *The* STEAM HEATING *Group* Vapor heating and steam specialties
4 HOT WATER HEATING *Group*
5 AIR *Group* Warm Air furnaces and stoves, registers and grilles, cooling towers, air filters, motors, fans, blowers, conditioning equipment, ventilators (room and industrial types), unit heaters, etc
6 CONTROL *Group*· Instruments of precision for indicating, controlling or recording temperature, pressure, volume, time, flow, draft or any other function to be measured
7 REFRIGERATING Compressors, condensers, cooling apparatus, contingent apparatus and refrigerants for homes, factories, railroads, etc
8 CENTRAL HEATING Apparatus and materials especially designed or adapted to the use of central heating and central heating station supplies
9 INSULATING Structural insulators (refractory and cellulose materials), asbestos, magnesia clays and combinations thereof, pipe and conduit covering, etc, weather-stripping, etc
10 *The* MISCELLANEOUS *Group* Electric Heaters, boiler and pipe repair alloys, liquids and compounds, etc, which are not included above
11 MACHINERY AND GENERAL EQUIPMENT
12 AIR CONDITIONING *Group* Equipment which circulates and filters the air, and in summer dehumidifies and cools, in winter heats and humidifies, and does all these in proper season for complete, all year round air conditioning

VISITOR ATTENDANCE

Comprises a registered attendance invited to the exposition and includes

(Figures are 1934 analysis)

INDUSTRIES

Governmental	186
Distributional Channels Contractors, Dealers, Jobbers, Supply Houses, 32 classifications	5998
Home Owners	414
Industrial Users, 49 classifications	3428
Professional and Service Organizations, 23 classifications	2017
Public Utilities	922
Real Estate Management and Operation, 10 classifications	945
Educational Institutions	303
Miscellaneous	1043
TOTAL	15 256

OCCUPATIONS

Executive (44 titles)	7850
Construction (16 titles and trades)	1022
Operation (44 titles and trades)	2018
Technical (64 titles)	2144
Not Classified including Educators, Publishers, Home Owners, etc	2222
TOTAL	15,256

The registered attendance at the 1936 Exposition in Chicago was 27,743 but the analysis of industries and occupations has not been completed at the time this goes to press These visitors came from 273 cities and towns in 44 States of the United States and 35 cities and towns in 10 foreign countries

Industrial Expositions in America lead the expositions of the world in style, business effectiveness, industrial influence and educational value This Exposition stands among the leaders in Industrial Expositions in America It is an educational institution which biennially brings together the research developments and improvements in equipment and materials

American Society of Refrigerating Engineers

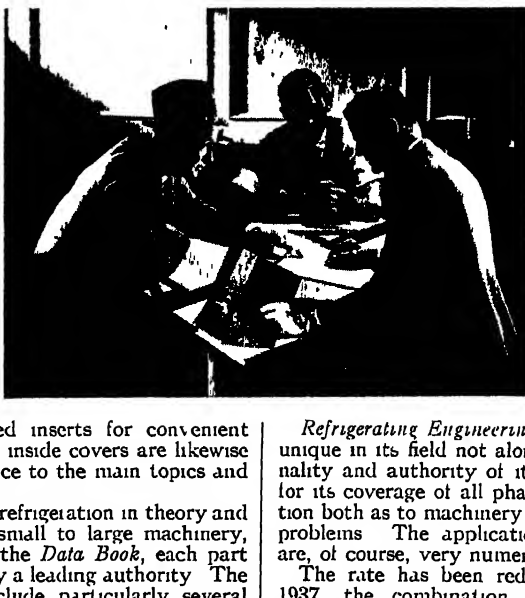
37 West 39th Street, New York, N. Y.

REFRIGERATING DATA BOOK

THE '37-'38 *Refrigerating Data Book*, a one-volume encyclopedia contributed to by 73 refrigerating experts, achieves an excellence matched by few engineering or scientific books in any field Like its predecessors it contains all the fundamental data available on the art of refrigeration, with many additions in the light of recent researches Unlike some earlier editions it is, however, arranged to permit of use by the novice or student in refrigeration, and as a general referance work

This book contains 520 pages of text plus a catalog section with various directories and lists The technical contents are divided into four sections separated by section indices on colored inserts for convenient reference The inside covers are likewise used for reference to the main topics and the 33 chapters

All phases of refrigeration in theory and practice, from small to large machinery, are covered in the *Data Book*, each part being written by a leading authority The applications include particularly several excellent chapters on all phases of air conditioning in design and layout, for large and small installations In hard leather binding, $4 00, express charges collect



REFRIGERATING ENGINEERING

Refrigerating Engineering in its 33rd volume continues to be the periodical source of authoritative information on all phases of the arts and sciences of refrigeration In its most solid aspects it carries the *Journal* of the *A S R E*, a living record of the technical advance of practice and research in its field The larger portion of its pages is, however, devoted to material of wider appeal, with original articles, none the less authoritative, but written in a journalistic style to enable the reader to get the background of each subject and an understanding of it that will stick *Refrigerating Engineering* also prints news, features, write-ups of interesting personalities

Refrigerating Engineering has long been unique in its field not alone for the originality and authority of its contents, but for its coverage of all phases of refrigeration both as to machinery and application problems The applications of this art are, of course, very numerous

The rate has been reduced to $4 for 1937, the combination rate with the *Refrigerating Data Book* being $7 50

MEMBERSHIP ACTIVITIES

IT is the policy of the *A S R E* to treat in its meetings current subjects touching upon all phases of the art of refrigeration Membership is in four grades with dues from $7 50 to $17 50 Sections hold meetings in the following cities Boston, New York, Philadelphia, Detroit, Chicago, Milwaukee, St Louis and Los Angeles The Society holds its 33rd Annual Meeting in 1937

CODES AND STANDARDS

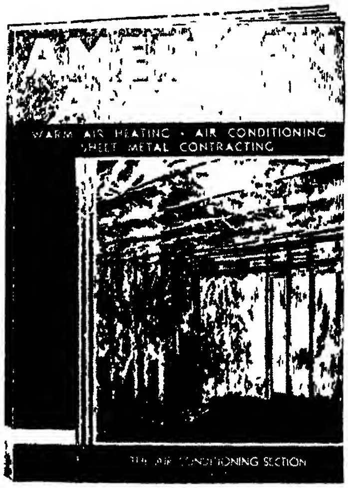
THE *A S R E* has a number of technical codes in its series of *Circulars*. Recent additions include the code for *testing and rating mechanical condensing units* (No 13, 15c) and the code for *testing and rating air conditioning equipment* (No 14, 20c) Other current data Plant test code, corrosion prevention code, safety code, code for testing iced refrigerators Write for free booklet *Pointers to Authors*, including *style sheet* and *directions for locating information* in refrigeration.

American Artisan

Published by

KEENEY PUBLISHING COMPANY

6 North Michigan Avenue, **Chicago, Ill.**



AMERICAN ARTISAN, now in its 58th year of publication, covers the field of warm air heating, residential air conditioning, and sheet metal contracting A special section of each issue has been devoted to air conditioning since 1932, when it first became apparent that air conditioning for homes was to be along the lines of the central, forced warm air heating system

Its readers are warm air heating and sheet metal contractors, dealers, jobbers and manufacturers, and also architects, engineers, and public utility companies who take it for its thorough coverage of air conditioning for the home field

To answer the industry's need for a dependable guide to equipment purchases, it publishes in each January issue a complete and up-to-the-minute directory of warm air heating, air conditioning and sheet metal products and equipment This directory lists all products used in the field, their trade names, and the full names and addresses of all manufacturers The January 1937 Directory Number, containing the Fifth Annual Directory, will be used by readers as a buying reference throughout the year.

Almost from the day interest in residential air conditioning began to develop, the advantages of the warm air type of heating system, with its duct distribution of air, were plain to see It was adapted to all air conditioning factors, either through a self-contained central unit or through a central furnace to which could be added step-by-step or as a whole, fan, washer, humidifier, filters, controls, cooling, and automatic firing

Today, as a result of this ready adaptability as well as economy, tens of thousands of homes have winter air conditioning—supplied through forced warm air heating with air cleaning and humidification Cooling apparatus can be attached to these systems readily whenever complete, year-'round air conditioning is desired

This trend in residential air conditioning has placed a premium on air handling knowledge, and has brought to the fore the one man experienced in "treating" air at a central place and getting it properly distributed—the warm air heating and sheet metal contractor The warm air heating industry has, furthermore, undertaken and made notable progress toward the solution of the many new engineering problems involved All this has helped to put warm air heating in the center of residential air conditioning

In aiding to develop this trend and assist in the solution of new problems, AMERICAN ARTISAN has provided a service to its field which has made it the recognized authority on residential air conditioning practice

To manufacturers whose products are used in residential air conditioning, AMERICAN ARTISAN offers full coverage of the leading buying factors Such manufacturers will be interested in obtaining a copy of "AMERICAN ARTISAN—Warm Air Heating, Residential Air Conditioning, Sheet Metal Contracting" This study will be sent upon request

AMERICAN ARTISAN is published monthly It is a member of the A B C and A B P

Subscription rates—$2 00 per year, $3 00 for two years in U S, Canada, Mexico, Central and South America Foreign $4 00 per year

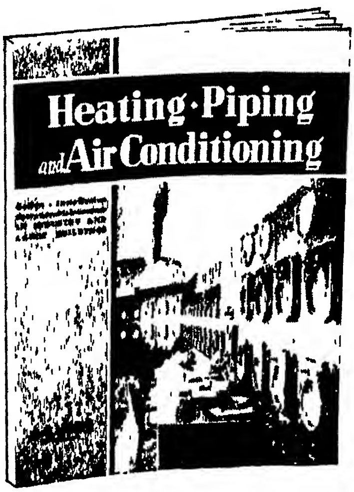
Advertising rates furnished upon request

Heating, Piping and Air Conditioning

Published by

KEENEY PUBLISHING COMPANY

6 North Michigan Avenue, **Chicago, Ill.**



HEATING, PIPING AND AIR CONDITIONING is the publication which carries in each issue the official JOURNAL OF THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in addition to its own regular editorial section.

Its field is that of industry and large buildings. Editorially, it gives specialized attention to the design, installation, operation, and maintenance of heating, piping, and air conditioning systems in such plants and buildings.

In addition, there is published in each January issue a complete Directory of Commercial and Industrial Heating, Piping and Air Conditioning Equipment, which lists all products used in the field, their trade names, and the full names and addresses of all manufacturers. This directory has been established as the industry's buying and specifying guide, and is consulted by readers throughout the year, whenever equipment purchases are up for consideration.

H. P. & A. C. is read by consulting engineers and architects . . . contractors . . . and engineers in charge of heating, piping, and air conditioning in industrial plants, large commercial and public buildings, federal, state, and city governments, school boards and public utilities. Among its subscribers are numbered all members of the A. S. H. V. E., who represent about 30 per cent of its total circulation.

Such a coverage means, for the advertiser, consideration at all points in the selling of a heating, piping, or air conditioning product . . . consideration in the selection of a product during the preparation of plans and specifications, consideration in the actual purchase of a product for installation, consideration in the year 'round buying of a product for operating and maintenance requirements.

It has been evident for some time that the air conditioning field is made up of two distinct markets: (1) Industrial and Commercial, (2) Residential.

These two markets are different in equipment used, different in engineering problems involved, different in engineering, distributing, and consuming personnel . . . require, therefore, different selling jobs.

To sell the industrial and large building field for air conditioning, the manufacturer must win acceptance from the engineers who design, specify, install, operate, and select the system to meet the particular requirements of the plant or building. The system may be central, unit, or "split," but it is these engineers who are the influencing or purchasing factors.

It is to such groups that HEATING, PIPING AND AIR CONDITIONING editorially caters—exclusively in the industrial and large building field. Without waste, the manufacturer of air conditioning products and accessory equipment, such as motors, drives, controls, etc., can reach through its pages those from whom he is seeking the necessary engineering acceptance.

These facts are clearly outlined in a folder which will be sent to interested manufacturers—"HEATING, PIPING AND AIR CONDITIONING—In Industry and Large Buildings."

HEATING, PIPING AND AIR CONDITIONING is a member of the A. B. C. and A. B. P.

Subscription rates—$2.00 per year, $3.00 for two years in U. S., Canada, Mexico, Central and South America. Foreign, $4.00 per year.

Advertising rates furnished upon request.

DOMESTIC ENGINEERING

1900 Prairie Avenue

Chicago

IN a normal period plumbing and heating represents a market which does an annual business far in excess of a billion dollars The first of the building industries to experience a noticeable upturn in business, plumbing and heating had a two-year headway toward the return to normal Virtually every plumbing and heating contractor of consequence in the country has felt this business upswing in a substantial measure. With the existing housing shortage it is inevitable that their business will soon reach even higher peaks than those of previous normal years

As the leading publication in this vast market, reaching more than 19,000 plumbing and heating men, *Domestic Engineering* has lent its untiring efforts, its invaluable resources to this vital cause, playing no minor roll in bringing improved conditions to its industry When new construction was still at a low ebb, *Domestic Engineering* embarked on a planned program to stimulate business Two far-reaching, widely followed F H A prize contests for plumbing and heating contractors fostered by *Domestic Engineering* had the enthusiastic support of the industry in general

As this program continues manufacturers of plumbing and heating products, particularly those whose advertising messages have been appearing in *Domestic Engineering*, are directly benefiting

However, this is only one example of the editorial leadership which has gained for *Domestic Engineering* recognition and prestige in the industry It is only one phase of the service which *Domestic Engineering* renders its industry. Through the Marketing and Research Bureau, a division of the *Domestic Engineering* organization, the manufacturer is enabled to properly analyze his market and to judiciously plan his sales program. As an advertiser, a variety of effective mailing services are at his disposal Through the facilities of *Domestic Engineering* he is enabled to develop an efficient sales staff and to obtain distribution through wholesalers In short, *Domestic Engineering* offers him the means to market his product effectively, with a minimum expenditure

For nearly a half century, *Domestic Engineering* has been a leader in every major movement in its field Throughout these years, it has been looked up to as an authority in its industry

To reach and to sell this billion dollar market, every sound advertising program in the plumbing and heating field should unquestionably include *Domestic Engineering* Complete data concerning this publication and its market are thoroughly covered in "Selling the Plumbing and Heating Market " Write for a copy

Automatic Heat *and* Air Conditioning

1900 Prairie Avenue

Chicago

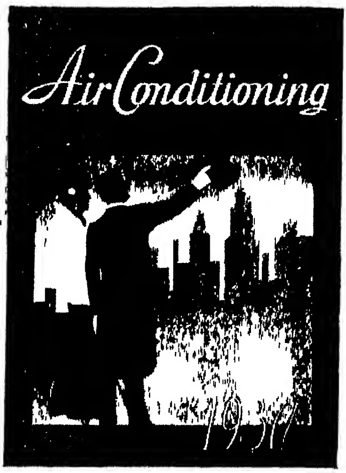
ATTRACTED by the great possibilities which automatic heat and air conditioning opens up to them encouraged by the tremendous public interest it has been accorded many of the better, more aggressive specialty merchandisers from more than thirty other closely allied fields have entered this ever growing industry

Previous to the establishment of *Automatic Heat* and *Air Conditioning*, no single medium existed through which all of the men who comprise this group could be reached In order to reach all the factors concerned with the specification, sale, installation or servicing of automatic heat and air conditioning equipment, it was necessary to use each of the various publications covering the many segments of the industry as a whole The expenditure thus involved was prohibitive and placed a heavy burden upon the advertiser

Now, at a cost that is extremely low in comparison, the advertiser may thoroughly blanket the entire automatic heat and air conditioning industry through just one publication *Automatic Heat* and *Air Conditioning*

In *Automatic Heat* and *Air Conditioning* the manufacturer may reach the consulting engineers and architects who specify equipment for use in this field the dealers, distributors and wholesalers who sell it the manufacturers who make it . all of the men who constitute the $200,000,000 automatic heat and air conditioning market and who have been increasing this market by 40 per cent each year even through lean years

Because it provides them, every month, with a wealth of engineering and merchandising knowledge . knowledge which they may secure from no other single source, and without which they could not



hope to sell with any degree of success in this highly technical field, these men find *Automatic Heat* and *Air Conditioning* an indispensable part of their daily business lives

In company with editorial material of such high calibre and so utterly vital to the men in the field, the advertising message of the manufacturer of automatic heat and air conditioning equipment is placed before its most responsive audience

A complete marketing service is available to sales and advertising executives who must sell the vast automatic heat and air conditioning market Write, on your business letterhead, for the latest copy of the monthly bulletin "Current Conditions in the Automatic Heat and Air Conditioning Industry "

OILHEATING & AIR CONDITIONING *fueloil* JOURNAL

Published Monthly at

420 Madison Avenue

New York

MARKET The oilheating market is an integrated 4-way market—oilburners, heating, air-conditioning, fueloil The complete oilheating dealer sells all four—a good oilburner, using good fueloil, firing a good heating or air-conditioning system

From 1919 to 1930, the only oilheating product sold by burner dealers was the conversion oilburner In 1930 the sale of conversion burners represented 77 3 per cent of the dealer's gross income

By 1936, the average oilheating dealer got only 44 1 per cent of his gross income from conversion burners But, beginning in 1932, he has added three other major oilheating lines heating (starting with boiler-burner units in 1932, now including all types of heating), fueloil, and winter air-conditioning 1936 gross dollar volume of the average dealer was divided Conversion burners, 44 1 per cent, boiler-burner units, 7 3 per cent, heating, 25 7 per cent, fueloil, 16 8 per cent, winter air-conditioning, 3 5 per cent, other, 2 6 per cent

In 1936, 23 3 per cent, or 45,691, conversion oilburners were sold with new cast-iron or steel boilers In addition, dealers sold 14,761 boiler-burner units Oilheating dealers did a total dollar volume in 1936 of $74,385,540 in heating, and $18,102,390 in winter air-conditioning

SERVICES FOR ADVERTISERS

The 1936 Key Market Study
Oilheating & Air-conditioning Merchandising News
Specific Product Reports
Unit sales and brand preference summaries for many heating products and installation materials

OILHEATING & AIR-CONDITIONING FUELOIL JOURNAL gives coverage of this integrated 4-way oilheating market It is the oldest paper in the field (established 1922) Editorially, it has consistently fostered every progressive development in the field and it has encouraged the trend to the complete oilheating dealer

Every issue is carefully balanced editorially to cover the dealer's need for workable information on all four sides of his business

Heating equipment manufacturers have long known FUELOIL JOURNAL as a powerful sales aid Its reader interest is unique among trade papers

CIRCULATION Like its editorial content, the circulation of FUELOIL JOURNAL is carefully controlled to give complete coverage of this 4-way market A detailed break-down from the latest circulation statement (June 30, 1936) follows

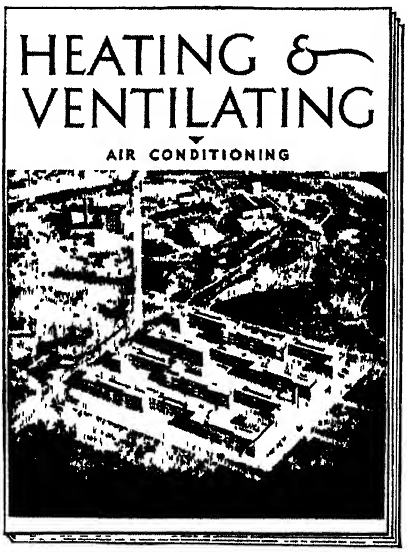
Power oilburner dealers and distributors	9,949	
Key heating contractors, plumbing and heating contractors and engineers	2,053	
Fueloil distributors, selling fueloil and range oil, and branches	3,053	
Accessory and heating supply distributors	682	
Total dealers and distributors		15,737
Power oilburner manufacturers and their executives	541	
Accessory manufacturers	357	
Total manufacturers		898
Total dealers and manufacturers, 95 71 per cent of toal circulation		
Others, and miscellaneous		746
Grand total		17,381

FUELOIL JOURNAL circulation covers the automatic heating field, and its rate per thousand readers is low It will pay you to get full details Write, wire, or telephone

HEATING & VENTILATING

AIR CONDITIONING

THE INDUSTRIAL PRESS . . . *Publisher*
140-148 Lafayette St New York, N. Y.



HEATING & VENTILATING reaches the "key men" of the industry—the engineers, contractors and manufacturers who have the final word in the specification, installation, production and maintenance of the mechanical equipment utilized in the heating, ventilating and air conditioning fields

An editorial program of outstanding alertness and authority is directed by qualified heating and ventilating engineers Special sections and timely feature articles are included from time to time in line with the forward-looking policy that has characterized the publication since its inception in 1904 News, trends, developments, personalities—every side of this important industry is faithfully and authoritatively reported in this outstanding publication

There is a regular section devoted to new equipment, profusely illustrated and comprehensively reported Degree-days and unit fuel consumption for various large cities in the country has been a regular monthly feature of the publication for over eight years The weather in large cities in typical localities of the country is accurately charted Two pages of reference data appear in every issue Reports of meetings, the activities of manufacturers, abstracts of current papers, books and pamphlets, and editorials on the planning, installation and operation of heating, ventilating and air conditioning systems in public buildings, offices, factories, schools, hospitals and homes are other contents of continuous interest

Air conditioning, now coming into its own, has had a champion in HEATING & VENTILATING since 1904 when, in its very first issue, an article on this then infant industry appeared Since that time, for more than thirty years, HEATING & VENTILATING has consistently published the news and developments of air conditioning up to its present high state of perfection and its pages have carried an impressive total of editorial lineage on this subject

Subscriptions to HEATING & VENTILATING are $2 00 a year Advertising rate cards, sample copies and market data will be gladly submitted on receipt of application

Heating Journals, Inc.

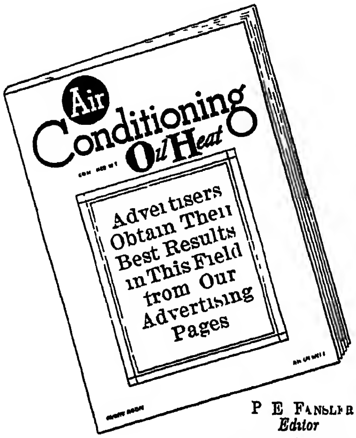
232 Madison Ave Lex. 2-4566 **New York, N. Y.**

Chicago—228 No LaSalle St , Randolph 6086

San Francisco—Don Harway & Co , 155 Montgomery St , Exbrook 6029

Los Angeles—Don Harway & Co , 318 W Ninth St , Tucker 9706

Baltimore—Candler Bldg , Plaza 7065



P E Fansler
Editor

A G Winkler
Advertising Manager

THE air conditioning equipment for the home is being sold installed and serviced in increasing volume by the group formerly known as "oil burner men"—both manufacturers and dealers

The two basic reasons for this are (*a*) the merchandising problem involved, (*b*) technical problems of installation and service

Air conditioning equipment selling requires customer education and specialty merchandising tactics at which the oil burner dealers have been unusually successful for years It also requires figuring heating loads and knowledge of firing equipment, automatic controls, etc. and the oil burner man has these, else he could not survive

The air conditioning-oil burner men (manufacturers and dealers) are taking on other forms of automatic firing equipment 75 per cent of the manufacturers are either producing or developing air conditioning equipment

This paper with its large circulation among oil burner air conditioning men enjoys reader interest that is extremely keen, its editorial content is outstanding in the excellence of its air conditioning articles, advertising rates are 10 per cent lower than any paper of comparable circulation, its record of producing results for advertisers is excellent In 1936 we carried 518 pages of advertising and 175 advertisers used our pages

Available on request: Reprints of articles "How About Air Conditioning?"; "Do you Know the Air Conditioning Market in Your Town?", etc. For advertisers: Regularly issued free lists of dealers who want air conditioning equipment; other valuable lists and services.

CIRCULATION

Oil Burner and Other Air Conditioning Manufacturers	**402**
Air Conditioning and Oil Burner Accessory Manufacturers	**513**
Oil Burner and Air Conditioning Manufacturing Executives	**180**
***Prospective Dealers (Heating Contractors, Electrical and Refrigerator Dealers, Oil and Coal Dealers, etc) (Rotational)**	**6,000**
Miscellaneous (Government Bureaus, Schools, Branch Offices, etc)	**87**
Oil Burner and Air Conditioning Dealers	**12,199**

100% handle oil burners
48.8% or 5,953 handle winter air cond eqpt
36 2% or 4,416 handle summer air cond, eqpt.
33.1% or 4,038 handle complete air cond. eqpt
22.2% or 2,708 handle coal or oil

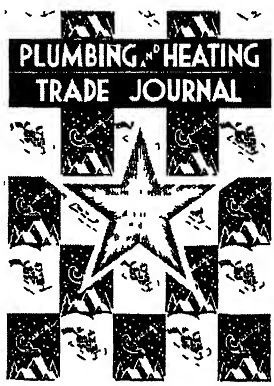
61.0% or 7,441 handle furnaces
69.2% or 8,442 handle boilers
21.9% or 2,672 handle stokers
26.6% or 3,245 handle gas burners
38 0% or 4,636 not now selling air conditioning but are "interested"

Plumbing and Heating Trade Journal

Published by

PLUMBERS TRADE JOURNAL PUB. CO.

515 Madison Ave , **New York City**



PLUMBING and Heating Trade Journal is edited to furnish a well-rounded, efficient service to the men engaged in the plumbing, heating, ventilating and air conditioning fields To this end, it covers both the technical and business phases of their work, as well as many minor but exceedingly important ones

It gives free technical service through a staff of practical engineers, expert merchandising assistance, and its technical and business articles are by men of recognized competence

Thousands of readers come to THE JOURNAL each year for solutions to their technical problems and while some of the questions and answers are published in the Readers' Technical Service section in each issue of the magazine, the vast majority of them—having to do with practically every phase of heating, ventilating and air conditioning as well as plumbing—are answered by mail, because most of the requests for help are urgent and a delay in answering would, in some cases, entail actual monetary loss to the contractor

The Readers' Technical Service Department of THE JOURNAL is staffed by editors who have spent their lives in the business, men who were successful plumbing, heating, ventilating and air conditioning engineers before they wrote a line for publication, and who now devote their entire time to keeping abreast of the field's technical developments and using their knowledge and experience for the benefit of JOURNAL subscribers

The technical service rendered its readers by THE JOURNAL is closely paralled by what it strives to do for them in a business way, for it also publishes many authoritative articles on and answers questions concerning the various ramifications of business management and prints as part of every issue, a special section devoted to selling

One associate editor spends his entire time in the field writing articles on the business management problems of JOURNAL readers, and the practical solution of those problems

Supplementing the business and technical articles is a large amount of exclusive, staff-gathered news that high-lights the background of the trade's activities

This news background is vital It completes the industrial picture for the reader. It keeps him in intimate touch with what the various important associations and his fellow members of the craft are doing throughout the nation and it charts the trends that are likely to have a very definite influence on the future operation of his business

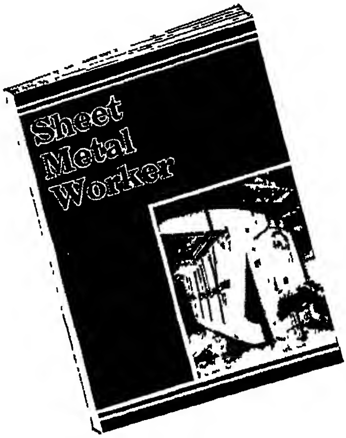
THE JOURNAL editorial department draws its news from over a hundred trained correspondents located at strategic points throughout the country—by far the largest group of exclusively editorial workers used by any paper in the industry.

It is this combination of the technical, business and news aspects of the industry that enables THE JOURNAL to achieve a finely balanced magazine that gives the reader the type of information he wants and needs, in brief, compact, time-saving form.

THE JOURNAL is a member of the A B C , and costs $2 00 per year by subscription

Sheet Metal Worker

45 West 45th Street
New York



SHEET METAL WORKER—established 1874—the oldest publication in its field—is a monthly merchandising, business and technical journal basic to the use of sheet metal in that it serves the various unified merchandising and installing branches of the industry, consuming sheet metal for the erection maintenance and operating equipment of buildings Such equipment includes central air conditioning equipment, warm-air heating, ventilation, dust and refuse removal and systems for handling materials by air, kitchen and restaurant work, a wide variety of interior and exterior sheet metal work for commercial, industrial, institutional, residential and Federal buildings, etc

Subscribers are mainly merchandising-contractors, purchasing practically all products and equipment which they fabricate, erect or install Principal manufacturers of heating and air conditioning equipment receive issues as advertisers or subscribers, leading jobbers also subscribe

The market has three main sub-divisions

(1) Equipment for resale in connection with erection or installation work

(2) Materials for fabrication

(3) Shop equipment and supplies

CIRCULATION

SHEET METAL WORKER, has a total distribution of 8000 monthly, and a paid of 6750, largely in the upper 30 per cent class Its renewal rate has been above 70 per cent even in depression years Sworn figures, January-June, 1936, showed only 367 copies going to manufacturers, jobbers, etc, the balance of 6449 reaching sheet metal, warm-air heating, ventilating, air conditioning, and roofing contractors

EDITORIAL

SHEET METAL WORKER is outstanding in editorial service Its editor is an engineer who has been identified with this journal since 1909 He served as Industrial Advisor under NRA, chairman of the Publication Committee of the *National Association of Sheet Metal Contractors* in charge of producing the book, Standard Practice in Sheet Metal Work, headed the committee of the *National Warm Air Heating and Air Conditioning Association* delegated to publish the Digest of Research, Gravity Warm Air Heating, Engineering Experiment Station, University of Illinois

ADVERTISING

SHEET METAL WORKER has an enviable record of long-term advertising Of the 165 advertisers using space currently in 1936, 41 per cent began from 25 to 62 years ago

Recognition of its position in the fabricating and installation market is indicated by the fact that of the 12 sheet steel companies advertising nationally to this field all are using SHEET METAL WORKER, 9 exclusively Of 32 equipment manufacturers, 22 are exclusive advertisers Similar classifications are equally impressive

SHEET METAL WORKER is well qualified to cooperate with manufacturers regarding sales, merchandising and advertising programs, and invites consultation

Annual subscription rates—$2 00 per year, United States and Mexico, in Canada, $2 50, Foreign, $3 00

Advertising Rates Furnished on Request

American Steam Pump Company

Plant and General Offices. **Battle Creek, Michigan**

DIRECT FACTORY BRANCH
17 BATTERY PLACE, NEW YORK CITY

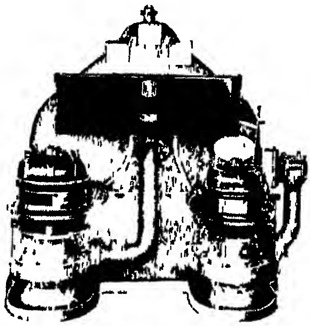
AMERICAN-MARSH PUMPS

SALES AND SERVICE AGENCI
THROUGHOUT THE WORLD

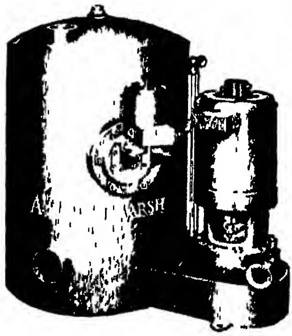
PRODUCTS AND SERVICE—This Company, organized 1873, offers a complete line of centrifugal, steam and power driven pumps Your inquiry for descriptive bulletins or specific recommendation is invited

REDI-VAC HEATING PUMPS—An improved outfit. Removes air and condensation from return lines, discharging air to atmosphere and water to boiler, both functions automatically controlled Bronze fitted throughout No close clearances to wear rapidly, or "freeze up" during idle season—an important advantage over other types, practically eliminating periodic service expense

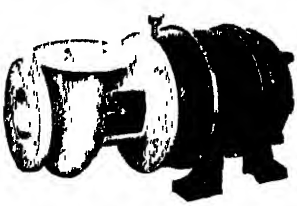
RECIPROCATING VACUUM HEATING PUMPS—Time-tested design and construction Standard equipment includes bronze fittings throughout Motor driven, if desired. (We also build simplex and duplex steam pumps for boiler feed, etc)



Redi-Vac Heating Pump



Redi-Return Condensation Unit

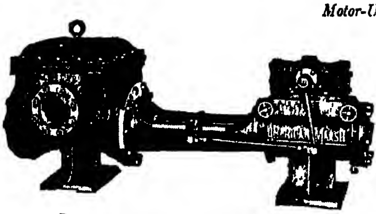


Motor-Unit Centrifugal Pump

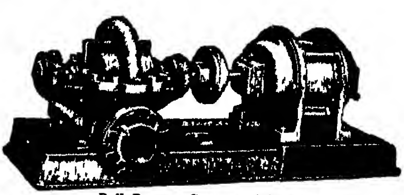
REDI-RETURI CONDENSATIOI UNITS—Compact, de pendable and exception ally low priced. De signed to collect an pump back to the boile returns from high an low pressure heating sys tems, handling water a 210°F Furnished com plete as shown with ful automatic control Twi units can be furnished Large range of sizes

MOTOR-UNI CENTRIFUGAI PUMPS — Simple an inexpensive Ideal fo air washer and coolin work, pumping clea water or brines Pum attached directly t motor Bearings, bas and coupling eliminated Only one stuffing box Very compact Fifteen sizes, 5 to over 500 g p m

BALL BEARING CENTRIFUGAL PUMPS—Single-stage, horizontally split-case type with deep groove ball bearings, stainless steel shaft and bronze seal rings High efficiencies insure low operating cost (We also build multi-stage centrifugal pumps)



Reciprocating Vacuum Heating Pump



Ball Bearing Centrifugal Pump

Buffalo Pumps, Inc.

450 Broadway, **Buffalo, N. Y.**

Branch Offices

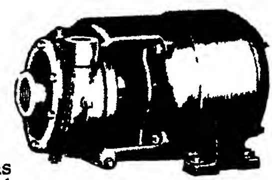
Albany N Y 611 Standard Bldg H S Johnson
Atlanta Ga, 724 First National Bank Bldg, E T Gorbandt
Boston Mass P O Box 71 Melrose Station, E D Johnson
Charlotte N C P O Box 1436 J W Fraser
Chicago, Ill, 20 No Wacker Drive, L D Emmert
Cincinnati, Ohio, Building Industries Bldg, F W Twombly
Cleveland Ohio 418 Rockefeller Bldg, T A Weager
Denver Colo 1718 California St Stearns Roger Mfg Co
Detroit Mich, 2051 W Lafayette Blvd, Coon-De Visser Co T E Coon
Erie, Pa, P O Box 144, F W Allen
Houston, Texas 900 St Charles St, Southern Engine & Pump Co
Indianapolis Ind, 3712 Brill St, C A Merrick
Kansas City, Mo, 315 Dwight Bldg, T H Anspacher
Knoxville, Tenn 903 General Bldg, Buford Bros
Los Angeles, Calif, 708 Pershing Square Bldg, P R Adriance
Minneapolis, Minn, 619 Foshay Tower, E F Bell
Nashville, Tenn, 154 Second Ave, No Buford Bros
New York, N Y, 39 Cortland St W S Kothan
New Orleans, La, Devlin Bros, 1005 Maritime Bldg
Philadelphia, Pa, 703 Cunard Bldg, Davidson & Hunger
Pittsburgh, Pa 912 Fulton Bldg, H L Moore
San Francisco, Calif, 550 Fifth St, Moore Machinery Co J G Scott
Seattle, Wash, 500 First Ave South, A T Forsyth
St Louis, Mo, 1596 Arcade Bldg, J W Cooper
Toledo, Ohio, 1922 Linwood Ave, C M Eyster
Washington, D C, 820 Woodward Bldg, G S Frankel

Complete Line Manufactured in Canada By Canada Pumps Limited, Kitchener, Ontario

PRODUCTS—A complete line of Steam Pumps, Single and Multi-stage Centrifugal Pumps and Special Pumps for special purposes, etc

Descriptive literature furnished on request.

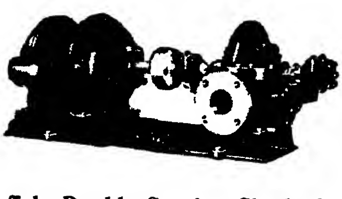
Buffalo Single Suction Close-Coupled Pumps



This pump is close-coupled to electric motor, eliminating the necessity for bearings The impeller is overhung on the motor shaft, providing a compact, easily-serviced unit Permanent alignment is assured and the pump mounted in this manner requires very little space

Buffalo Close Coupled Pumps are suitable for handling hot water with low submergence on suction, or for operating with suction lift as high as 25 ft

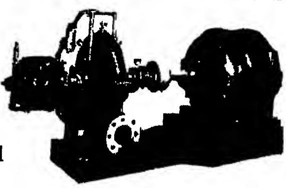
These pumps are also available in special alloys



Buffalo Double Suction Single Stage Centrifugal Pumps

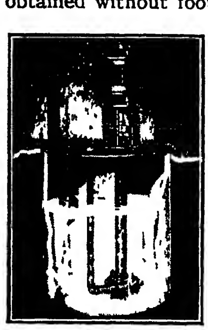
These pumps embody all of the accepted modern features of centrifugal pump design Built for capacities from 10 to 50,000 U S Gal per minute Recommended for almost any service where clear water is handled High efficiency and absolute reliability are assured.

Buffalo Self-Priming Single and Double Suction Centrifugal Pumps



Buffalo Single and Double Suction Pumps can now be had with positive self-priming device built with the pump This primer is built under license from the Nash Engineering Company, and fully covered by patents.

Self-priming pumps have these advantages (1) All working parts are above the liquid to be pumped (2) There is complete access to all parts of installation (3) Rotors are balanced—vibrationless (4) Buffalo Self-Priming Pumps are very quiet—no long shafts to vibrate and fewer bearings (5) Constant positive prime obtained without foot valves



Buffalo Automatic Sump Pumps

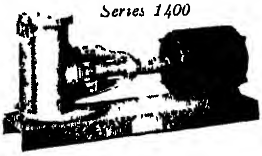
Buffalo Sump Pumps are self-contained and have unusually high efficiencies thus permitting the use of small motors Ball Bearing thrust and enclosed shaft especially adapt these pumps for their service. Operating costs are low, no expensive shutdowns

Decatur Pump Co. Decatur, Ill.

BURKS SUPER TURBINE PUMPS AND WATER SYSTEMS

Self-Priming High Head Units

Series 1400



BURKS
SELF PRIMING
PUMPS
Only one moving part

Series 1100

Burks Self-Priming Centrifugal Pumps

High Efficiency Open-Impeller Type Self-Priming Centrifugal Pumps Built to high Engineering Standards Capacities to 24000 G P H

Burks Condensation Return Unit Serves up to 18,000 Sq Ft Radiation

Furnished with 12 and 24 gal tanks constructed of copper bearing steel Automatic Float Switch governs the operation of motor and also provides overload protection for motor

Designed to operate against a maximum pressure of 100 lb per square inch

Will serve 6000 sq ft of radiation at 10 lb pressure or 1100 sq ft of radiation at 100 lb pressure

Ask for Bulletin No 104 covering larger return units

Burks Heavy Duty Turbine Pumps Self-Priming

A powerful high pressure heavy duty pump

Pressures to 100 lb per square inch Capacities to 1500 G P H A general utility pump suitable for many service applications, including Domestic Water Systems, commercial building installations, booster and hot water service Ideal for returning condensate to heating boilers Pumps air, hence will not vapor-bind Suction lift ability unsurpassed A pump for service where powerful, economical and efficient performance are first considerations A high pressure pumping unit recommended by thousands of Industrials for its outstanding performance and operating economy

BURKS *Autobuilt Deep Well Pump available for depths beyond 28 ft*

BURKS AUTOMATIC
SELF PRIMING

Water System

Has Everything

—Positive Impeller Adjustment
—Self-Priming at Maximum Lift
—Guaranteed 28 Ft Lift
—Completely Automatic
—Manual—Automatic Switch
—Double Suction—Double Discharge
—Large Water Capacity
—Low Maintenance Cost
—Trouble Free Performance
—New Super-Powered Motor
—Steady Even Stream of Water
—Compact—Requires Small Space
—Capacities Up to 500 G P.H

New type motor eliminates radio interference and provides greater hourly water capacity

Available with Larger Storage Tank

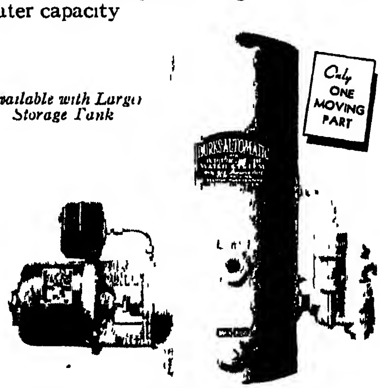


Fig 3336-9

Equipped With The Famous Burks Auto-Air Control

Chicago Pump Company

SEWAGE - CONDENSATION - CIRCULATING
BILGE - FIRE - HOUSE - VACUUM

2330 Wolfram Street **BRUnswick 4110** **Chicago**

PRODUCTS—Return Line Vacuum Heating and Boiler Feed Pumps, Condensation, House, Booster, Fire Pumps, Circulating, Brine, Sewage, Bilge, Sludge, Pneumatic and Tankless Water Supply Systems and Automatic Alternator.

Condensation Pump and Receiver

for Low, Medium and High Pressures
Systems up to 150,000 Sq Ft Radiation

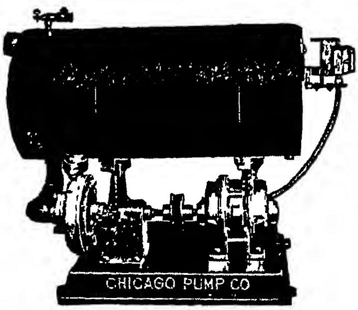


Fig 1931—F C Condensation Pump

"Chicago" Condensation Pumps are built for systems ranging from 2,000 up to 150,000 sq ft of radiation, and for boiler pressures up to 200 lb Units are built in either single or duplex—the duplex being alternated in their operation by the Automatic Alternator *For tables and complete description ask for Bulletin 129*

Vertical Condensation Pump

for Low and Medium Pressure for Systems up to 100,000 Sq Ft Radiation

Fig 1940 Vertical Condensation Pump

The vertical condensation pump is designed to receive returns from lowest radiation The receiver is placed underground—an ordinary hole sufficing if necessary — and requires very little floor space Unit is shipped complete, easy to install, assembled so as to prevent steam leaks Special bearings will stand up under hot water for several years A special float mechanism is guaranteed not to leak or stick in stuffing box *Complete data and description in Bulletin 133*

"Sure-Return" Condensation Pump

for Low and Medium Pressure, and Systems up to 35,000 Sq Ft Radiation

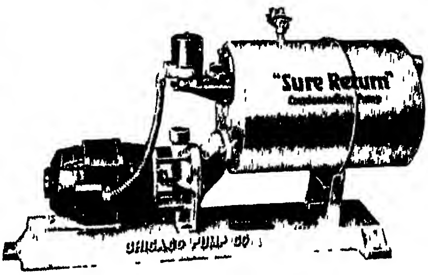


Fig 1946

"Sure Return" Condensation Pumps and Receivers are built for systems up to 35,000 sq ft of direct radiation and for low and medium pressures Built in either single or duplex units Duplex units are alternated in their operation by the Automatic Alternator *Complete data in Bulletin 131*

Horizontally Split Pumps

for all Services

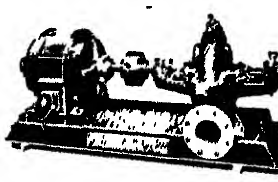


Fig 1881—Single Stage Type "D" Pump

For any service (such as boiler feed, water supply, tank filling, circulating, fire protection, etc) Chicago Pump Co builds a line of horizontally split case centrifugal pumps in both single and multistages Completely bronze fitted (except where special fittings are required) ball bearings, internal water seal, oil is filtered "Chicago" Horizontal Pumps are built for efficient performance and long life

"CONDO-VAC"

Return Line Vacuum Heating and Boiler Feed Pumps

Automatic Alternator is available on Duplex Return Line Vacuum Heating and Boiler Feed Pumps

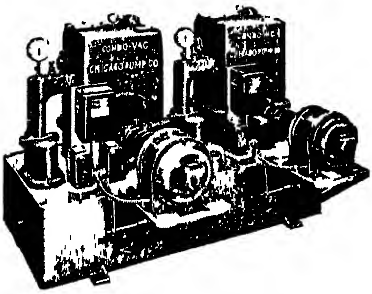


Fig 2102—Duplex "Condo-Vacs" with Duplex Double Automatic Control

Sewage Ejectors

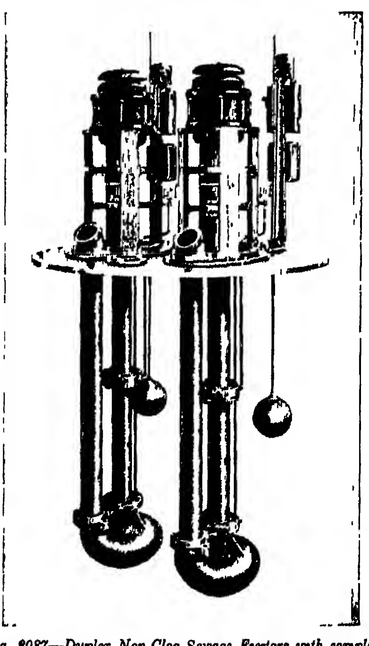


Fig 2087—Duplex Non-Clog Sewage Ejectors with complete control equipment mounted on basin cover "Automatic Alternator" transfers operation from one pump to the other

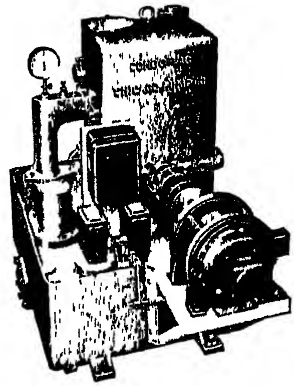


Fig 2100—Single "Condo-Vac"

No vacuum on stuffing boxes, ample clearance in rotating member. It costs less to operate a Condo-Vac Condo-Vac reduces corrosion in piping and boiler to minimum —because pump does not take in air from atmosphere and entirely eliminates all air coming back from system. Condo-Vac is quiet, has a low inlet, entirely automatic, fool-proof, easy to maintain *Ask for Bulletin 137*, learn more about the modern vacuum pump with the long life principle of operation

Close-Coupled Pumps

Boiler Feed, Circulating, Tank Filling, Water Supply

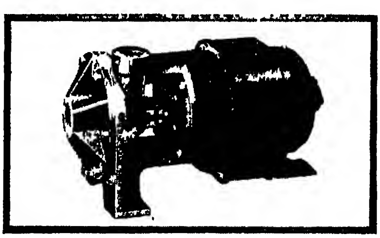


Fig 2130—Close-Coupled side suction pump Capacities range from 3 to 600 G P M against heads up to 180 ft Motors from 1/6 to 20 Hp Discharge 1 to 3 in Both closed and open type impellers

The Nash Engineering Company

South Norwalk, Conn., U. S. A.

Sales and Service Offices in all Principal Cities

Centrifugal Pump

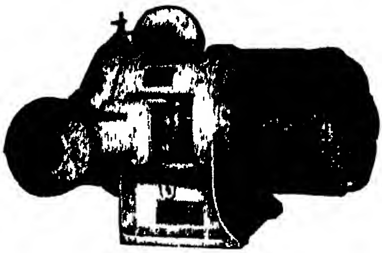
Made in standard and suction (self-priming) types For circulating hot and cold water, boosting city water pressure, handling water in air washing and conditioning, handling ash sluicing water, etc

Compact—motor armature and pump impeller are mounted on the same shaft Simplified—no bearings in pump casing, one stuffing box Accessible—impeller removable without disturbing piping or shaft alignment

Self-priming types will handle air or gas continuously with liquid being pumped, and can be operated intermittently without foot valve

Supplied in 1¼, 1½, 2, 3, 4, 6, and 8 in sizes with capacity up to 2000 g p m Heads up to 300 ft

Complete data in Bulletin No 155 on request



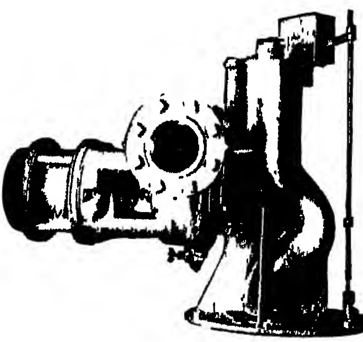
Suction Sump and Sewage Pumps

Jennings Suction Sump Pumps are self-priming centrifugals for handling seepage water and liquids reasonably free from solids The Suction Sewage Pumps are equipped with a non-clog type impeller for liquids containing solids Suction piping only is submerged Centrifugal impeller and vacuum priming rotor are both mounted on same shaft that carries rotor of the driving motor, forming a single moving element and rotating without metallic contact

These pumps will handle air or gas with liquid being pumped, and because of self-priming feature are installed entirely outside of pit This affords perfect accessibility for inspection or cleaning

Capacities to meet all requirements

Complete data in Bulletins 159, 161 and 188 on request



Air Compressor and Vacuum Pump

The Nash Air Compressor operates on a unique and different principle The one moving part rotates in casing without metallic contact There are no valves, pistons, or sliding metal vanes There is nothing to wear, and no internal lubrication Nash Compressors deliver absolutely clean air

Unit illustrated is built integral with electric motor. Compact, may be installed anywhere Ideal general service compressor. Suitable for priming pumps on water systems, handling CO^2 gas, agitation of liquids, as blood sucking pumps in hospitals, etc

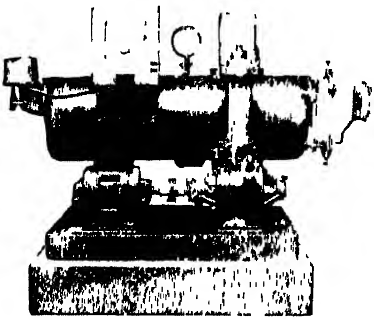
Pressure 75 lb or vacuum 28 in of mercury Equipment furnished for any capacity.

Complete data in Bulletins 258 and 261 on request

The Nash Engineering Company

South Norwalk, Conn., U. S. A.

Sales and Service Offices in all Principal Cities



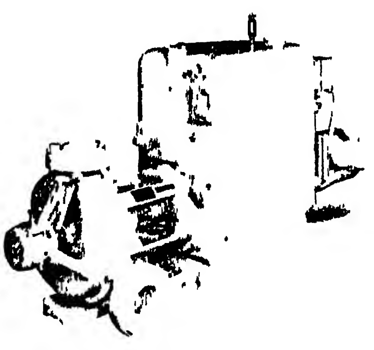
Jennings Return Line Vacuum Heating Pumps

Standard with the heating industry for over sixteen years They remove air and condensation from the return lines of vacuum steam heating systems, discharging the air to atmosphere and returning the water to the boiler

Two independent units are combined in a single casing—an air unit and a water unit Impellers of both are mounted on the same shaft The pump is bronze fitted throughout

Supplied either direct connected to standard electric motors, for belt drive, or for steam turbine drive For continuous or automatic operation against pressures up to 40 lbs Supplied standard in capacities up to 300,000 sq ft. E D R

Complete data in Bulletin No 85 on request

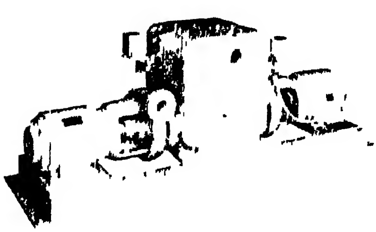


Jennings Vapor Turbine Vacuum Heating Pumps

The Jennings Vapor Turbine Heating Pump combines all of the advantages of the standard return line heating pumps with a new type of drive, a specially designed low pressure turbine which operates directly on steam from the heating mains on any system, requiring a differential of only 5 in. of mercury, and returns that steam to the heating system with practically no heat loss

This pump affords the safety and economy which goes with a continuous condensation return and steady vacuum, and at no cost for electric current Furnished standard in capacities up to 65,000 sq ft E D R Over 65,000 sq ft and up to 150,000 sq ft. information will be furnished upon request.

Complete data in Bulletin No. 246 on request.



Condensation Pump and Receiver

Removes condensation from radiators in return line steam heating systems and pumps condensation back to the boiler

They are sturdy and compact in construction, and combine receiving tank, pump and driving motor in a single assembly. Bronze fitted throughout, with Tobin bronze shaft Impeller is of special design adapted to handling hot water with highest efficiency

Jennings Condensation Pumps are furnished in standard sizes with capacities ranging from 1½ to 225 g.p m of water For serving from 1,000 up to 150,000 sq ft of equivalent direct radiation.

Complete data in Bulletin No 241 on request

The John H. McGowan Company

General Offices 54-58 Central Avenue

Cincinnati, Ohio

CENTRIFUGAL-STEAM AND POWER DRIVEN-TYPES

FOR WIDE RANGES OF APPLICATION

CENTRIFUGAL PUMPS

Side Suction Type

Double Suction Split Case Type

All units embraced in the various groups of this type may be furnished for stated capacities and heads and equipped with any suitable type of drive desired

Side Suction—Single Stage—Capacities 5 to 125 gpm for heads up to 70 ft *Bulletin No 100*

Double Suction—Split Case Single Stage –30 to 3500 gpm for heads 10 to 131 ft *Bulletin No 104B*

Multi-Stage—Split Case Single Suction Pumps Capacities 120 to 1000 gpm and for heads up to 680 ft *Bulletin No 106*

SPECIAL CENTRIFUGAL PUMP UNITS

Condensate Return Units—Horizontal Pump and Receiver, Motor Driven for 1000 to 100,000 sq ft radiation and return pressures up to 150 lb

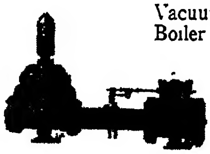
Vertical Type for 3000 to 100,000 sq ft radiation and for return pressure up to 50 lb

Vertical Sump Pumps—Single and Duplex for capacities from 10 to 500 gpm and for heads up to 108 ft

Multi-Stage—Split Case Single Suction Pumps Capacities up to 3500 gpm and for heads up to 1630 ft

SINGLE STEAM DRIVEN PUMPS

Vacuum Pumps for radiation up to 190,000 sq ft *Bulletin No 40*



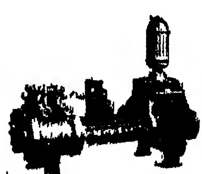
Single Steam Driven Boiler Feed or Circulating Pumps

Single Steam Driven Vacuum Pump

Boiler Feed Pumps—Packed Piston, Valve Plate Design for Boilers up to 1300 hp and pressures up to 200 lb, including General and Low Service Pumps up to 350 gpm and for pressures up to 150 lb *Bulletin No 43-A* End Packed Plunger Types built for corresponding capacities and pressures

DUPLEX STEAM DRIVEN PUMPS

Duplex Condensate Pump and Receiver



Duplex Boiler Feed and Circulating Pumps

Include Packed Piston Valve Plate Design Boiler Feed Pumps—for Boilers up to 3000 hp and for water pressures up to 200 lb *Bulletin No 1*

Packed Piston Valve Plate Design Low and General Service Pumps for capacities up to 350 gpm and for pressures from 100 to 150 lb *Bulletin No 2*

Automatic Condensate Return Units Packed Piston Valve Plate Design for return pressures up to 200 lb *Bulletin No 3*

Center Packed Plunger Pump Valve Plate and Straightway Design for Boiler Feed or General Service suitable as Boiler Feeders up to 6000 hp and for pressures up to 200 lb *Bulletin No 4*

End Packed Pot Valve Pumps for Boilers up to 2000 hp and for pressures up to 300 lb *Bulletin No 5*

Packed Piston Turret Type, General Service Pumps for capacities up to 700 gpm and for pressures up to 150 lb *Bulletin No 6*

Fuel Oil Heater Sets, Single or Double in Packed Piston Valve Plate Design *Bulletin No 46.*

All pumps fully guaranteed for service applying to sale

Anemostat Corporation of America

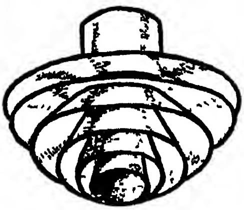
551 Fifth Avenue, **New York City, N. Y.**

THE ANEMOSTAT AIR DISTRIBUTOR

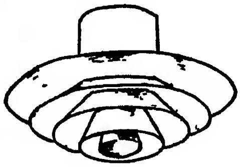
Anemostat

Anemostat is a device of unique design for providing draftless air distribution Mixing room air with conditioned air within the Anemostat permits the engineer to specify high air velocities in ducts, great range temperature differential, resulting in small ducts, smaller plants, reduced operating expense

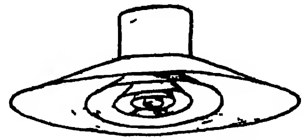
The Anemostat can be connected to the air outlets of any existing or new system



Type A
For Industrial Use



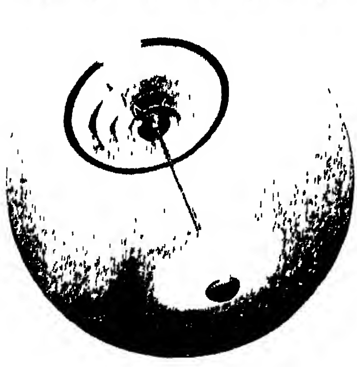
Type B
Standard



Type C
For Residential Use

Anemostat Velocities Feet per Minute (All types)

Size No	Neck Dia Inch	800	900	1000	1100	1200	1400	1600	1800	2000	2200	2400	2600	Area of Neck Sq In
		Velocities Feet per Minute												
		Output Cubic Feet per Minute												
10	4	69	78	87	95	104	122	139	157	174	192	209	227	12
15	6	157	176	196	215	235	274	314	353	393	432	471	510	28
20	8	279	314	349	383	419	489	558	629	698	767	837	906	50
22.5	9	353	399	442	486	531	619	708	796	884	972	1061	1150	63
25	10	436	490	546	599	655	764	873	982	1092	1200	1320	1419	78
30	12	628	706	786	863	944	1100	1258	1415	1572	1730	1888	2042	113
35	14	855	962	1069	1175	1280	1495	1710	1920	2138	2350	2562	2780	153
40	16	1116	1256	1396	1535	1674	1954	2232	2510	2792	3070	3350	3630	201



Anemostat in combination with lighting fixture, Julius Kayser Stores

Anemostats in Madison Square Garden Installation

"No Air Conditioning System is better than its Air Distribution"

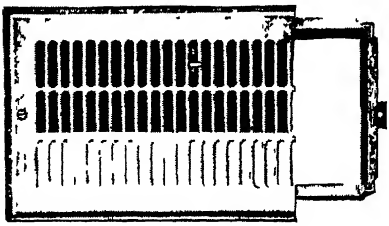
The Auer Register Co.

3608 Payne Avenue, **Cleveland, Ohio**

Manufacturers of Registers and Grilles for Gravity and Air Conditioning Systems; Wrought Metal Grilles for Concealing and Protecting Radiation

AIR CONDITIONING REGISTERS AND GRILLES

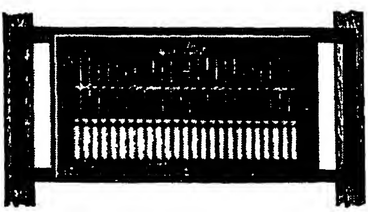
Auer Registers and Grilles for Air Conditioning have been designed to meet all modern requirements—appearance, practicability, and simplicity in installation and operation They are made in various designs to harmonize with modern interiors Constructed and tested to operate with high efficiency in any system of forced air, they are a departure, not merely an adaptation, from gravity registers They are designed for easy installation in either new construction or remodelling



No. 2030 Sidewall

The design shown is our standard design, termed *Classic* It is very adaptable to most interiors, other designs, however, can be furnished

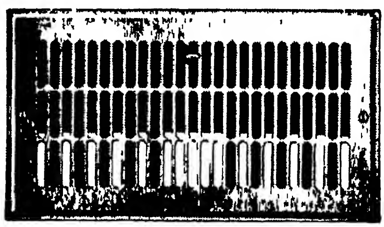
This type is made with *double* band frame for extra rigid installation and for elimination of any possible warping of the face The same model can be used without a valve, as a return



No. 2010 Sidewall

This type is used in new construction exclusively. The sliding frame is anchored to studdings before plastering After plastering, the register is easily attached to the frame and can be slid to the position desired between the studs

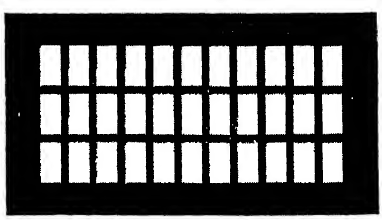
The same type furnished without a valve is used as a return



No. 2230C Sidewall

This illustration shows register directing air in three directions

It can be made to direct air in one direction only or in two directions The directing air blades can be set at any angle specified Installation can be made with band frame or sliding frame as shown in previous illustrations



No. 2005 Oblong

Auer Registers and Grilles can be furnished in many designs

Auer Registers and Grilles can be furnished in any standard finish Illustrated catalogue showing complete line of registers for all purposes and chart of open areas and capacities will be forwarded on request.

The Independent Register Co.

ESTABLISHED 1898

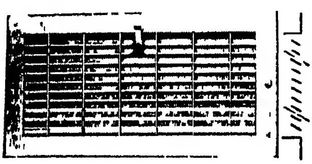
3753 East 93rd Street, **Cleveland, Ohio**

INDEPENDENT "Fabrikated"

Reg U S Pat Office

AIR CONDITIONING REGISTERS AND GRILLES

No. 311-A—ADJUSTABLE DIRECTED AIR FLOW



No 311-A—Air Flow Downward Adjustable from straight to 45 deg

With "INDEPENDENT" Adjustable Directed Air Flow Registers and Grilles the Engineer is in complete control of the direction of air flow

The directional adjustment can be made at the time of installation —or after the system is operating Thus correcting unforeseen, or changed conditions

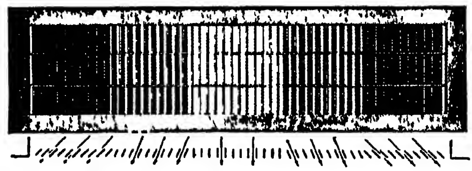
Each interior grille bar is adjusted individually

The method of adjustment is simple as shown, and many directions and combinations can be developed to suit the need

No. 321-A—ADJUSTABLE DIRECTED AIR FLOW

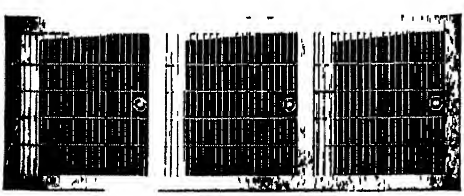
Standard registers are furnished with single valves,—they are also supplied with "Multiple Valves"—and with "Multiple Valves" adjusted individually—the Engineer is given a dual control of the air flow—being able to secure right and left together with up and down deflection at the same time

Grille Bars set for right and left deflection The grille bars are set in a firm tension, yet easily adjusted, with the tool sent with each order

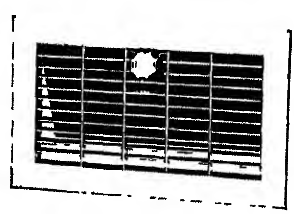


No 321-A—Showing a combination of Adjustments

REGISTERS WITH TANDEM VALVES



Knob Control



Independent Catalogues Tell The Complete Story—Yours For The Asking.

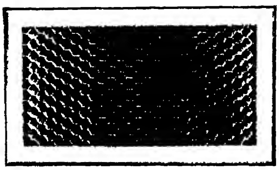
Hart & Cooley Manufacturing Co.

Established 1901

Air Conditioning Registers and Grilles - Warm Air Registers
Damper Regulators - Furnace Regulators - Pulleys - Chain

61 West Kinzie Street, **Chicago, Ill.**

A COMPLETE LINE OF AIR CONDITIONING GRILLES AND REGISTERS TO MEET ANY NEED



No. 90 DESIGN—
Dual Control Directional Flow

No 90 Design Grilles and Registers are made up of a number of thin strips, which are shaped into a series of grooves The strips, when assembled, form an attractive grille with openings ½ in wide and 1 in in depth The tubes formed by the grooves in the strips may be straight, at an angle of 45 deg to the left or right, or up and down, or in any combination thereof, thus making it possible to secure positive control of the air flow in any desired direction

Characteristics of No. 90 Design

1. **Concealment of Duct**—The depth of the grille (1 in), together with the tubular shape of the openings, result in exceptional concealment of the duct

2. **Dual Control of Air Flow**—The air is controlled in two planes - horizontally as well as sideways The air leaves the grille in a horizontal plane regardless of the approach.

3. **Resistance**—Directional flow is obtained by changing the path of the air gradually. The turbulence which would be caused by an abrupt change in the path of the air is eliminated, thereby greatly reducing the resistance of the grille

4. **Noise**—The elimination of turbulence likewise eliminates the greatest cause of noise in the grille Velocities of 2000 ft per minute or more may be used without adding to the noise level Complete acoustical ratings and their application are shown in H & C Bulletin No. 1

5. **Air Capacity**—The grille itself has a free area of approximately 89 per cent When used with the 3-piece and 1-piece frames the free area is slightly reduced, owing to the clearance necessary for the frames.

6. **Directional Control**—The grille is available with a type of directional control for every condition.

No. 77 DESIGN—Non-adjustable Vertical Bar Close Mesh

No. 78 DESIGN—Non-adjustable Horizontal Bar Close Mesh

Nos 77 and 78 Design Grilles and Registers are made up of fixed bars with ⅜ in mesh These grilles and registers fulfill the demand for an inexpensive yet attractive register with straight air flow and duct concealment

No. 72 DESIGN—Non-adjustable Vertical Bar Open Mesh

No 72 Design Grilles and Registers are made up of fixed bars with ¾ in mesh These registers are of rigid construction and offer a maximum of free area

No. 84 DESIGN -Adjustable Vertical Bar Close Mesh

No. 85 DESIGN—Adjustable Horizontal Bar Close Mesh

Nos 84 and 85 Design Grilles and Registers are made up of solid bars, rounded at the front and back to offer a minimum of resistance to air flow The construction allows for adjustment of the bars after installation has been completed, to provide any necessary air deflection The bars are connected in 2 in groups to facilitate ease of adjustment and allow a wide range of deflection settings

Characteristics of Nos. 84 and 85 Design

1. **Concealment of Duct** The depth of the bars together with their close spacing ($^3/_8$ in mesh), result in exceptional concealment of the duct

2. **Air Capacity** The grille itself has a free area of approximately 85 per cent When used with installation frames the free area is slightly reduced owing to the clearance necessary for the frames

3. **Adjustable Directional Control** The bars may be adjusted, before or after installation, to give any type of directional control desired

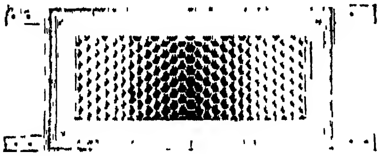
No. 71 DESIGN --Perforated Plain Lattice

No 71 Design Grilles and Registers are easily decorated to match their surroundings, and the openings ($^3/_8$ in) are small enough to conceal the duct They result in a satisfactory installation in those cases where directional control of air flow is not required

No. 76 DESIGN- Fancy Perforated

No 76 Design Grilles and Registers are attractive in appearance and will harmonize with any style of architecture

Six Types of Air Conditioning Registers Available



All of the designs described are available as grilles or registers, with or without valves, and are likewise available with any of the six types of frames described below

(1) Sidewall Register with Streakproof Frame Frame and duct are embedded in plaster Overlapping face with rubber gasket insures a streakproof installation

(2) Sidewall Register with Band Iron Frame Ideal for old house installations Face has ample margins to cover opening in wall

(3) Baseboard Register with Streakproof Frame - Duct is permanently secured to frame, eliminating streaking Face is readily removable for cleaning or redecorating

(4) Baseboard Register with Integral Frame and Face -Easy to install and less expensive

(5) Return Air Intake with $^7/_8$ in Projection Designed for use where intake extends above the top of the baseboard

(6) Return Air Intake, Flat Designed for installations in which the baseboard height is greater than that of the intake

Complete Separate Catalogs on Air Conditioning Registers and Grilles or Warm Air Registers Available on Request

Tuttle & Bailey, Inc.

New Britain, Conn.

Branch Offices BOSTON, NEW YORK, CHICAGO, PHILADELPHIA

Air Conditioning	**Ornamental Grilles**
Grilles and Registers	**Cast or Wrought Metals**
Air Control Devices	**Convection Heaters**

PRODUCTS

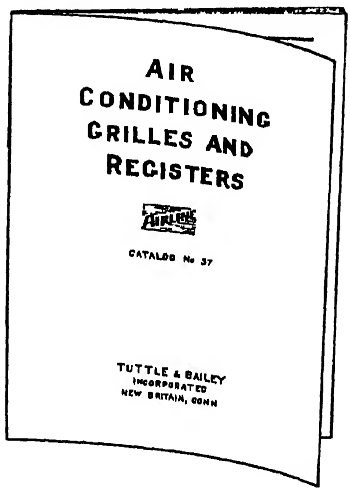
Tuttle & Bailey Engineers have developed a complete line of grilles, registers and air control devices for air conditioning work In addition, new items are constantly being added to the line in an endeavor to satisfy almost any requirement on an installation

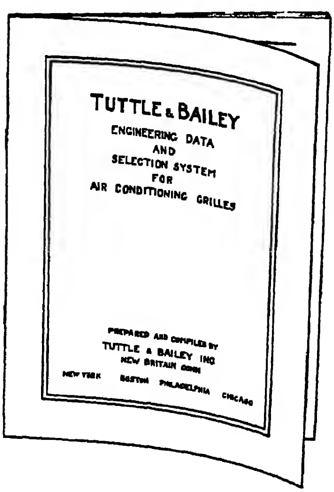
A NEW CATALOG!

Tuttle & Bailey, Inc announces Catalog No 37 covering its entire line of Air Conditioning Grilles and Registers as well as other air control devices

This new catalog contains complete information on Tuttle & Bailey's comprehensive line of Forced Air Registers, Air Conditioning Grilles, Ducturns, Santrols, Ceiling Outlets, and many specialties

For complete information on a modern and up-to-date line, write for your copy of Catalog No 37





ENGINEERING DATA

Tuttle & Bailey Engineers have developed after many months of constant experimenting in their research laboratory, a brochure of engineering data on air action from a grille or register outlet In this special booklet is fully described a new system for the proper selection of grille sizes and constructions to meet specified conditions of air volume, air throw, etc

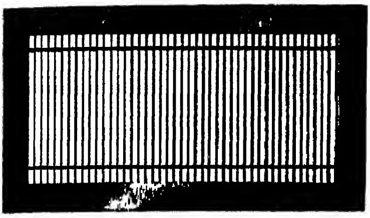
The engineering data is not theoretical but was determined from actual tests run in Tuttle & Bailey's laboratory under conditions which would resemble an actual job installation

Copies of this special booklet are available free of charge to Engineers Write for "Engineering Data and Grille Selection System"

Tuttle & Bailey, Inc.

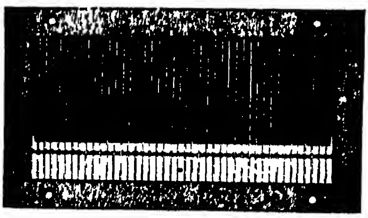
New Britain, Conn.

Listed on this page are suggestions of a few of the more popular T & B products available for air conditioning installations For complete data, refer to Catalog No 37



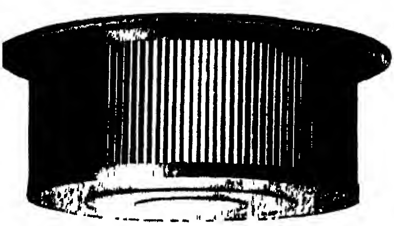
AIRLINE GRILLE

A popular style among the various air conditioning grilles is the T & B Airline design This grille is unique in design, neat and attractive, and provides fixed air deflection, from one to seven air streams may be directed from a single outlet



FORCED AIR REGISTERS

Tuttle & Bailey have a complete line of Forced Air Registers for use on individual home construction There are six standard face designs all available in various types of sidewall or baseboard frames

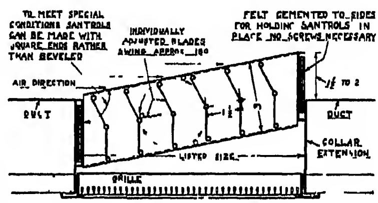


CEILING OUTLET

Tuttle & Bailey have two standard ceiling outlets, one made from steel and the other from cast aluminum both furnished with or without cone dampers

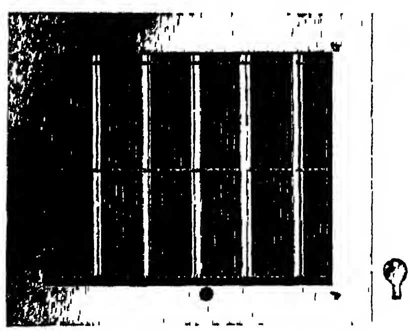
HIVELAIR GRILLE

The Hivelair grille is an entirely new grille especially designed for use on high velocity systems Its special construction controls air throw and also permits the use of air velocities as high as 2500 fpm



SANTROLS

A new device for use on an exterior duct system of an air conditioning installation Santrols provide volume control and uniform distribution of the air from the outlet.



McKNIGHT REGISTERS

McKnight registers are a tested and proved device for perfect control of air volume on an air conditioning installation All adjustment is made at the outlet and accomplished by merely turning a key

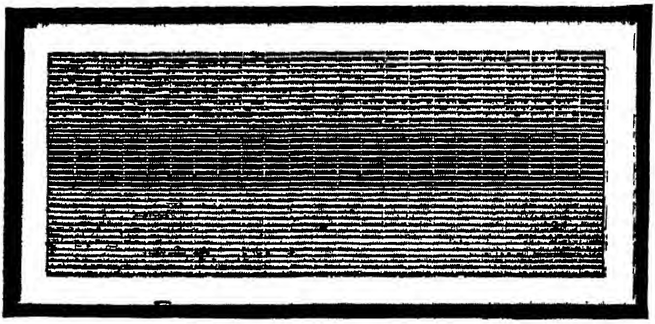
For complete information on all Tuttle & Bailey products for air conditioning work, write for the new Catalog No 37

The Waterloo Register Company

Waterloo, Iowa **Seattle, Wash.**

Established 1902

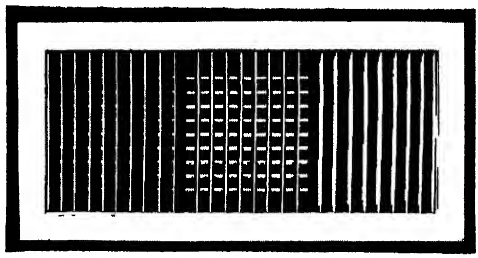
AIR-MASTER GRILLES



Depth ¾ inch—Blade Depth 1 1/16 inches

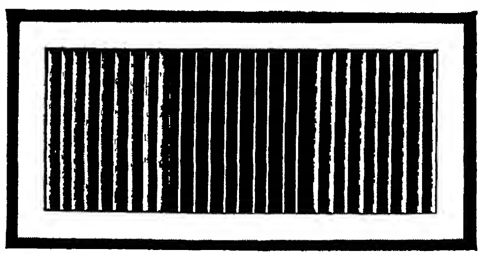
FG-90 Permanently fixed exterior bars spaced at ¼ in intervals are supported by individually adjustable interior blades providing fixed "spray nozzle" control on one plane and adjustable deflection control on the other plane Any arrangement or angle setting of the fine exterior bars can be furnished, either parallel with short dimension or long dimension of grille Absolute area 74 per cent Effective area 87 per cent

FG-80—Double grilles, such as the AIR-MASTER "80" are arranged with two panels of individually operated blades described as "rear" and "front" deflectors, making possible complete deflection control The "80" has blades spaced at ½ in intervals with maximum screening for high velocity air stream Absolute area 39 2 per cent Effective area 69 6 per cent Air-Master FG-70 with bars spaced at 1 in intervals has absolute area 67 5 per cent and effective area 83 7 per cent All blades formed with tube edge exposed to air current



Depth Including Blades 2 inches

FG-60—Individually operated blades formed from same stock as in The AIR-MASTER "80" but incorporating one panel of bars instead of two Bars are spaced at ½ in. intervals and may be ordered parallel with either long or short dimension of grille, thus control is complete, either laterally or vertically whichever is desired



Depth Including Blades 1 inch

All grilles illustrated are intended primarily for commercial installations, and may be furnished in any finish to fit any size of opening which measures in even inches A complete line of residential air conditioning registers is manufactured and also available are latest type floor registers, cold air faces, modernistic grilles, gravity sidewall registers and accessories

The American Rolling Mill Company

Executive Offices, Middletown, Ohio

Atlanta, Ga,	1437 Citizens and Southern National Bank Bldg	Kansas City Mo	7100 Roberts St
Boston, Mass	201 Devonshire St	Middletown Ohio	703 Curtis St
Buffalo, N Y	501 Seventeen Court St Bldg	New Orleans, La	3505 S Carrollton Ave
Chicago, Ill	310 S Michigan Bldg	New York, N Y	50 Church St
Cleveland, Ohio	1516 B F Keith Bldg	Philadelphia, Pa	1808 Lincoln Liberty Bldg
Dallas, Texas	1111 Santa Fe Bldg	Pittsburgh, Pa	1652 Oliver Bldg
Detroit, Mich	5-261 General Motors Bldg	San Francisco Calif	540 Tenth St
		St Louis, Mo	1725 Ambassador Bldg

ARMCO

Choose the Correct Armco Grade

These grades of Armco Sheet metal are recommended for the air conditioning applications shown For detailed information get in touch with the nearest district office or write direct to the American Rolling Mill Company, Middletown, Ohio

Armco Ingot Iron

(Galvanized)

Ducts
Washer Chambers
Plenum Chambers
Steam Line Casings
Furnace Casings
Spray Towers
Drip Pans
Housings
Machine Guards
Unit Conditioners
(Industrial)
Roof Ventilators
Eliminator Blades

Hot Rolled

(Sheets and Strip)

Fan Blades
Blower Casings
Fuel Oil Tanks
Unit Conditioners
Stoker Hoppers

Cold Rolled

(Sheets and Strip)

Furnace Casings
Room Unit Casings

Plates

(Armco Ingot Iron)

Smoke Stacks
Coal Hoppers
Breeching
Unfired Pressure Vessels
Low-fired Boilers
Tanks

Stainless Steel

(Sheet, Strip and Plate)

Furnace Construction
Heat Flues and Tubes
Humidifier Pans
Casings for Room Controls
Furnace Casing Trim
Grilles
Heat Resistance
Corrosion Resistance
Fan and Blower Blades

Spiral Welded Pipe

Air Piping
Water Piping
Low Pressure Steam
Cooling Tower Piping
Spray Pond Piping

Other Armco Products

The grades recommended for these applications are only a few that Armco makes Others include copper bearing sheets and plates and open hearth steel, either galvanized or uncoated Armco galvanized Paintgrip sheets are recommended where immediate painting is necessary They require no acid treatment or natural weathering and help prolong paint life Put your problem up to Armco—and remember there are distributors and contractors nearby to meet quickly your needs for the sheet metal you specify

USS COPPER-STEEL SHEETS—BLACK AND GALVANIZED

Below are some questions which users of steel have asked us The answers have been supplied by USS metallurgists who have tried to bend over backwards to be conservative

What is Copper-Steel?

An alloy of copper and steel, wherein a very small percentage of copper is advantageously used to impart its natural rust resistance to a very large quantity of steel

What are USS Copper-Steel Sheets?

The product of the subsidiaries of the United States Steel Corporation, the original manufacturers of copper-steel sheets The proper blending of copper with well made steel

To what extent have the aims of the metallurgist been realized?

By the use of a small amount of copper dissolved in molten steel, the life of unprotected steel may be said, conservatively, to have been doubled, and this has been accomplished at an additional cost of a few per cent

Why should copper be used in the steel base if the sheets are galvanized, or painted, or both?

Because these protective coatings are impermanent at best, and wherever they have become damaged or removed, the bare metal exposed needs the protection that copper gives it against the concentrated corrosive attack

The information contained in these questions and answers may well save you thousands of dollars a year With an increase in service of at least twice as much as against a slight increase in cost, USS Copper-Steel is particularly interesting to heating, ventilating and air conditioning engineers, architects and contractors

For durable ducts, demand USS Copper-Steel Sheets

Gauges of Steel Sheets Used for Duct Construction

HEATING AND VENTILATING				Planing Mill and Other Exhaust Systems	
Round Ducts		Rectangular Ducts			
Diam , Inches	Gauge	Width, Inches	Gauge	Diam , Inches	Gauge
6 to 19	26	4 to 18	26	Up to 8	24
20 to 29	24	19 to 30	24	9 to 14	22
30 to 39	22	31 to 60	22	15 to 20	20
40 to 49	20	61 to 118	20	21 to 30	18
50 and above	18	118 and above	18		

In the above table rectangular ducts are to have cross breaks for the gauges shown, otherwise two gauges heavier should be used One inch standing seams should be used on widths up to 48 in , 1½ in seams on widths over 48 in , and for widths over 60 in , the seams should in addition be provided with reinforcing bars or angles

(This material is reprinted by permission from "Fan Engineering," Buffalo Forge Co)

USS Black and Galvanized Sheets

Two principal types of black sheets are used by air conditioning engineers They are USS Hot Rolled and USS Hot Rolled Annealed These steel sheets may be had in a number of different finishes, suitable for all sorts of forming operations and for painting

Remember that every kind of USS Black Sheet is available in Copper-Steel Modern makers and users of air conditioning equipment are specifying USS Copper-Steel Black Sheets for blowers, refrigerator cabinets, dust collectors, tube collectors, fans, ducts and a thousand other specialty products

Five types of galvanized sheets made by USS subsidiaries are used extensively in heating and ventilating work The best types for your particular products or installations may be ascertained by consulting USS engineers Write the nearest branch office

Milcor Steel Company

Milwaukee, Wisconsin **Canton, Ohio**

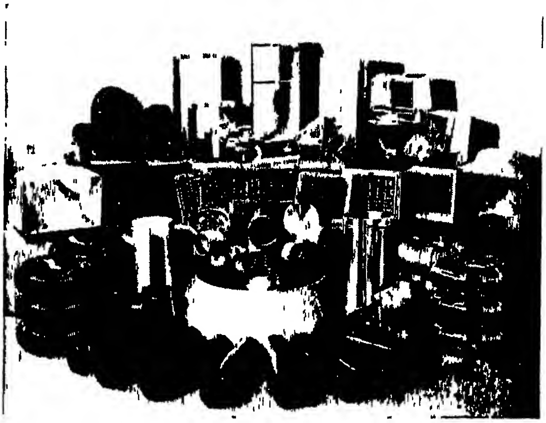
CHICAGO, ILL KANSAS CITY, MO LA CROSSE, WIS

Milcor Manufactures a Complete Line of Furnace Pipe and Fittings, Forced Air Pipe.

For thirty-five years Heating Contractors have known they can always depend upon Milcor for

1. Quality of Product. During its 35 years in business, Milcor has constantly maintained high standard of quality in its products

2. Outstanding Features. Milcor Products have exceptional features in construction and design which enable quicker and better installations



A COMPLETE LINE OF FORCED AIR PIPE AND FITTINGS

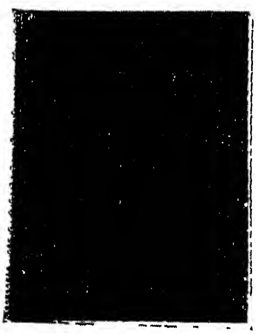
The most exacting air conditioning engineer will be thoroughly satisfied with the complete line and appreciate the many possibilities of installation afforded

Milcor has just produced a new booklet giving all necessary data including list prices on these products **Send for your copy today!**

MILCOR GALVANIZED ROUND FORCED AIR PIPE

The efficiency of Round Pipe is appreciated by every heating contractor Now with the new Milcor long-length Galvanized Round Furnace Pipe, installations can be made for Forced Air Systems which will provide greater efficiency at lower installation and material cost Made with air-tight seams and of high quality material, heavily galvanized

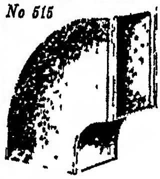
Send for Your Copy of the New No. 35 Catalog



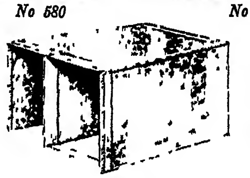
Equivalent Round and Rectangular Ducts —for Equal Friction

Dia Round Pipe	Area	Size Rectangular Pipe	Dia Round Pipe	Area	Size Rectangular Pipe	Dia Round Pipe	Area	Size Rectangular Pipe
5″	21 2	3x 8	12″	113 7	10x12	22″	376 7	12x35
6″	29 2	4x 8	14″	158 4	8x22	22″	376 7	14x29
7″	37 4	6x 7	14″	158 4	10x17	24″	444 9	12x42
8″	40 7	5x10	16″	198 6	8x28	24″	448 6	14x35
8″	52 8	7x 8	16″	203 5	10x22	24″	449 9	15x32
9″	66 5	7x10	18″	254 5	10x28	26″	526 9	14x42
9″	65 0	8x 9	18″	248 8	12x22	26″	522 8	15x38
10″	78 5	7x12	20″	307 9	10x35	26″	574 7	16x35
10″	75 4	8x10	20″	307 9	12x28	28″	594 0	15x44
11″	96 8	8x13	20″	314 0	14x24	28″	594 0	16x41
11″	102 1	9x12	22″	373 3	10x43	28″	594 0	17x38
12″	111 2	8x15						

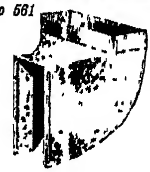
No 515



No 580



No 561



Armstrong Machine Works

851 Maple Street **Three Rivers, Mich.**

Exclusive Manufacturers of Armstrong Inverted-Bucket Steam Traps

ARMSTRONG REPRESENTATIVES

Atlanta, Ga, J M Tull Metal & Supply Co, Inc, 285 Marietta St, N W

Baltimore, Md, Milby & McKinney, 218 Water St

Birmingham, Ala, Southeastern Products Co, 1401 Lomb Ave

Boston, Mass, Files Steam Specialty Co, 261 Franklin St

Buffalo, N. Y., Herr Steam Specialty Co, 360 Warwick Ave

Charleston, W Va, Baldwin Supply Co, 518 Capitol St

Chicago, Ill, Barrett-Christie Co, 108-112 N Clinton St

Dallas, Texas, Geo B Allan & Co, North Texas Bldg

Denver, Colo, Hendrie-Bolthoff Mfg & Supply Co, 1637-17th St

Des Moines, Iowa, E B Carr, 627 Insurance Exchange Bldg

Detroit, Mich, A F Squier, 2081 Blaine Ave

Duluth, Minn, John E Smith, 1721 W Michigan St

Erie, Pa, Coblentz Equipment Co, 1119 Peach St

Evansville, Ind, Evansville Supply Co, Esco Bldg

Fond du Lac, Wis., A N Goff, 94 Eighth St

Honolulu, T. H, The von Hamm-Young Co, Ltd

Indianapolis, Ind., Indiana Belting & Supply Co, 34 S Capitol Ave

Kansas City, Mo, Hughes Machinery Co, 342 Mfrs Exchange Bldg

Knoxville, Tenn, Leinart Engineering Co, 427 Walnut St

Los Angeles, Calif., Guy L Warden, 114 West 17th St

Louisville, Ky, Gratt-Pelle Co, 116 N Third St.

Memphis, Tenn, The Power Equipment Co, 1352 Madison Ave

Milwaukee, Wis, Hamacher & Williams, 2540 W Wells St

Minneapolis, Minn, Albert C Price Co, 257 Fourth Ave

Montreal, Quebec, Preston, Phipps Inc, 955 St James St W

New Orleans, La, Louisiana Steam Equipment Co, 109 Tchoupitoulas St

New York, N Y, Advance Engineering Co, 69 Dey St

Philadelphia, Pa, Brogan & Co, 810 Race St

Phoenix, Ariz, John W Ladlow, Box 1784

Pittsburgh, Pa, R S Eastman Co, 222 First Ave

Portland, Ore, Heating and Ventilating Equipment Co, 927 S W Oak St

Richmond, Va, A T Shepherd, 111-12 Tenth St Bldg

St Louis, Mo, O'Brien Equipment Co, 2726 Locust Blvd

Salt Lake City, Utah, Mfrs Sales & Service Co, 144 S Fifth West St

San Francisco, Calif, Refrigerating & Power Specialties Co, 380 Brannon St

Seattle, Wash, Heating & Ventilating Equipment, Inc, 500 First Ave, South

South Bend, Ind, Smith-Monroe Co, 1912 S Main St

Spokane, Wash, R A Halstead, P O Box 1359

Syracuse, N Y., The Hopton Co, 321 Denison Bldg

Tampa, Fla., G W Neale, 504 E LaFayette St

Toronto, Ont, Arthur S Leitch Co, Ltd, 1123 Bay St

Vancouver, B C, General Equipment, Ltd, 410 Homer St

Winnipeg, Man, Kipp-Kelly, Ltd, 68 Higgins Ave

Wooster, Ohio, Steam Economies Co, 1011 Beall Ave

Armstrong traps are offered in two types for heating service, "standard" traps and "blast" traps Standard traps are used for dripping headers and unit heaters where little air is to be handled Where large volumes of air must be removed quickly, the blast trap is available

The "Blast" Type Trap—The standard Armstrong trap can easily be furnished as a "blast" type trap by the use of a thermic bucket The air handling capacity of this bucket is approximately 100 times as great as with the regular air vent

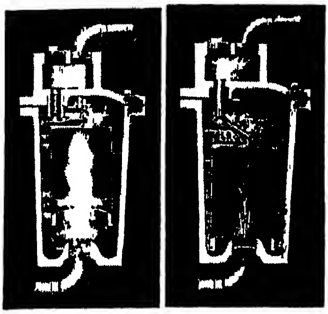
Simplicity— The Armstrong Steam Trap has only two moving parts—the valve lever assembly and the inverted bucket. Friction is practically eliminated in this mechanism All wearing parts are made from nickel chrome steel except the discharge valve and seat which are made from a special chrome steel heat treated after machining to obtain maximum hardness and toughness

Avoid Steam Trap Troubles—The intermittent action of this trap and the metal used in the valves stop scoring and wire-drawing, the common sources of leakage Air-binding is impossible because the air passes out of the bucket ahead of the steam through the vent at the top When the trap is discharging, the flow of water under the bottom of the bucket prevents the accumulation of any dirt or sediment

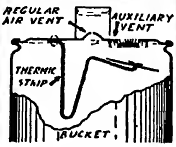
Large Capacity—Discharge orifices used in Armstrong traps are very large in

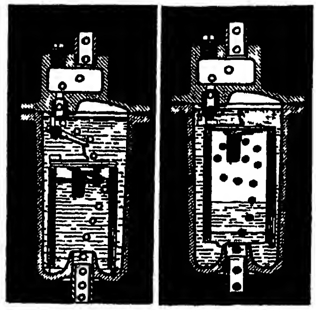
HOW THE ARMSTRONG STEAM TRAP WORKS

The Standard Trap **The "Blast" Trap**



LEFT
Standard trap (1) closed and (2) discharging
BELOW
Thermic bucket in blast trap
RIGHT
Blast trap (1) discharging air, (2) closed by steam

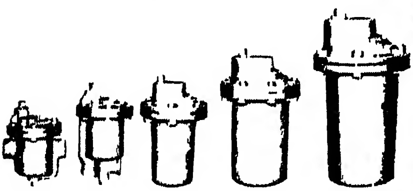


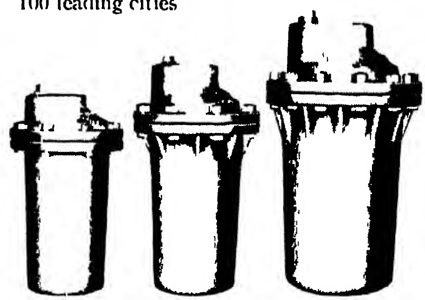


proportion to the size of the pipe connections

Armstrong trap capacity ratings are not theoretical but show actual test capacities when handling condensate at steam temperature The effect of flash steam and pipe friction to and from the trap is thus automatically taken into consideration

Service Organization The satisfactory operation of all Armstrong traps is assured by 44 district representatives in the United States, Canada and Hawaii Stocks of these traps are carried in nearly 100 leading cities





200 201 211 212 213 214 215 216

Sizes, Capacities and List Prices of Armstrong Traps

Trap Size	Lbs pressure	No 200 and 201	No 211	No 212	No 213	No 214	No 215	No 216
Pipe Connections		1/2"*	1/2"	1/2" or 3/4"	1/2" or 3/4"	1"	1" or 1 1/4"	1 1/2" or 2"
List Price (Regular)		$7 00	$9 25	$15 00	$20 75	$29 00	$38 00	$55 00
List Price (Blast Trap)		$8 50	$10 75	$17 00	$22 75	$31 50	$40 50	$60 00
Telegraph Code (Regular)		Acacia / Acanthus	Aspen	Birch	Walnut	Hemlock	Larch	Tamarack
Telegraph Code (Blast Trap)		Acacette / Acanette	Aspette	Birette	Walette	Hemlette	Larette	Tamrette
Height		4 11/16"	6 3/8"	8"	10 1/4"	12 1/2"	14"	16 3/4"
Diameter		4 1/4"	4 1/8"	5"	6 1/8"	7 1/2"	8 1/2"	10 5/16"
Weight		4 Lb	5 1/2 Lb	10 1/2 Lb	19 Lb	32 Lb	47 Lb	76 Lb
Maximum Pressure		125	200	200	250	250	250	250
Continuous discharge capacity in lbs of water per hour at pressure indicated For more complete information, see the Capacity Chart in the Armstrong Steam Trap Book	5	450	840	1560	3000	4600	7600	14,300
	10	560	1000	1900	3500	5600	9100	17,200
	15	640	1080	2060	3900	6300	10,000	19,000
	20	690	890	1800	3100	5900	8500	18,200
	25	460	940	1940	3390	6300	9200	20,000
	30	500	970	2050	3600	6600	9800	17,600
	40	550	780	1700	3450	5700	8400	20,000
	50	600	840	1840	3750	6200	8900	18,200
	60	635	900	1950	4050	6600	9300	19,600
	70	660	940	2030	3700	6100	9200	18,500
	80	690	800	1650	3920	6400	9700	19,800
	90	640	840	1750	3220	6100	10,100	18,000
	100	650	880	1840	3400	6300	10,400	18,500
	125	660	960	2040	3880	6700	10,900	20,400
	150		820	1530	3500	5900	9500	18,600
	200		900	1680	3200	5400	9500	17,400
	250				3500	5700	10,200	19,000

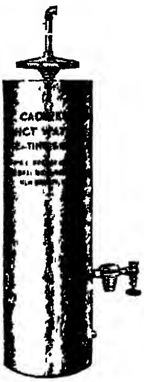
*If 3/4 in connections are desired, order No 202 for straight way or No 203 for angles

The Beaton & Cadwell Mfg. Company

Main Office and Factory. **New Britain, Conn.**

New York Office: 234 Water Street

CADWELL No. 45 SYSTEM FOR HOT WATER HEATING SYSTEMS



A complete unit—nothing else to buy

A departure from the conventional type of equipment as used with tank in basement systems

To keep the expansion pressure in the system within practical working limits even under sudden firing methods, as in oil burners

Providing means for elastic pressure distribution within the system

Keeping the system filled to any desired pressure, and above all—to positively protect the boiler

All of this is achieved in a novel manner

The combined pressure governing and relief feature—which is new

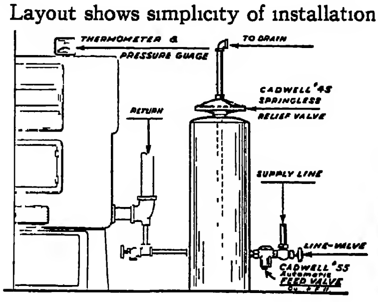
1. Has no springs
2. Is governed entirely by physical laws
3. It cannot increase the setting at any time
4. It tests itself automatically as long as there is any water in system
5. It is responsive to the slightest difference of pressure within the system
6. Absolutely guaranteed to protect the boiler

The filling arrangement is automatic and can be varied in pressure from 5 to 20 lb by adjusting screw

A large strainer prevents foreign matter from entering system The strainer, valve disc and seat can be cleaned at any time without losing water out of system

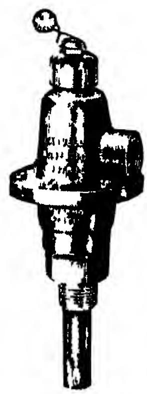
Valve and tank supplied in Maroon color

Layout shows simplicity of installation



CADWELL CONTROL UNIT AND VACUUM RELIEF VALVES

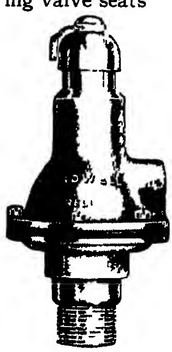
Cadwell No 5
Control Unit



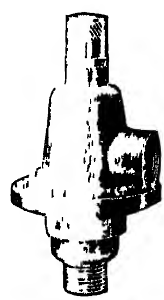
Cadwell No 25
Relief Valve

Cadwell No. 5 control unit for hot water heating system A Feed valve—Relief valve—Ball check—Screen and Back pressure valve all in one unit Feeds to 15 lb relieves at 30 lb

Cadwell No. 25 pressure temperature and vacuum relief valve for range boilers Self closing, both valves closed by internal pressure—not by heavy springs—no sticking valve seats Diaphragm operated



Cadwell No 15
Relief Valve

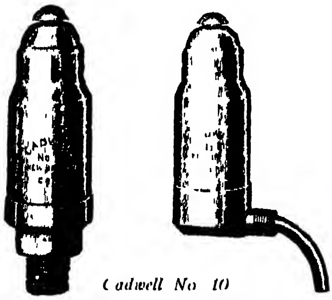


Cadwell No 35
Pressure and Vacuum Relief Valve

Cadwell No. 15—built on same principle as No 5 for relief of pressure only in hot water heating systems

Cadwell No. 35—This Valve is similar to No 25 except it does not operate by temperature Standard valve set to relieve pressure at 150 lb and slightest vacuum

CADWELL THERMOSTATIC AIR VALVES FOR ONE PIPE STEAM AND VACUUM SYSTEMS

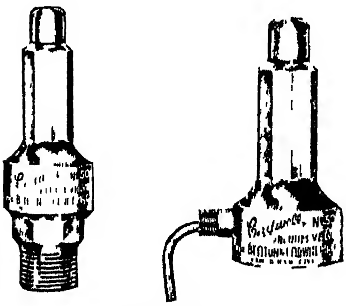


Cadwell No 10

Bottom Outlet ⅛ and ¼-in and ¼ x ⅛-in sizes *Angle Type*

Cadwell No. 10 positive action for radiators, non-vacuum, our best all metal syphon air valve

Cadwell 10 S.S. same as No 10 except straight shank for venting return lines, etc



Cadwell No 50

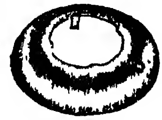
Bottom Outlet ¼-in and ¼ x ⅜-in sizes *Angle Type*

No. 50 Diaphragm operated vacuum air valves, closes efficiently against the escape of steam or vacuum Large closing surface of diaphragm, through nickel silver pin forces check off seat with every close of main valve by steam or vacuum, positively preventing sticking of valve Equalizes the system—port can be increased in size to take care of distant or sluggish radiators

Nickel silver pins are used in all Cadwell air valves to prevent corrosion

"PERFECTION" FLOOR AND CEILING PLATES

No 1



No 3A

No. 1 -Sectional floor and ceiling plate Cast iron or Brass—1 in flange—sizes ⅛ in to 12 in

No. 2 Same as No 1 with set screw instead of springs

No. 3--Hinged floor and ceiling plate Cast iron or Brass -1 in flange—sizes ⅜ in to 4 in

No. 3A—Same as No 3, with set screw--sizes ¼ in to 4 in

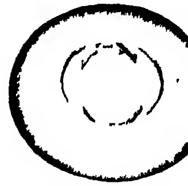
No. 3 S.—Solid floor and ceiling plate, cast iron or brass 1 in Flange with or without set screw—sizes ⅛ in to 8 in

No. 6—Sectional floor and ceiling plate cast iron or brass—1 in flange—with set screw 1¼ in high —size ½ in to 4 in.

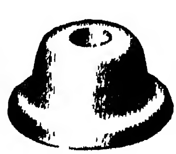
No. 6A -Solid floor plate cast iron or brass - size ¼ to 5 in -with or without set screw

No. 7 -Same as No 1 -But with 1½ in flange Size ½ in to 4 in

No. 9 Same as No 1 but with 2 in. flange size ½ in to 4 in



No 10 Hinged



No 6A

No. 10—The original No 10--Perfection -1 in flange--size ¼ in to 6 in Can be furnished in grained oak finish

No. 11—Same as No 10 but with set screw

Note:—No 10 type Copper Service Tube Plates—⅜ in to 2 in tube sizes

All plates can be furnished in plain, nickel or chromium plated finishes, as specified

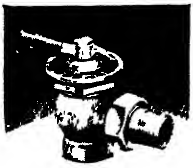
Complete Catalog Upon Request

Barnes & Jones
INCORPORATED

129 Brookside Avenue, Jamaica Plain, Boston, Mass.
New York Office· 101 Park Avenue

Barnes and Jones Vapor and Vacuum Systems of Steam Heating; Modulation Valves, Packless Quick Opening Supply Valves; Metering Orifice Supply Valves; Thermostatic Radiator Traps and Cage Units, that provide instant trap repair; Thermostatic Traps for medium and high pressures; Condensators (Boiler Return Traps); Float and Thermostatic Traps; Strainers, Damper Regulators; Gages; Proportionator Systems with Zone Control.

Modulation Valves Type K



With non-tarnishable indicating dial, non-rising stem, renewable disc seat. Tail piece extra-heavy to prevent breakage—extra-long to facilitate connection to radiator Three models lever handle, wheel handle and lock shield

Size	½ In	¾ In	1 In	1¼ In
Cap Sq Ft Rad	30	60	100	180

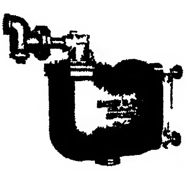
(8 ounces pressure)

Packless Quick Opening Valve Type F

Non-rising stem and renewable disc seat Large unobstructed passages to prevent trouble from dirt or scale Furnished with wheel handle only

Condensators

For returning water of condensation to boiler from open return line systems independently of boiler pressure, without change in operating conditions, air binding, or admitting steam to return side Simple and rugged in construction, positive in operation All working parts are of best bronze metal



No 32 Condensator

No	Capacity in Sq Ft
31	700
32	1,600
33	3,500
34	6,000
35	10,000
36	16,000
37	32,000

Thermostatic Radiator Traps

The Cage Unit, complete operating unit in itself, carries its own seat of special alloy Calibrated under actual working pressure at the factory and permanently locked in adjustment Unit easily and quickly replaced without special tools, lift out old unit and insert a new one Available in sizes to fit almost any make of trap

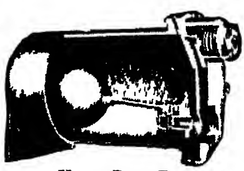
Symbol	120	12	124	134	13	14
Inlet Tapping	½"	½"	½"	¾"	¾"	1"
Outlet Tapping	½"	¾"	¾"	¾"	¾"	1"
Capacity Sq Ft C I Rad	200	200	400	400	700	1200

Capacities based on 1½ pounds pressure differential

Float and Thermostatic Traps

Drip Type

For use on Unit Heaters, also drips from supply mains and risers and on returns from water heaters and indirect stacks Float controlled valve governs discharge of water, thermostatically controlled valve allows passage of all of air but prevents passage of steam Made with ¾ in, 1 in, 1¼ in, and 1½ in tappings Capacities 200 lb to 1200 lb of water per hour at 2 lb pressure differential



Heavy Duty Type

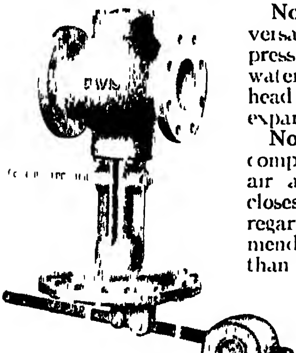
Combination float and thermostatic traps with air and water capacity large enough to take care of the load from the largest vent stacks, dry kiln coils, hot water heaters and other units condensing large quantities of steam at low pressures Made in sizes from 1 to 2 in Capacities to 14,000 lb of water per hour 2 lb pressure differential

Davis Regulator Company

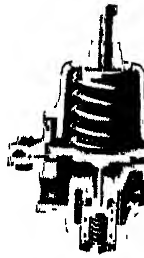
2549 South Washtenaw Avenue, **Chicago, Ill.**

Sales Representatives in Principal Cities

Manufacturers of Automatic Valve Specialties for Pressure and Flow Control



No 15



No 13

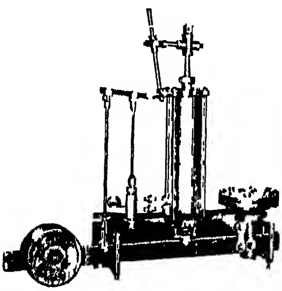
No. 15 Pressure Regulator - Universally used in vapor, vacuum and low pressure heating systems Has deep water seal, interchangeable diaphragm head and no packing box Globe and expanded outlet patterns

No. 13 Pressure Regulator A compact self-contained valve for steam, air and gas Has built-in strainer, closes tight Maintains reduction regardless of circulation Recommended for conditions requiring less than full pipe capacity Sizes ⅜ to 2 in

No 14

No. 14 Pressure Regulator - Will maintain a constant reduced pressure Balanced disc is not affected by high pressure fluctuations Has spring loading and interchangeable diaphragm head

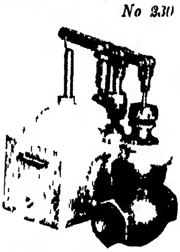
Pump Governors -Constant Pressure and Excess or Differential Pressure Governors are of similar construction to the No 11, diaphragm or piston actuated Vacuum Pump Governors are counterweighted and diaphragm actuated



No 230

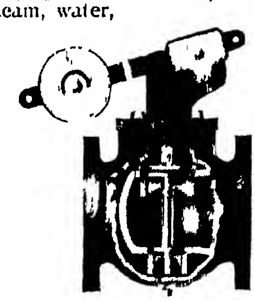
No. 230 Damper Regulator--A sensitive instrument designed to operate dampers, valves, or any mechanical or electrical device dependent on pressure or temperature changes

No. 93 Solenoid Valve -A single seated, tight closing, electrically operated flow Control Valve For steam, water, air, gas and other fluids Made to either open or close when energized A C or D C



No 93

Motor Operated Valves- Suited to zone control of heating systems or other, gradual action electrically operated flow control problems



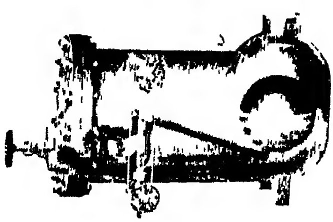
No 80

No. 80 Back Pressure Valve—A double ported, semi-balanced valve for maintaining pressure of 20 lb or less Brass seats and iron disc prevent sticking A patented construction Globe pattern may be used horizontally or vertically Angle pattern made to order

No 75

No. 70 Steam Trap—Continuous Flow Type—Duplex balanced valves give large capacity. Works on any pressure Will discharge into a vacuum line

No. 75 Strainer—"Y" type, self-cleaning. Other types for suction and pressure lines



No 70

C. A. Dunham Company

Administrative and General Offices

450 E Ohio Street, **Chicago, Ill.**

Factories Marshalltown, Iowa, Michigan City, Ind , Toronto, Canada, London, England

C. A. Dunham Co , Ltd ,
1523 Davenport Road,
Toronto, Ont , Canada

dunham differential heating

C. A. Dunham Co , Ltd ,
(of the United Kingdom)
Morden Road, London, S W 19, England

SALES OFFICES

Akron, Ohio
Albany, N Y
Allentown, Pa
Atlanta, Ga
Baltimore, Md
Bangor, Me
Birmingham, Ala
Boston Mass
Buffalo, N Y
Champaign, Ill
Chattanooga, Tenn
Chicago, Ill
Cincinnati, Ohio
Clarksburg W Va
Cleveland, Ohio
Dallas, Texas
Davenport, Iowa
Denver, Colo
Des Moines, Iowa
Detroit Mich
Duluth, Minn
El Paso, Texas
Grand Rapids, Mich
Greenville, S C
Harrisburg, Pa
Houston, Texas
Huntington, W Va
Indianapolis, Ind
Johnstown, Pa
Joliet, Ill
Kalispell, Mont
Kansas City, Mo
Kingston, Pa
Little Rock Ark
Los Angeles, Calif
Louisville, Ky
Memphis, Tenn
Milwaukee, Wis
Minneapolis Minn
Missoula, Mont
Newark N J
New Haven, Conn
New Orleans, La
New York, N Y
Oklahoma City, Okla
Omaha, Nebr

SALES OFFICES

Philadelphia, Pa
Pittsburgh, Pa
Plattsburgh N Y
Portland, Ore
Poughkeepsie, N Y
Providence, R I
Richmond, Va
Rochester, N Y
St Louis, Mo
Salt Lake City, Utah
San Antonio, Texas
San Francisco, Calif
Seattle, Wash
Springfield, Mass
Syracuse, N Y
Tampa, Fla
Toledo, Ohio
Trenton, N J
Washington, D C
Wichita, Kan

Over eighty sales offices in the United States, Canada and the United Kingdom bring Dunham Heating Service as close to you as your telephone These representatives are available for engineering counsel in correct selection of Dunham Systems and Appliances for any type of building The accumulated experience of the entire Dunham Organization is put at the disposal of the Heating and Ventilating Engineer This cooperation is available for *Modernization Work*, as well as for new construction in industrial, commercial and other projects

The Dunham Differential Vacuum Heating System is a two-pipe steam heating system which *continuously* distributes steam at a variable rate equal to the heat loss from the building It provides control of building temperature without resorting to "on" and "off" circulation of steam

It governs the heat output by variation of both the steam temperature and steam volume within the radiator *The variation in steam temperature governs the heat output from supply piping as well as radiation* Since supply piping output ranges from 10 per cent to 25 per cent of the entire output of the heating system, this added control not only adds to the precision of building temperature regulation but also to the economical operation of an installation, particularly where supply piping alone is used to heat bathrooms, kitchens and other space making intermittent heating undesirable

When the rate of heat output from a Differential System has been reduced to its practical minimum by a reduction in steam temperature and mild weather calls for a further reduction in heat supply, the system automatically reduces the volume entering the radiators establishing a condition of partial filling

The Dunham Differential System is adapted broadly for all classes of buildings in which human comfort, efficiency and health, and low operating costs are to be given consideration In its engineering design it is adaptable to the diversity of building requirements including the sub-division of larger buildings into a plurality of heating zones

The system is suitable for use with all types of radiation or combinations of various types including unit ventilators It is likewise adaptable for installations having air conditioning or ventilation services and may be advantageously used to govern the heat input to such services

Applications include existing as well as new buildings In older buildings which are unsatisfactorily heated, engineering standards are modified to give maximum improvement in heating service in relation to investment The Differential System may be economically applied to existing two-pipe steam heating installations and to one-pipe steam systems The cost of "change-over" installations varies with the type of heating system already in service

The primary function of Differential Heating is to maintain, within buildings, healthful heat-comfort and satisfactory winter air conditions not obtainable with other steam

heating systems Success in this function, however, has eliminated the waste of mild weather overheating (which is inevitable with uncontrolled installations) This results in substantial steam and fuel economies

We will gladly cooperate with architects, consulting engineers and contractors in supplying data as the basis for the surveys upon which modernizing recommendations can be made to clients

DUNHAM DIFFERENTIAL VACUUM HEATING

1 **The Dunham Differential Vacuum Heating System for High Duty**—This is a two-pipe system giving excellent room temperature control and winter air conditioning The pressure, the temperature and the volume of steam in circulation are varied under a control which is a normal function of system operation The wide range over which the vacuum and quality of steam in the radiators is regulated (from atmosphere to 25 in) establishes correct rates of heat emission with radiators either complete or partially filled as required A continuous valuation of heat requirements under positive temperature control may be secured through nine thermostats in a building, or zone of a building The High Duty Differential System is a functioning part of the Dunham Average Temperature Control System

2. **The Dunham Differential Vacuum Heating System for Low Duty**—The design of this system embraces equipment of the same general type as used in the High Duty System Its fuel economy closely approaches that system but it does not claim the same preciseness of temperature control as characterizes the High Duty System. However, the principle of control is the same as in that system, the radiator temperatures are varied and radiators may be either completely or partially filled according to the heat loss requirements The vacuum, however, is limited to fifteen inches The Low Duty System can be very effectively related to existing buildings in changing over ordinary vacuum return line systems to Differential operation One or more thermostats coupled with Dunham Heat Balancer may be used with a Low Duty System

3. **The Dunham One-Pipe Vacuum Heating System with Sub-Atmospheric Steam**—In this system the range of steam temperatures and pressures is ample to give great flexibility of operation in the lower range of vacuums This system is designed primarily to enable owners of existing one-pipe systems to obtain the benefits of the Dunham Differential Vacuum principle of operation by rearranging the system to operate on that principle The change-over can be made without extensive cutting of floors or walls Systems having air line will usually require no such cutting

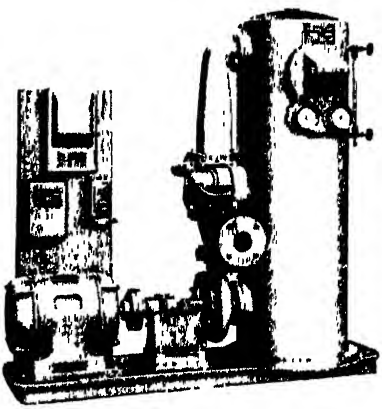
Dunham Line of Pumps for Differential Vacuum Heating Systems and for Vacuum Return Line Heating Service

The operation of these pumps is characterized by a quietness which is outstanding Their use contributes to this essential of satisfactory heating quietness They provide a vacuum-producing means of great effectiveness which is proved in its ability to perform consistently over long periods with minimum of attention and maintenance

The unit combines added refinements of construction including ball bearing centrifugal pump of improved design and discharge valves operated by a powerful mechanism

The pumps are built in eleven sizes ranging from capacity of 2,500 to 150,000 sq ft of radiation inclusive

These pumps fulfill the adopted standard for the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS and likewise the requirements of the Vacuum Return Line Heating Pump Manufacturer's Section of the *Hydraulic Institute* Pumps to meet special requirements will be supplied



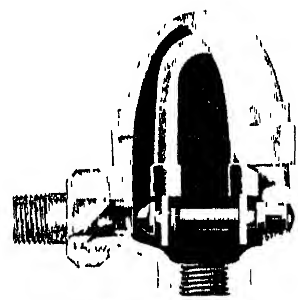
"DV" Series Pump for Dunham Differential Vacuum Heating System
"VR" Series Pump for Dunham Vacuum Return Line Heating Service

William S. Haines & Company

12th and Buttonwood Sts , **Philadelphia, Pa.**

Manufacturers of Equipment for Vapor and Vacuum Heating Systems

PRODUCTS—Haines Vento Radiator Traps, Medium Pressure and Blast Type Traps, Combined Float and Thermostatic Blast Traps, Air Eliminators, Heavy Duty Float Traps, High Pressure Traps, Boiler Return Traps and Packless Inlet Valves and Modulating Supply Valves.



Haines Thermostatic Trap

All Haines Traps, whether designed for pressures below atmosphere or pressures in excess of 100 lb per square inch, employ as their operating member a specially constructed Bourdon Tube the principle that actuates the steam gage

The tube is of tempered steel It is charged with a volatile fluid and hermetically sealed It is the expansion and contraction of the fluid, under varying temperatures, that furnishes the operating power

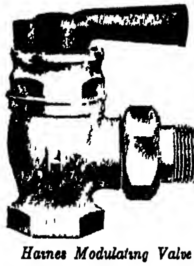
The tube is mounted vertically on a horizontal valve motion The end opposite the valve is anchored so that the travel of the tube either opens or closes the valve piece

The thermostatic member is outboard the valve seat and closes the valve against the flow of steam This arrangement prevents fouling of the trap due to scale or other foreign matter and permits a thorough draining of the unit to which it is attached

THERMOSTATIC TRAPS

Haines Thermostatic Traps are constructed to endure, as well as to operate efficiently, in sizes ranging from ½ to 1½ in They are thoroughly inspected and proved in our test laboratory under operating conditions to insure their serviceability

MODULATING VALVES



Haines Modulating Valve

Haines Modulating Valves are permanently packed, furnished with a genuine Jenkins Bros valve disc Its modulating features permit varying the amount of steam admitted to the radiator

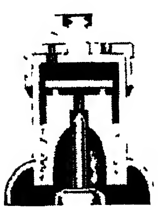
They seat tightly and open full area on less than a complete turn Furnished in Lever, Round Handles or Lock Shield type The body is made of heavy brass, nickel plated with polished trimmings

Made in sizes from ½ to 2 in in angle, straightway or corner patterns

Hoffman Specialty Co., Inc.

General Sales Department
500 Fifth Avenue, **New York, N. Y.**
Main Office and Factory: **Waterbury, Conn.**
Sales Representatives in Principal Cities

Manufacturers of Adjustable Port Radiator Venting Valves, Quick Vents and Air Eliminators for One and Two Pipe Steam and Vacuum Systems, Hoffman Supply Valves, Traps and Basement Specialties for Controlled Heat Systems and Hoffman-Economy Vacuum and Condensation Pumps.

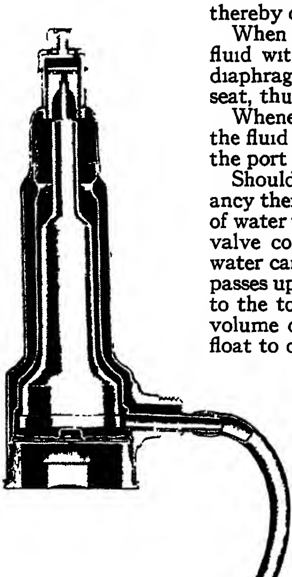


Hoffman offers a complete line of Radiator Air and Vacuum Valves and Quick Vents for every venting purpose on One or Two Pipe Steam Systems The entire line of radiator venting valves is equipped with the Six-Speed Adjustable Orifice Venting Port (illustrated at left), making it possible to balance the steam distribution in One-Pipe Steam Systems accurately, by increasing or decreasing the rate of venting of each radiator, which controls the flow of steam into that radiator

SIPHON AIR VALVES

The Nos 1, 70 and 71 are used for venting radiators on One or Two Pipe Steam Systems, and the Nos 4, 5 and 75 are used in conjunction with these valves for venting steam mains, risers and other quick venting services

OPERATION OF THE No. 1 VALVE



One of the vent ports is always open, regardless of the port adjustment The larger ports allow a more rapid rate of venting than standard and the smaller ones a slower rate of venting and thereby control rate of steam flow into the radiators

When steam reaches the float, vaporization of the heat sensitive fluid within the float expands, with "snap" action, the flexible diaphragm forming the float base and raises the valve pin to its seat, thus preventing the passing of steam.

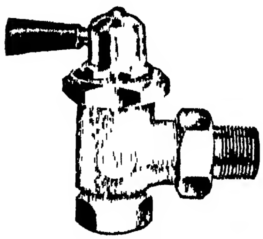
Whenever air reaches the valve, its lower temperature reduces the fluid pressure within the float The diaphragm contracts and the port is opened for the escape of the air

Should water surge into the valve, the float raises by its buoyancy thereby forcing the pin to its seat and preventing the escape of water through the vent port As soon as water drops below the valve connection, air must enter the float chamber before the water can run out Air enters through the valve connection and passes up through the air channels in the double shell construction to the top of the valve. As the air enters it displaces an equal volume of water through the siphon connection and allows the float to drop and open the vent port, allowing the air to escape

VACUUM VALVES

The Nos. 2, 77 and 78 Vacuum Air Valves operate on a similar principle as described above, but in addition feature the Hoffman Double Air Lock consisting of the vacuum check and vacuum diaphragm These valves are used on One Pipe Vacuum Systems, and for venting the ends of steam mains or heating risers, where it is also desired to prevent the return of air into the system, the Nos 6, 16 or 76 Float Vacuum vents are used

HOFFMAN CONTROLLED HEAT



No 7 Modulating Valve

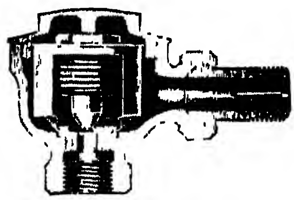
A Hoffman Controlled Heat System consists of the No 7 Adjustable Orifice Modulating Valve on the supply end of the radiator, the No 8-A Thermostatic Trap on the return end and either a Hoffman Differential Loop (for coal-fired installations operating at pressures up to 8 ounces), or a Boiler Return Trap where higher pressures are encountered, for returning the condensate to the boiler

SUPPLY VALVES

Besides the No 7 Adjustable Orifice Modulating Valve the Nos 37 and 17 series represent a complete line of Packless Supply Valves that meet the exacting requirements of architects and engineers

THERMOSTATIC TRAPS

The line of Bellows Type Thermostatic Traps, with hydraulically formed and tested bellows, consists of the Nos. 17-A, 18-A, 8-A and 9-A, and are principally used for low pressure steam or vapor systems These traps have nominal capacities from 200 sq ft up to 600 sq ft of radiation The Nos 8-A and 9-A have renewable elements, which combine the thermostat, valve pin and renewable seat into a single unit

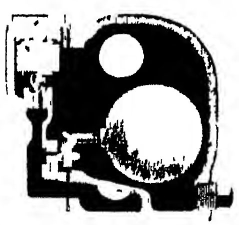


No 8-A—½ in

The No 10-A Hoffman Trap, which is equipped with waterhammer proof bellows, 1 in connection, has a nominal capacity of 2800 sq ft

The Nos 8 and 9 Traps have a thermostatic element consisting of three chambers each having a top and bottom diaphragm These chambers are all joined together and the complete thermostatic member is housed in a cage and is not attached to the valve body or cap This allows the thermostatic element and valve pin to be easily removed and replaced without adjustment These traps range in sizes from ⅜ in to 1 in, are medium pressure traps, and are recommended where pressures up to 50 lbs are encountered.

The Nos 20-A and 21-A High Pressure Traps are equipped with waterhammer proof bellows and integral strainer, for use on pressures up to 100 lbs Available in ⅜ in. to 1 in connection



DRIP AND HEAVY DUTY TRAPS

Where large amounts of condensation are encountered, it is recommended to use one of the float and thermostatic traps, which are available with or without the thermostatic element These traps are available in large capacities and are mainly used for venting and dripping risers, steam mains, unit heaters, blast coils, etc

VACUUM AND CONDENSATION PUMPS

The Hoffman-Economy line of Vacuum and Condensation Pumps offers a dependable method of economically returning the condensation from larger heating systems to the boiler These pumps are made in single and duplex units, for varying capacities and pressures

HOFFMAN SALES AND SERVICE

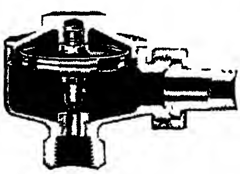
Hoffman Products are sold and stocked by leading wholesalers of heating and plumbing supplies everywhere Hoffman representatives are available to assist in selection of suitable equipment for various services

ILLINOIS ENGINEERING COMPANY

General Offices
and Factory
Chicago

Branches and Representatives in Principal Cities

Illinois Thermo Radiator Traps



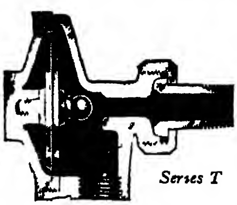
Series G

Illinois Thermo Radiator Traps for vacuum, vapor and low pressure heating systems Has cone type valve Flushes thoroughly and seats perfectly at all times Valve and seat are of Nitralloy The duplex diaphragm is of special phosphor bronze Scientific design and rugged construction assure flexibility and long life These diaphragms have withstood over three million strokes on a breakdown test

Made in three sizes ½ in , ¾ in and 1 in and in a variety of patterns

Special thermostatic traps can be furnished for working pressures up to 125 lb

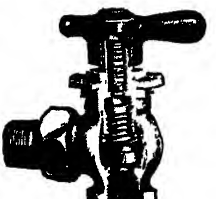
Illinois Thermo Radiator Trap



Series T

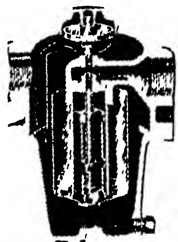
The Original Vertical Seat Trap Self cleaning, non-adjustable Positive and sensitive in operation Thousands in use for over fifteen years without diaphragm replacements Furnished in all sizes and patterns

Illinois Modulating Supply Valve



Quick-opening, packless Steam tight on 50 lb pressure Large diameter of thread spool and machine cut threads make valve operation easy. Furnished in a complete line of sizes and patterns

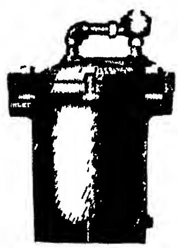
Illinois Combination F & T Traps



Series 7G

Unsurpassed for draining ventilating units, unit heaters, and for dripping mains and risers—wherever it is desirable quickly to vent air from the main as well as handle the water of condensation in quantity, whether hot or cold

Illinois Combination Trap

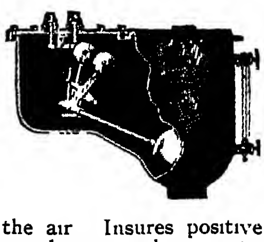


Series 36

This is the heavy duty type similar in operation to the 6G, 7G, and 8G traps

Series 36 traps are available in a complete range of sizes They are equipped with separate thermostatic by-pass and are furnished where specifications require this type, or where capacities are beyond the range of the 6G, 7G or 8G

Illinois Return Trap



Automatically returns the condensation to the boiler, regardless of pressure on the boiler up to 8 lb , at the same time discharging the air Insures positive and complete circulation, and prevents cracked boiler sections

Trap is self-contained, with no external working parts to be misadjusted, tampered with or injured. No stuffing boxes or packed joints, which insures continuous tightness against air or water leakage

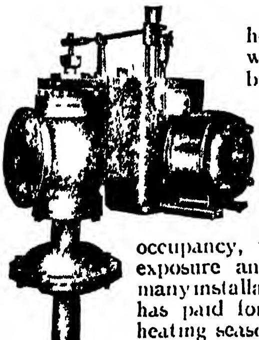
Write for Bulletins

ILLINOIS ENGINEERING COMPANY

General Offices and Factory
Chicago

Branches and Representatives in Principal Cities

Illinois Thermal-Zone Control

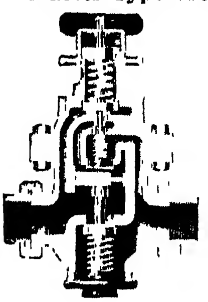


Prevents overheating and fuel waste in large buildings or groups of buildings heated from one central power plant Buildings may be zoned as to occupancy, time, location, exposure and so on In many installations this valve has paid for itself in one heating season

Illinois Reducing Valve

In general use on vacuum or low pressure heating systems Will reduce to 4 oz pressure from an initial pressure of 150 lb The large diaphragm insures sensitive operation Made in both straightway and expanded outlet bodies in sizes from ¾ in to 12 in

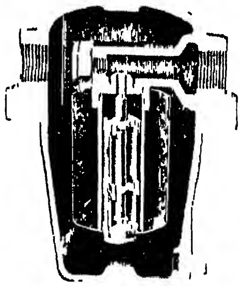
Master Type Pressure Regulator



For exacting requirements, such as tire and rubber vulcanizing, chemical process pressure control, and wherever high pressure steam must be accurately reduced in varying amount to any steady lower pressure not less than 10 lb

It will reduce initial pressure up to 300 lb down to any lower pressure not less than 10 lb, and does not build up pressure on a closed or dead end line Made of bronze with monel metal valves and seats

Illinois Steam Trap



Series 80

Valve and stem are separate from the bucket and operated only by the bucket at the extreme top and bottom of travel - result - *valve is always either full open or tight closed* No wire drawing or cutting of valve and seat which are of Monel metal

Eclipse Spring Controlled Regulating Valve

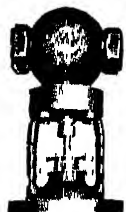


Fig 121

Furnished in either single seated or double seated type as the service conditions require, for the control of steam, air or gas Controlling spring is completely enclosed, protecting it from dirt and rust Valves are furnished with the proper size diaphragm and the proper length spring to give satisfactory service under all operating conditions

Steam and Oil Separators

Vertical Standard Separators

Eclipse steam separators are made in both horizontal and vertical type, and also the special receiver separators for standard or extra heavy pressures

Eclipse oil separators are furnished in the horizontal type and have a removable baffle plate to facilitate cleaning of baffle and keeping the separator's efficiency at the highest point

Write for Bulletins

Kieley & Mueller, Inc.

Established 1879

Engineering Specialties for Pressure and Flow Control

34 West 13th Street, **New York, N. Y.**

Factory NEWARK, N J

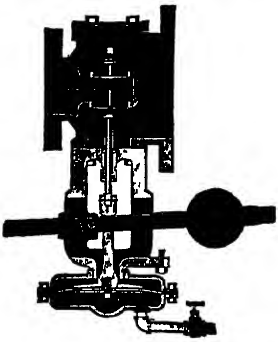
Agents in All Principal Cities

PRODUCTS—Valves: Altitude, Stop and Check, Pressure Regulating, Float, Pilot Reducing, Back Pressure, Tank Control.

Liquid Level Controllers, Water Feeders, Pump Governors, Steam Traps, Y-Type Strainers.

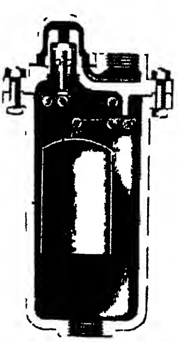
Also Damper Regulators, Hot Water Temperature Controllers, Oil Separators, Steam Separators, Return Traps, Water Columns, etc

Catalogs sent upon request



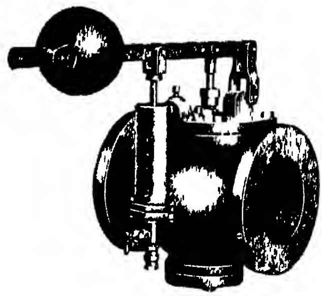
Pressure Regulating Valve

Spring and lever weighted valves for all services and for initial pressures up to 250 lb. and reduced pressures from 0 to three-quarters of the initial pressure Single or double seated in sizes ⅜ to 16 in Suitable for steam, water, air, oil and gas Controlled by a small feeler pipe connected from diaphragm to low pressure side



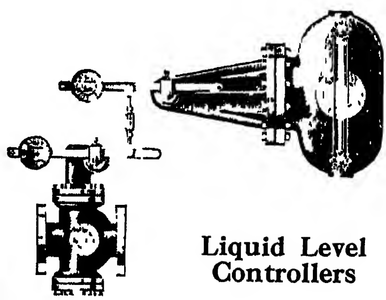
Steam Traps

Large capacity, small sized inverted bucket traps, quick-acting, self-cleaning and non-air binding Sizes ⅜ to 2 in Pressures up to 250 lb. Body and cover, semi - steel Valve and seat, stainless steel Removable cap allows inside inspection or replacement of valve parts without disturbing pipe connections (All parts are interchangeable).



Back Pressure and Atmospheric Relief Valve

For use where plant is operated either condensing or non-condensing Outside air dash pot insures noiseless operation Maintains exhaust line back pressure from 0 lb. to 25 lb Made horizontal or vertical lever and weight or spring operated



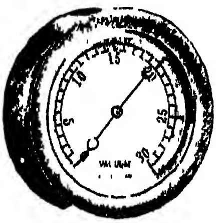
Liquid Level Controllers

For the accurate control of liquids in tanks or other vessels, suitable for use in industrial plants, gasoline plants, refineries, etc Direct connected or remote control, ball bearing spindle and easy-to-pack stuffing box rotary or sliding valve Write for special bulletin C-3

J. E. Lonergan Co.

207 Florist Street, **Philadelphia, Pa.**

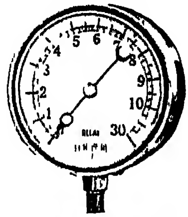
Pop Safety Valves; Relief Valves; Steam Gauges; Hydraulic Gauges; Air Gauges; Water Gauges; Pressure and Temperature Gauges; Test Gauges; Gauge Boards; Oil Gauges; Clocks; Counters; Gauge Cocks; Steam Gauge Syphons; Lubricating Specialties.



Model "GV"

Model GV—Vacuum Gauge Gauges graduated to 30 in vacuum Ten sizes, 2½ to 10 in dial

Model BLGR—Gauge for indicating height of water in feet Graduation 70 ft Three sizes, 3½, 4½ and 5 in dial



Model "BLGR"

Model "GLP"

Low pressure, Iron Body, Brass Mounted

Set to blow off at 10, 15, 20, 25 or 30 lb

Five sizes–2½ to 4½ in

Special valve for vacuum breaking

Six sizes, ½ to 2 in

Positive in its action

Model "VAK"

Model "U"

Relief Valve Snifter, Water or Cylinder—Bronze

Recommended for steam engines, pumps, pipe lines, etc

Ten sizes, ½ to 4 in

Oil Relief Valve

Sizes ⅜ to 2 in

For use on oil burning systems Has large relieving capacity

Model "ORV"

Model "WRV"

Model WRV Water Relief Valve for tank service

Sizes ⅜, ½ and ¾ in

Model HHD Pop Safety Valve A S M E "House Heating Boiler Standard pressures, 5, 10 and 15 lb

CATALOGUE

Write for our new 100-page catalogue, describing and illustrating the complete "Lonergan Line" or ask us about specialties in which you are interested

Model "HHD"

Milwaukee Valve Company

Milwaukee, Wisconsin

Manufacturers of a complete line of Heating Specialties for vapor, vacuum and gravity heating systems. Each item is scientifically designed, accurately machined and thoroughly tested before final approval. MILVACO engineers, located in principal cities, from coast to coast, render an intelligent, courteous service to architects, engineers and heating contractors in the design and installation of modern heating systems.

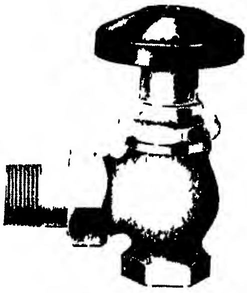


Fig No 215 Radiator Packless Angle Valve Highly polished nickel plated bonnet, tail piece and nut Body nickel plated rough finish Available in chrome finish and lock and shield pattern Sizes—½ in to 2 in inclusive

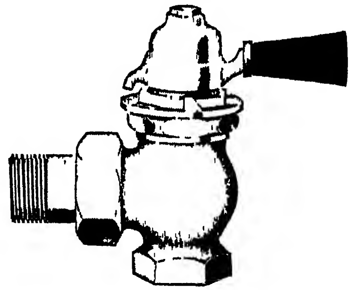


Fig No 209 Radiator Packless Angle Valve, Lever Handle, Graduated Finish same as No 215 Milvaco Radiator Valves cannot bind or stick in operation Sizes—½ in to 2 in inclusive

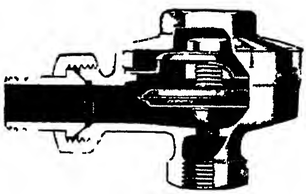


Fig No 350-6 Thermostatic Trap A single, full-floating thermal element closes in the steam Distinctive design insures positive seating accuracy and dependable, sensitive operation. Nickel finish rough body with polished cover, nut and tail piece. Available also in chrome finish Made in sizes ½ in to 1 in inclusive and capacities 200 to 1000 sq ft E D R

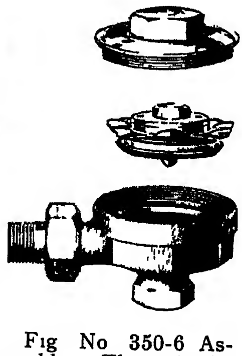


Fig No 350-6 Assembly Thermal element is secured in cage Trap cover may be removed and diaphragm assembly lifted out and replaced without disturbing the adjustment

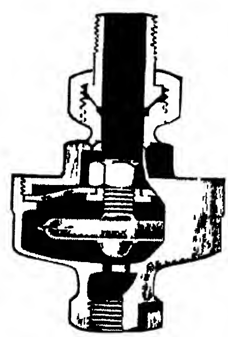


Fig No 100ST Straight Through Thermostatic Trap Recommended and especially adapted for use in connection with convector and concealed radiation

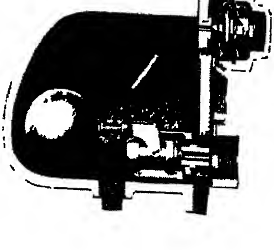


Fig No 13-2 Float and Thermostatic Trap Made in 14 sizes and capacities ranging from 600 lb to 8,000 lb of water an hour at a 2 lb pressure differential Recommended for use on industrial heaters, unit heaters, vento, blast and dry kiln coils, evaporators, etc and other units condensing large quantities of steam at pressures not to exceed 15 lb Sizes ½ in to 2½ in

Fig No 12 Air Eliminator For rapid elimination of air from mains, coils or any low pressure system Air is rapidly vented past a check valve and through large ports Float rises to prevent passage of water and thermal element closes against steam Sensitive, positive in operation Pipe connections ½ in, ⅜ in, ½ in and ¾ in

Fig No 32 Quick Vent For quick venting of mains, blast coils or any high point where a large volume of air must be rapidly vented and water is not a factor ½ in and ¾ in sizes

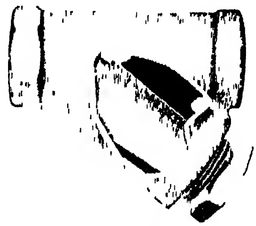


Fig No 17 Sediment Strainer Prevents passage of dirt, scale or solids Easily cleaned Sizes ½ in to 3 in inclusive 2 in, 2½ in and 3 in sizes have flanged strainer cap

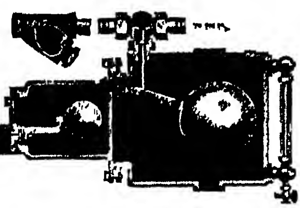


Fig No M-80

Fig. No. M-80 Duplex Water Feeder

A positive means of providing constant water level in boilers, receiving tanks, steam generators or other apparatus requiring control of water level Replaces natural losses in the system and, in addition, should the return of the condensate be obstructed, or, if the water is syphoned out of the boiler, this feeder not only supplies water to maintain a constant level but also discharges excess amounts which may accumulate and be suddenly relieved

All mechanism is completely enclosed and the operation is entirely automatic and thoroughly dependable Capacity ratings from 1,500 lb to 5,500 lb of water per hour

Fig. No. 85 Boiler Return Trap

When the heating system is working under a very low pressure or under atmospheric pressure only, this return trap acts as a receiver and air vent When the boiler pressure increases, the condensation rising in the trap causes the float to rise The float rising automatically opens the steam valve admitting boiler pressure to the top of the trap, and at the same time closes the vent valve The pressure in the boiler and trap is thus equalized and the water of condensation flows by gravity back to the boiler without collecting in the return mains or risers, regardless of boiler pressure When the condensation is discharged the float drops to the bottom of the trap, opening the vent valve and closing the steam valve. This action is repeated as long as there is a differential in pressure between boiler and return line piping

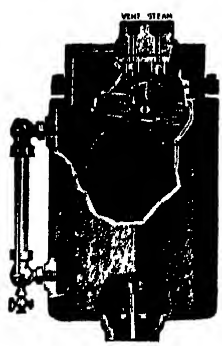


Fig No 85

Operates on any pressure not exceeding 15 lb Capacities 1500 sq ft and 1000 sq ft of radiation Pipe connections 1½ and 2 in

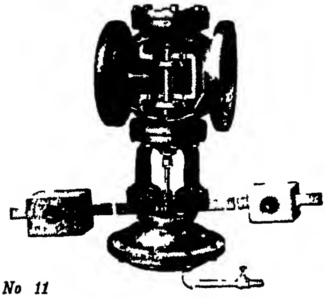
For complete detailed information, roughing-in dimensions, weights, etc. consult current condensed catalog of MILVACO Heating Specialties.

Mueller Steam Specialty Co., Inc.

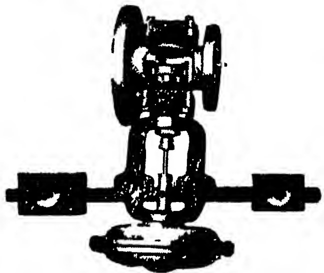
349-351 West 26th Street, New York City

Steam, Water, Air, Oil and Gas Specialties for Heating and Power Plants

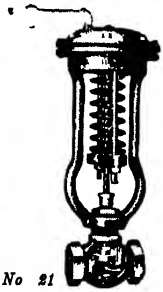
Pressure Reducing Valves—Straight Pattern and With Increased Outlet



No 11



No 17



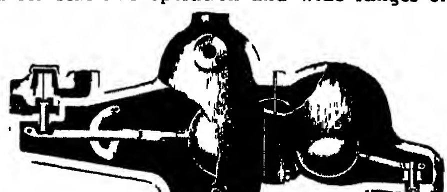
No 21

No 11—For Vacuum, Vapor and Low Pressure Heating Systems Initial Pressures, up to 200 lb , Reduced Pressures, 0 to 10 lb.

No. 17 and 21—For automatic control of reduced pressures on dead-end service, requiring a tight closing valve, such as tank heaters, kitchen utensils, sterilizing apparatus, laundry equipment, kettles, cookers, driers, etc Initial Pressures up to 200 lb Reduced Pressures 0 to 150 lb.

Constructed with full globe bodies Center guide eliminates the wings on discs, and increases efficiency, assures minimum noise and prolongs the life of the seats and discs Lever and weight operates on a steel roller bolt, assuring a most sensitive valve Spring type furnished with special long springs for sensitive operation and wide ranges of reduced pressures

Automatic Water Feeders



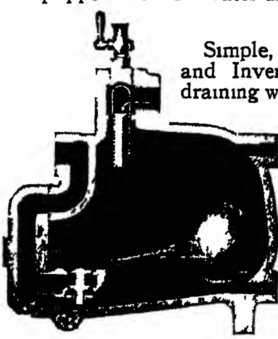
No 517—Duplex Up to 22,500 Sq Ft

No 511—Simplex Up to 22,500 Sq Ft

With a powerful leverage to control the water line in steam boilers, etc. They supply make-up water to compensate for evaporation, leaks, steam utilized in process work and condensation wasted Where condensation held up in the system eventually returns in large quantities, our Duplex type protects the boiler against flooding All working parts of non-corrosive metal, are accessible without breaking pipe connections Provided with an integral strainer. For steam pressures up to 100 lb , water pressures up to 120 lb

Equipped with low water and pressure Mercoid Tube Switches for all services

Steam Traps



Ball Float
No 219—Up to 30 lb
No 221—Up to 150 lb
Sizes ½ to 3 in

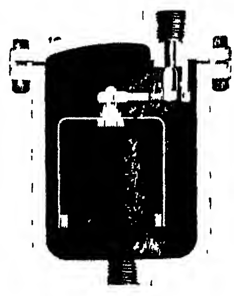
Simple, Sturdy and Compact Ball Float and Inverted Bucket Steam Traps for draining water of condensation from steam apparatus and steam mains.

Powerful leverage enables them to take care of large quantities of condensation

Ball Float Steam Traps equipped with integral strainer, water gages, air cocks, blow-off and integral by-pass valve, when desired

All working parts are accessible without disturbing any pipes

Valves are sealed with several inches of water, making the escape of steam impossible.



Inverted Bucket
No 211—For Pressures Up to 250 lb
Sizes ½ to 2 in

Catalogue and Bulletins covering our Complete Line gladly furnished on application

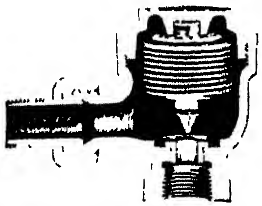
Sterling Engineering Company

MANUFACTURERS OF STEAM AND WATER HEATING EQUIPMENT

3710 North Holton Street, **Milwaukee, Wis.**

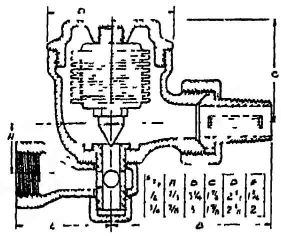
Offices in Principal Cities Throughout the United States

See Telephone Directory for Local Addresses



Sterling Bellows Trap No. 7

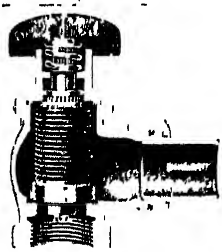
Low pressure thermostatic bellows trap Discharges all air and water from radiators and closes tight against steam Bellows are of hydron type, with self-centering valve of stainless steel and stainless steel seat Built in ½ and ¾ in sizes



No 7 Trap with Adjustable Outlet

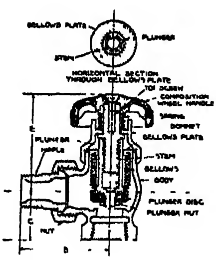
Sizes, Capacities and Roughing-in Dimensions of Sterling No. 7 Thermostatic Trap

Sym	Capacity	A	B	C	D
7	200 sq ft	½"	½"	3¼"	1⅜"
7	200 sq ft	½"	¾"	3¼"	1⅜"
7	400 sq ft	½"	¾"	3¼"	1⅜"
7	400 sq ft	¾"	¾"	3"	1⅜"



Sterling No. 22 Bellows Packless Radiator Valve

The No 22 Valve employs a hydron bellows with patented anti-torsion feature that makes valve everlasting Valve is of disk type Complete details in Bulletin 231-A



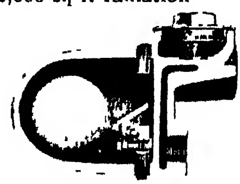
Dimensions, Inches

Valve Size, In	B	C	E	Cap., Sq Ft Radiation
½	2⅝	1 9/16	2¾	50
¾	2⅝	1 5/16	2⅞	100
1	3	1½	3¼	200
1¼	3½	1¾	3 5/16	275
1½	4¼	1⅞	3⅜	350

Sterling Boiler Return Trap

Alternately accumulates condensation from Sterling Vapor Heating Systems and returns it, against pressure, to boiler Automatic in operation, positive in action Contains no springs or stuffing boxes Will function against boiler pressures up to 20 lb Four types, for 2500, 4000, 8000 and 16,000 sq ft radiation

Sterling Float and Thermostatic Trap

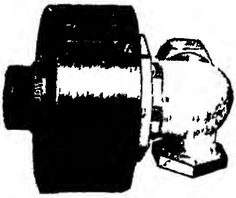


Low pressure traps for unit heaters, steam kettles, blast coils and main drips, seven sizes No 71 and 74 are small and compact units for handling condensate of unit heaters Nos 75, 78, 79, 80 and 81 are for general low pressure steam trap service Suitable for pressures up to 20 lb Complete details in Bulletin No 234-A

Sterling Thermotrols

Thermostatic self-contained temperature controls

Type E—No. 100 Thermotrol

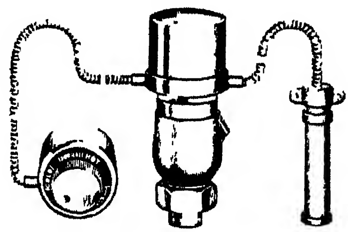


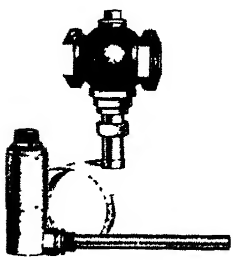
Designed for attachment to exposed radiator for the control of room temperature It is applicable to two-pipe steam, vapor, vacuum or hot water heating systems in any type of building The unit consists of a bronze valve upon which is mounted the thermostatic control The room air, circulating through the openings in the housing, causes expansion or contraction of the compound diaphragms controlling the valve The dial indicates normal, hot, cold or closed, as required

Made in ½, ¾, 1, 1¼ and 1½ in inlet and outlet sizes, for 50, 100, 150, 275 and 300 sq ft radiation *Complete details in Bulletin No 361*

Type G—Thermotrols

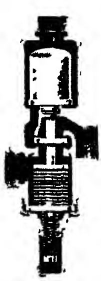
Designed for installations where the radiators are concealed The controls consist of a valve, a regulating cylinder, and a thermostatic bulb The regulating cylinder transmits the pressure, caused by the expansion of a volatile liquid in the bulb, to the valve stem This pressure causes the valve to close A spring opens it when the pressure decreases A dial on the regulating cylinder determines the room temperature at which the valve operates Sizes, ½ to 1½ in *Complete details in Bulletin No 361*





No. 110 Thermotrol

Designed to regulate the temperature of liquids in tanks It is applicable to water tanks, cooking kettles and sterilizers It depends upon the expansion of a volatile liquid, like all other Sterling Thermotrols, for its regulating properties and is provided with a similar adjusting dial Sizes, ½ to 3 in *Complete details in Bulletin No 361*

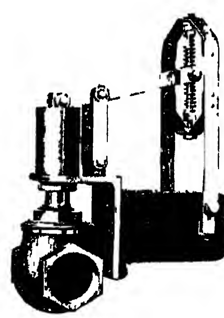


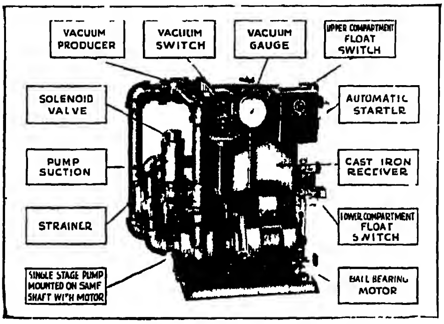
Sterling Motorized Valve

Controls the flow of steam, water or oil It is used with thermostats for the control of room temperature by controlling the flow of steam or water in mains or risers Valve sizes, ¾ to 6 in *Complete details in Bulletin No 361*

Sterling Mixing Valve

The Sterling No 130 Series Valve is designed to regulate the temperature of hot water It controls the flow of both hot and cold water through the valve *Complete details in Bulletin No 361*

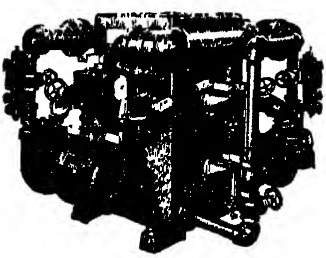




Sterling Simplex Vacuum Pump

Consists of a motor-driven pump, mounted on a cast iron receiver constructed with two compartments The centrifugal pumping unit is a single suction, single stage pump, and a squirrel cage motor mounted on one shaft, thus eliminating mis-alignment troubles Wearing rings protect the casing and cover from wear The discharge to the boiler is controlled by a solenoid valve, which in turn is controlled by the float switch mounted at the end of the upper receiver compartment

A float in the upper receiver operates the float switch The lower, or accumulator compartment, is similarly equipped with a float and float switch A strainer is located on the return line which enters the lower compartment of the receiver at the opposite end from the float switch A vacuum switch, automatic starter and vacuum gage are mounted on the side of the receiver The pump suction is connected into upper compartment of receiver Discharge from the pump is connected to the vacuum producer A solenoid valve is located in the discharge line to boiler A line from the lower portion of the accumulator compartment connects into suction side of vacuum producer



Sterling Duplex Vacuum Pump

Consists of two motor-driven pumps mounted on one cast iron receiver Each pump is equipped with a separate suction and discharge line and automatic starter so that either pump can be operated independently

One cast iron receiver, with one strainer, serves both pumps, but each pump has its own vacuum producer One vacuum switch controls the motors for both pumps *Complete details in Bulletin No 334.*

Sterling Dirt Strainers

FIGURES No. 5 AND No. 6

Designed for use at drip points, on blast coils or ahead of radiator traps Baskets are of heavy perforated sheet brass Housings are heavy grey iron castings designed for steam pressures up to 125 lb

Figure No. 5—Made in sizes from ½ to 2 in with screwed connections

Figure No. 6—Made in sizes from 2½ to 6 in with flanged connections

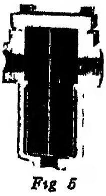


Fig 5

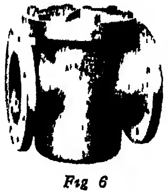


Fig 6

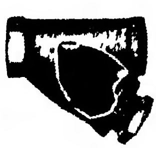


Fig 7

FIGURE No. 7

Figure No. 7 is of the Y-type for use on water lines to pumps, swimming pools, etc , or on steam lines to traps, etc Made in sizes from ½ to 2 in Designed for steam pressures up to 125 lb

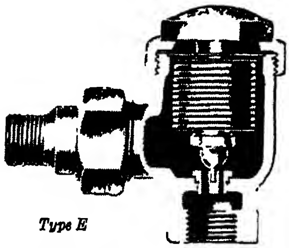
Sarco Company, Inc.

183 Madison Ave., **New York, N. Y.**

Branches in Principal Cities

SARCO CANADA LIMITED, FEDERAL BLDG, TORONTO, ONT

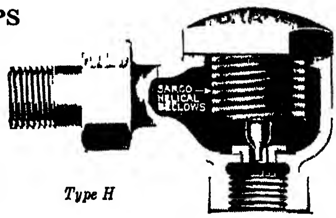
PRODUCTS—A complete line of Specialties for Vapor, Vacuum and Gravity Steam Heating Systems, combined with a competent Engineering Service to architects and heating engineers to assist them in providing modern heating



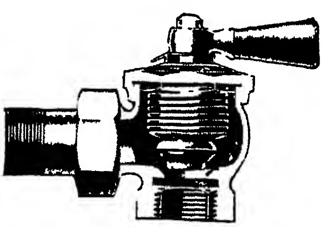
Type E

SARCO RADIATOR TRAPS

These efficient traps are equipped with bellows made from our own heavy wall, seamless, corrugated bronze tubing They are exceptionally rugged, have long life and high capacity Available in angle, straightway and corner patterns The body is heavy brass, nickel plated, with polished trimmings



Type H



Sarco Bellows-Packless Valve

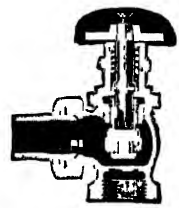
SARCO RADIATOR VALVES

For high vacuum heating systems, Sarco offers Sarco Bellows-Packless Valve It is of the truly packless, modulating type Leakage at the stem is impossible The stem is sealed to the cap by Sarco seamless, corrugated bronze tubing

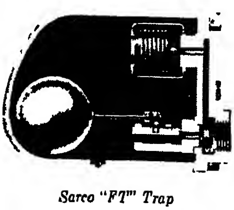
The Sarco-Marsh valve is of the commercial packless type in which the stem is sealed by a metal-to-metal cone seat, re-enforced by moulded packing held compressed by a spring

Valve bodies of both types are cast brass, heavily nickel plated with polished trimmings

Can be furnished with round or lever handles, or lock shield, in angle, straightway or corner patterns



Sarco-Marsh Packless Valve



Sarco "FT" Trap

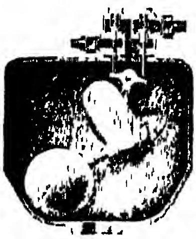
SARCO FLOAT—THERMOSTATIC TRAPS

Used for dripping the ends of mains and risers Also for stack or blast heaters, large unit heaters and hot water generators Have automatic thermostatic air vents built in. Available in seven sizes with connections ¾ to 2 in

SARCO AIR ELIMINATORS

For venting air from vapor systems at one central point in the basement Available in two sizes No 6 for systems up to 2,500 sq ft and No 12 for 15,000 sq ft Both are equipped with float valves to stop water escaping through the vent, and with check valves to prevent ingress of air when system is under vacuum

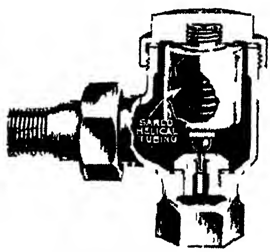
Air Eliminator No 6



Alternating Receiver

SARCO ALTERNATING RECEIVER

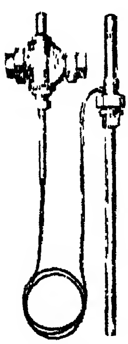
A complete line of boiler return traps for vapor systems Return water of condensation to boiler automatically, thereby assuring positive return of water under all pressure conditions Made in six sizes for from 1,500 to 25,000 sq ft of radiation



Type S-75 Steam Trap

SARCO MEDIUM PRESSURE TRAPS

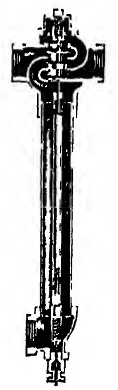
These thermostatic steam traps are designed specially for kitchen fixtures and hospital equipment. Available in sizes $\frac{3}{8}$ to 1 in in two types, S-65 for pressure 15 to 65 lbs and S-75, 0 to 100 lb



Type TR-21 Temperature Regulator

SARCO TEMPERATURE REGULATOR

A self-contained automatic valve for controlling temperatures from 0 to 300 F For hot water heaters, tanks, sterilizers and room control Available in sizes $\frac{1}{2}$ to 6 in



Water Blender

SARCO WATER BLENDER

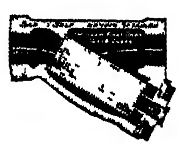
A self-contained automatic three-way valve for mixing hot and cold water Delivers the mixture automatically at any desired temperature between 30 and 250 F Both valves are fully balanced No reducing valves required For showers, wash rooms and to deliver tempered water to different departments

Also Damper Regulators and Compound Gauges specially designed for vapor and vacuum systems

SARCO STRAINERS

Pipe line strainers are effective insurance against injury to traps, valves, meters, pumps, etc

Sarco strainers are made with screens for steam, water, oil, gas, brine or ammonia



Strainer

CATALOG

For full information on all Heating Specialties write for Catalog HV-45.

WARREN WEBSTER & COMPANY

Pioneers of the Vacuum System of Steam Heating

-since 1888 Webster Systems of Steam Heating

Main Office and Factory.
Camden, N. J.
Branches in Over 60 Cities
Consult Your Local Phone Directory

WEBSTER MODERATOR CONTROL "Control by the weather"

PRODUCTS AND SERVICES

Improved Webster Systems of Steam Heating including Vacuum and Type "R" (vapor).

Webster Central Control Systems including HYLO and MODERATOR.

Modernization of Obsolete and Faulty Heating Systems.

Webster System Equipment including Light-Weight Concealed Radiation (Gravity Convection Heaters), Radiator Supply Valves, Metering Orifices, Thermostatic Traps, Drip Traps, Heavy Duty Traps, Dirt Strainers, Dirt Pockets, Boiler Return Traps, Vent Traps, Damper Regulators, Boiler Protectors, Lift Fittings, Expansion Joints, Separating Tanks, Steam and Oil Separators, Steam Vacuum Pump Governors, Air Separating and Receiving Tanks, Gages, Water Accumulators.

Webster Series "78" and Series "79" Traps for use at process pressures (10 to 125 lb. per sq. in)

IMPROVED WEBSTER SYSTEMS

The Improved Webster Systems are low pressure, two-pipe systems of steam circulation *with the addition* of accurately-sized metering orifices at radiator supply connections and, when required, intermediate metering orifices at points in branch mains Metering orifices effect even distribution of steam to all parts of the heating system and permit the successful application of a centralized control Improved Webster Systems are available for vacuum, open return or "vapor" operation The Type "R" System corresponds to the so-called Vapor type Fig 1 illustrates a typical arrangement of Boiler Return Trap, Vent Trap, etc , when low pressure boiler is the source of steam.

CENTRAL CONTROL SYSTEMS

These are patented systems for varying the amount of steam to all radiators according to outside temperature They provide *continuous heat delivery* with effective *fractional filling of radiators* The Hylo Systems may be provided for manual control, or if desired, may be semi-automatic by incorporation of inside thermostat or thermostat and schedule clock The Moderator Systems employ an automatic Outdoor Thermostat supplemented by a manual Variator

The latter is used for quick heating-up, night load, and unusual weather or occupancy conditions Use of Webster Central Control Systems results in (1) increased comfort because over-heating and underheating are minimized and (2) lower fuel or steam costs

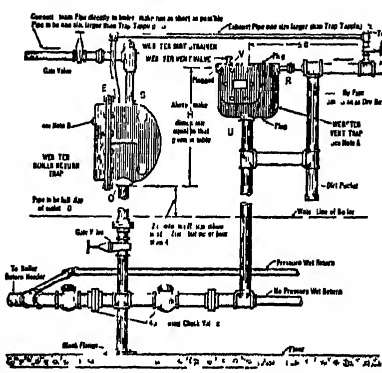


Fig 1 Conventional arrangement of piping around Webster Basement Equipment for the Webster Type "R" System

WEBSTER SYSTEM RADIATION

Concealed, non-ferrous type for use exclusively with Improved Webster Systems Is unique in that it combines *in a single unit,* a light-weight heating element of high efficiency with an orificed radiator supply valve, a radiator trap and supply

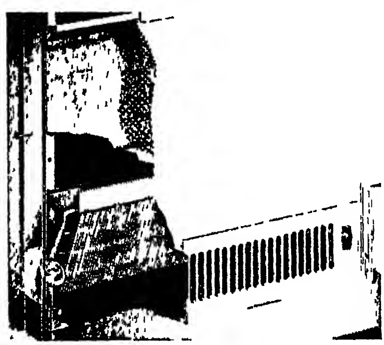


Fig 2 Webster System Radiation

and return piping connections Metal enclosures for installation within the wall and exposed metal cabinets are available Webster System Radiation and enclosures are so designed that the entire heating element can be quickly removed without damage to plaster or paint Space requirements reduced to a minimum and installation greatly simplified

RADIATOR SUPPLY VALVES

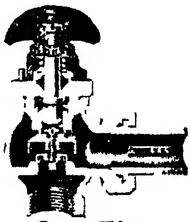


Fig 3 Webster Type "W" Valve

Type "W"—Standard supply valve of highest quality Slotted sleeve gives "modulation" of steam flow *provided proper pressure is maintained on system* Quick-opening, non-rising stem type Uses molded ring packing Valve disc is renewable Made in angle model in sizes of ½, ¾, 1 and 1¼ in Right-corner, left-corner, straightway single union and double union models in selected sizes Choice of wheel, lever, lockshield, chain wheel or extended stem handles

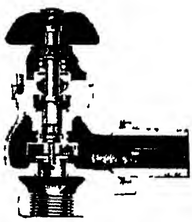


Fig 4 Webster Three-Point Valve

"Three-Point" Has sleeve orifice fitting into seat opening Especially desirable for hospitals, hotels, etc, where quick heating-up or higher temperatures are required in different rooms Is normally an *open-shut* type but may be easily converted to *shut-normal-excess* type *Excess* setting gives 140 per cent of normal steam flow Otherwise similar in design to Type "W" Valve. Sizes: ¾ and 1 in.

Sylphon Packless—Positively packless having flexible laminated Sylphon bellows completely enclosing stem Quick-opening, non-rising stem type Renewable composition disc Made in angle model in sizes of ½, ¾, 1, 1¼, 1½ and 2 in Right corner, left-corner and straightway single-union models in sizes of ½, ¾ and 1 in Choice of lever, wheel, lockshield, chain wheel or extended stem handles

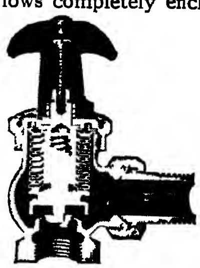


Fig 5 Webster Packless Valve

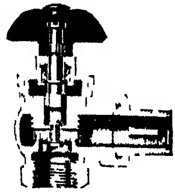


Fig 6 Webster Type "B" Valve

Type "B"—Quick-opening, non-rising stem, packless type Special gland takes up wear of molded ring packing Renewable composition disc Made in angle model in sizes of ½, ¾, 1, 1¼, 1½ and 2 in Right-corner, left-corner and straightway single-union and double-union models in sizes of ½, ¾ and 1 in Choice of wheel, lockshield, chain wheel and extended stem handles

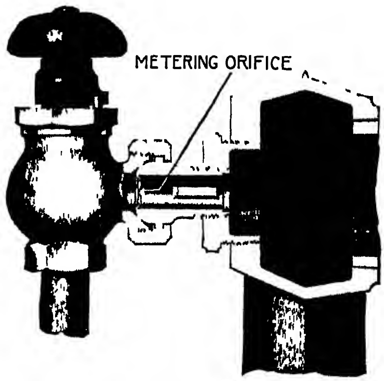


Fig 7 Metering Orifice Inserted in Union Connection of a Webster Supply Valve

Metering Orifices—Accurately sized and made of heavy gage Monel Metal to resist erosion and corrosion, amply thick to be free from vibration and shaped for quiet operation Available in a number of types to fit both new and installed radiator supply valves of Webster or other make

RETURN TRAPS

Sylphon—Original and highly perfected type of low pressure, thermostatic bellows trap Rugged in construction Renewable seat Factory adjusted Made in angle, right-corner, left-corner and straightway bodies Sizes ½, ¾ and 1 in Normal operating pressures up to 15 lb. per sq in Maximum occasional pressure 25 lb per sq in

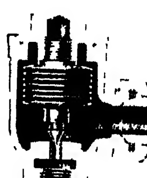


Fig 8 Webster 512 Sylphon Trap

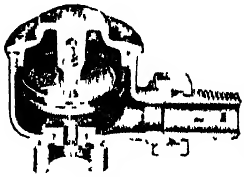


Fig 9 Webster Size 702 Trap

Series "7"—Perfected diaphragm-type thermostatic trap Unusually strong in construction Renewable seat Factory-adjusted, phosphor-bronze diaphragm Made in angle, right-corner, left-corner and straightway bodies Sizes ½, ¾ and 1 in Normal operating pressures up to 15 lb per sq in, maximum occasional pressure 25 lb per sq in

Series "7-M"- Similar in design to Series 7 but built for normal operating pressures up to 25 lb per sq in Maximum occasional pressure is 50 lb per sq in Uses Monel Metal diaphragm, valve piece and seat insert

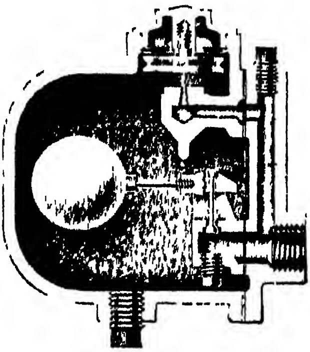


Fig 10 The Webster Size 00020-T Drip Trap is Rated 200 lb Water per Hour at 2 lb Pressure Difference

Series "26" A heavy duty trap for drips of mains, blast radiation, unit heaters, hot water generators and similar applications A rugged float-type trap available with and without thermostatic air vent Made in five sizes 200, 700, 1200, 2100 and 5000 lb water per hour at 2 lb pressure difference Maximum working pressure is 15 lb per sq in

Series "78" — thermostatic trap *built for* process steam pressures (10 to 125 lb per sq in) Monel Metal diaphragm Stainless Steel valve piece and seat insert Angle model only Sizes ⅜, ½, ¾ and 1 in Extensively used with laundry, cooking, sterilizing and other process-steam uses

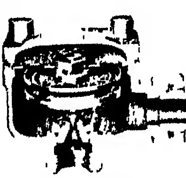


Fig 11 Webster Series "78" Trap

Series "79"—For use where large volumes of very hot condensate form more quickly than can be discharged by thermostatic traps alone Float and thermostatic traps designed for normal working pressures between 15 and 125 lb per sq in Water of condensation is passed through a float-controlled seat opening while air is discharged into the return piping by a thermostatically controlled vent Compact and light in weight Can be readily mounted in a pipe line without other support Available with either ¾ in or 1 in inlet and outlet

Cast iron body, copper asbestos gasket and cover bolted together with steel cap screws Monel Metal valve piece and stem Stainless steel seat Air vent unit is Monel Metal diaphragm with Monel Metal valve piece and brass seat with stainless steel insert

DIRT STRAINERS AND POCKETS

Placed in return lines of steam heating systems to prevent dirt, rust and scale from impairing tightness of traps

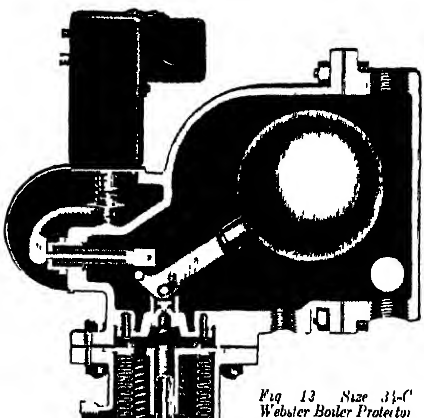


Fig 13 Size ¾-C Webster Boiler Protector with Low Water Electrical Cut-out Switch Size ¾ has no Cut-out Switch

BOILER PROTECTOR

Prevents breakage in low pressure heating boilers when water level becomes inadequate Automatically supplies raw water to boiler when water level drops to 1 in above bottom of gage glass

For maximum boiler pressure of 15 lb per sq in. Maximum cold water main pressure should not exceed 150 lb per sq in, minimum must not be less than 25 lb per sq in

Made with ¾ in connections, with or without electrical cut-out switch

Wright-Austin Co.

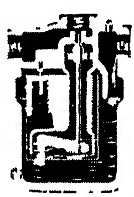
317 West Woodbridge St., **Detroit, Mich.**

PRODUCTS—Steam Traps, Strainers, Air Traps, Steam and Oil Separators, Compressed Air Purifiers, Exhaust Heads, Boiler Feeders, Alarm Water Columns, Water Gauges, Trycocks.

"Airxpel" Bucket Type Steam Traps

Are "double duty" traps, because they automatically discharge both air and condensate

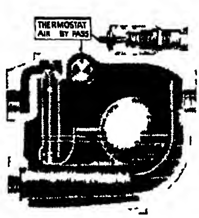
At left is illustrated the "Baby" Airxpel, a new streamline body design, which adds attractiveness to the installation Has the handiest type of pipe connection for radiation, etc



The Cub sizes are made in ½ in, ¾ in, 1 in Especially suitable for individual unit drainage on heating and process equipment

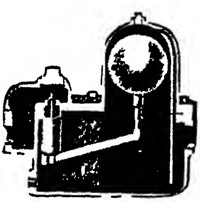
Also three "Master" sizes ½ in to 2 in, for general service

"Combination" Steam Trap



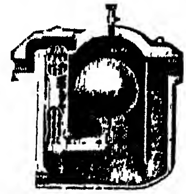
Float Type with internal thermostatic air bypass and strainer A modernly designed and very successful trap for vacuum and pressure heating

"Victor" Low Pressure Steam Trap

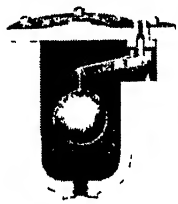


For heavy volumes of condensation at low pressures.

"Emergency" Float Type Steam Trap



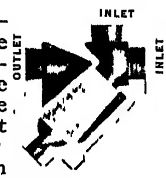
Three valve trap with large capacity at high pressures An exceptionally reliable trap for use in inaccessible places



Air Relief Trap

For relieving air from forced circulation hot water heating systems, water supply lines, closed tanks, receivers, pumps, etc

"Tuway" Strainer



May be used two ways—as a straight-way or angle strainer, in either horizontal or vertical pipe line, because it has the choice of two inlets at right angles to one another

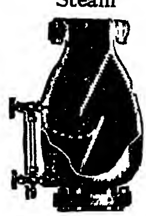
For cleaning, flush through blow-off connection, or remove screen by unscrewing bottom plug

Boiler Feeder

An iron watchman that automatically maintains uniform water level in low pressure boilers

Separators—Steam and Oil

Type "A" Vertical Steam



Type "S" Horizontal Oil

We make separators of every type.

Send for descriptive Bulletins on any of the items listed on this page

Yarnall-Waring Company

Philadelphia, Pa.

Branch Offices in Principal Cities

YARWAY

More than 100 Mill Supply Distributors Throughout the U S A

Manufacturers of the YARWAY Impulse Steam Trap

Principal Advantages of the Yarway Impulse Steam Trap

Light Weight—Yarway traps need no support—½ in trap weighs only 1⅛ lb 2 in trap weighs 8½ lb

Small Size—They practically eliminate radiation losses—can be installed in cramped quarters—½ in trap measures 2¼ in long—2 in trap, 4¾ in long

Will not air bind

Require no priming

Insure quick heating

Operate on exclusive Impulse principle (U S Patents No 2,051,732 Other Pat Pend)

Low Price—Often cheaper than repairing old traps

Six Sizes Serve Practically All Trap Requirements

Yarway Impulse Steam Traps in sizes from ½ in to 2 in are factory set for all pressures up to 400 lb Simply select the correct size from the table below

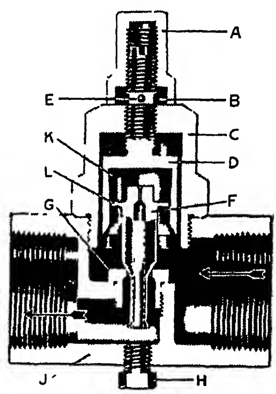
Operation—The Yarway Impulse Trap operates on an entirely new principle—the difference in flow characteristics of steam and water flowing through two orifices, with a chamber between Movement of valve is governed by variations in pressure in the space above the valve, called control chamber—these changes in pressure occurring with changes in temperature of condensate

When handling condensate, the pressure tending to lift valve is greater than reduced pressure in control chamber, and the valve opens As accumulation of condensate disappears, remaining condensate approaches steam temperature, flashing takes place, the flow is choked and pressure builds up in control chambers, closing valve

Construction—The Yarway Steam Trap is unique in that there is only one moving part—the simple valve F illustrated. This trap is made of "bar-stock" throughout—no castings used in its construction Body is cold rolled steel, valve and seat are of heat treated stainless steel, bonnet and cap of hex tobin bronze

Years of service of thousands of Yarway traps in leading plants have proven these materials suitable for the service.

A—Cap Nut
B—Lock Nut
C—Bonnet
D—Control Cylinder
E—Lock Pin
F—Valve
K—Control Chamber
L—Control Disc
G—Valve Seat
H—Test Plug
J—Body



Capacities—Maximum continuous discharge of condensate in pounds per hours at 30 deg below steam temperature

Pressure Pounds	½ In No 60	¾ In No 61	1 In No 63	1¼ In No 64	1½ In No 66	2 In No 67
10	475	825	1360	1910	2400	3425
20	650	1150	1725	2315	3050	4340
30	800	1375	1985	2650	3525	5050
40	900	1575	2220	2950	3950	5615
50	980	1725	2425	3200	4315	6100
60	1050	1850	2600	3425	4630	6525
70	1140	1975	2775	3660	4940	6930
80	1200	2080	2925	3875	5200	7260
90	1240	2175	3075	4075	5440	7600
100	1290	2260	3215	4280	5675	7875
125	1400	2460	3550	4760	6165	8530
150	1490	2640	3870	5175	6630	9075
200 to 400	1630	2900	4380	5960	7410	9950

Where greater capacity at low pressure is required, use two or more traps with check valve on discharge end of each trap

Prices, Weights and Dimensions

Size	Trap Complete	Wt Lb	Length Inches
½ in No 60	$15 00	1⅛	2⅜
¾ in No 61	22 00	1¾	2¾
1 in No 63	31 00	2½	3⅛
1¼ in. No 64	48 00	3⅞	3¾
1½ in No 66	68 00	6	4¼
2 in No 67	90 00	8½	4¾

For further information, send for descriptive bulletin—T-1721.

The Anchor Stove & Range Company, Inc.

New Albany, Indiana

ANCHOR KOLSTOKER

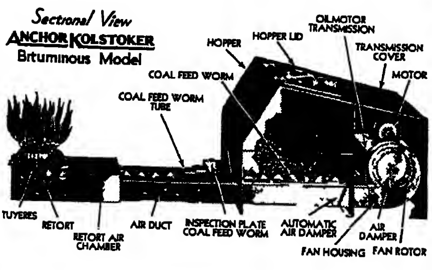
Over 1300 Representatives and Dealers in All Principal Cities and Towns

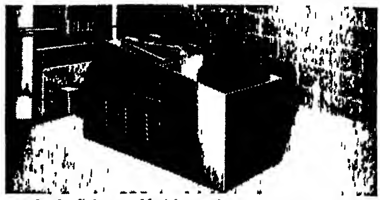
A top-quality stoker, designed by heating engineers and produced in a plant devoted for over 70 years to the manufacture of high-grade heating equipment Complete line, sizes and models for every domestic and industrial use Designed for and easily installed with any steam, vapor, warm-air, or hot water heating system or high-pressure boiler Under-feed principle, for efficient combustion of coal

Attractive cabinet design All working parts enclosed, assuring complete safety Room thermostat, limit control, hold-fire and relay, automatically control operation Exclusive double-action Oilmotor Drive, powerful, flexible, free moving parts, no springs, power-driven valves, sound-proof insulation to assure silent operation Low hopper—one-piece 10 gauge copper alloy steel, extra heavy, air-tight, joints electrically welded Big coal capacity, easy to fill Large diameter worm insures long life and efficient service Flexible as to length of worm and tube Pressure-operated automatic throw-out device—eliminating shearing pins and protecting motor against obstruction damage Alarm bell or light signal when throw-out operates Feed worm inspection plate makes removal of obstructions easy Automatic air control Motor rubber-cushion mounted, will not affect radio Sectional tuyere construction permits expansion and contraction without warping or cracking Absolute guarantee of satisfaction when installed and operated according to factory instructions

When you specify Anchor Kolstoker heat, you give your clients the convenience of gas or oil, with the safety and economy of coal Write for descriptive catalog and special folder for heating engineers

Sectional View
ANCHOR KOLSTOKER
Bituminous Model

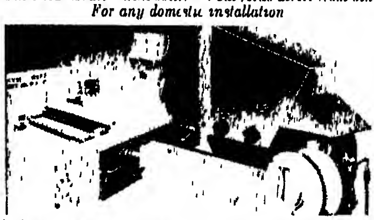




Standard Cabinet Model, with warm-air furnace—furnished for any type heating plant

Anthracite Model—automatic ash removal—with boiler Also Standard Type—for any heating plant

Bin Feed Model—with boiler Coal feeds direct from bin For any domestic installation



Industrial Model, Heavy Duty—furnished with either Dead Plates or Dump Plates in grate

Anchor-Arco Kolstoker-Boiler Automatic Heating Unit Developed together by two great companies

The Brownell Company

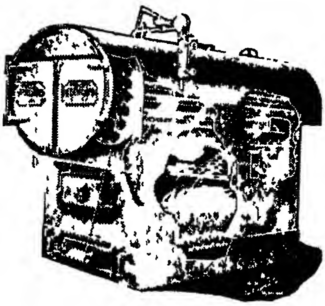
ESTABLISHED 1855

Dayton, Ohio

Manufacturers of

BROWNELL BOILERS AND STOKERS

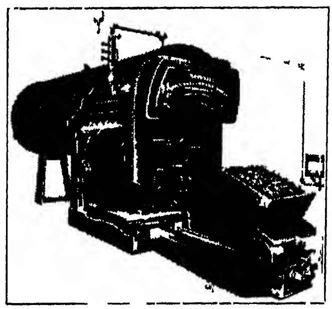
Representatives in All Principal Cities



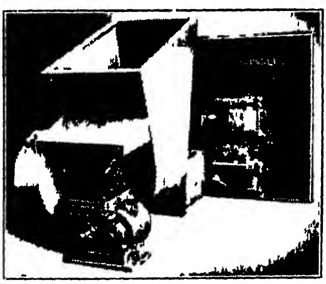
Brownell Standard Welded Smokeless Boiler

Welded Boilers built in Standard and Master types "Standard" type, Direct Draft or Smokeless, Coal Hand Fired – 500 to 34,000 sq ft Stoker Fired— 930 to 43,130 sq ft "Master" type, Coal Hand Fired—500 to 35,520 sq ft Stoker Fired— 1070 to 45,110 sq ft

Riveted Double Pass Firebox Boilers built for both high and low pressure Coal Hand Fired - 1000 to 35,000 sq ft Stoker Fired—4860 to 42,500 sq ft

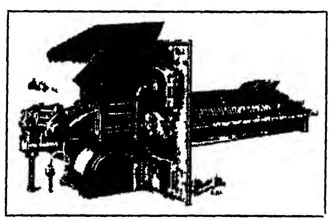


Brownell Riveted Double Pass Firebox Boiler, Stoker Fired



Brownell Type "C" Side Dump Stoker

The Brownell Type "C" Stoker is built in both stationary dead plate and side dump types, in sizes suitable for medium and large industrial and heating plants



Brownell Type "R" Heavy Duty Stoker

The Brownell Type "R" Heavy Duty Ram Feed Stoker is designed for use in the larger industrial and heating plants Ruggedness, flexibility, economy and low maintenance costs are features of this stoker

Brownell Matched Units, Boiler and Stoker combinations, are available in sizes from 500 sq ft to 45,000 sq ft

In addition to the above, a complete line of Domestic Stokers is built These Stokers are built in either ratchet or continuous feed

Send for Descriptive Bulletins

Detroit Stoker Company

Sales and Engineering Offices
General Motors Bldg., Detroit, Mich.

Main Offices and Works at
Monroe, Mich

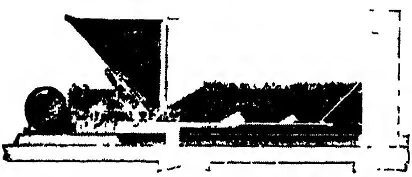
District Offices in
Principal Cities

Built in
Canada at London, Ont.

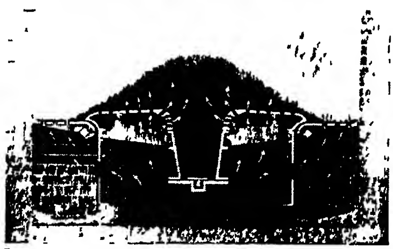
Since 1898

Detroit LoStoker

Built in various widths and lengths to fit furnaces of all types of boilers Compact, easily installed, responsive and automatic Savings, due to increased efficiency, combined with the ability to successfully burn less expensive grades of coal, make Detroit Stokers pay a handsome return on the investment *Write for Bulletin 369*



Detroit LoStoker, showing Motor Driven Blower and Stoker, One Compact Unit at the Front



Large Active Fuel Bed, with Provision for Admitting Air under the Dumping Grates at each side to burn out the Combustible Prior to Dumping Ashes

Detroit LoStoker Advantages include:

Agitator in coal hopper for positive coal feed Cannot stick or jam with wet coal

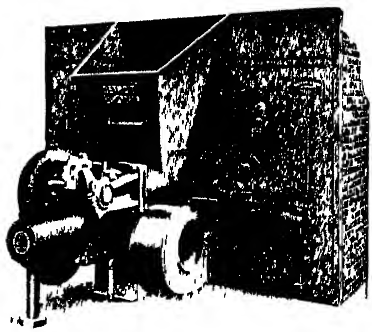
Adjustable Plunger Feed for the control of the quantity of coal

Heavy Mechanical Drive of simple design Little power is required for operation

Side Cleaning with dumping grates Ash doors are provided No hand cleaning.

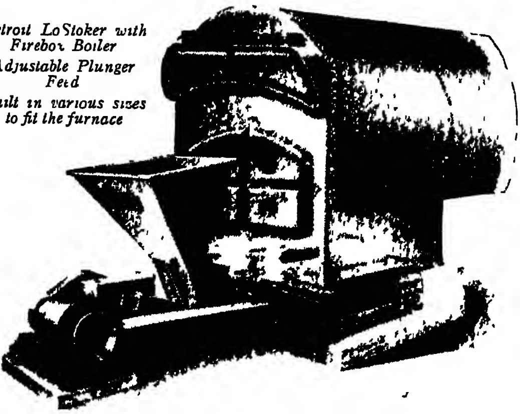
Automatically Controlled. Motor or turbine driven, controlled from steam pressure, water temperature or thermostat

Detroit UniStoker



Detroit UniStoker on Large Heating Boilers Each Boiler and Stoker, an Independent Unit

Detroit LoStoker with Firebox Boiler
Adjustable Plunger Feed
Built in various sizes to fit the furnace



Product and Service

Detroit Stokers: Built in various types and sizes to serve heating and power boilers from approximately 30 hp upwards. Bituminous coals obtainable in all sections are successfully burned Many features embodied in the various designs represent over 35 years of experience.

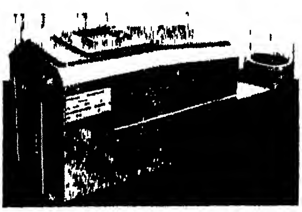
May we study your individual requirements. District Offices, located in principal cities, will furnish catalogues of any Detroit Stokers, or write DETROIT STOKER COMPANY, Detroit

Whiting Corporation

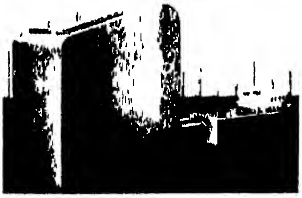
15620 Halsted Street, **Harvey, Ill.**
(Chicago Suburb)

Representatives in All Principal Cities

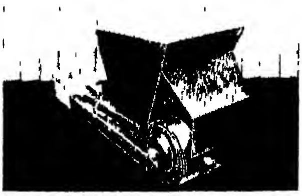
PRODUCTS—Complete line of Stokers and Pulverizing Equipment for firing furnaces and boilers (up to 5,000 h.p.) including Underfeed Stokers for small furnaces and boilers, Horizontal Compression Feed Stokers, Impact Pulverizers, Table Roller Pulverizers, Pulverized Coal Conveying and Feeding Equipment, Burners, etc.



STANDARD DOMESTIC MODEL No 2 and No 4 Capacities of 20 and 40 lb of coal per hour Famous Dual Draft Burner, Gas Eliminator, quiet 5-Speed Drive, Low Hopper and exclusive Whiting Heat Control Enclosed in smart cabinet Operating cost substantially less than any other type of automatic fuel



STANDARD DOMESTIC MODEL No 6 and No 9 Especially suitable for installation in large residences or small apartment buildings Available in capacities of 60 or 90 lb of coal per hour Whiting, now in its 53rd year, has had 17 years' experience in the manufacture of solid fuel conveying and combustion equipment



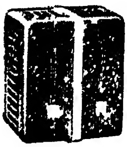
STANDARD UNDERFEED COMMERCIAL MODEL—For refractory or dead plate setting—ranging in capacity from 120 to 650 lb of bituminous coal per hour For all types of boilers—cast iron, horizontal return tube, vertical, down draft and brick set Famous Whiting Dual Draft Burner, Gas Eliminator and 5-Speed Drive

TWIN COMMERCIAL MODEL—In larger installations, 10 to 15 per cent more efficient than single retort design of same capacity Made to order only—for all classes of boilers—in capacities of 750—1,000—or 1,250 lb of coal per hour

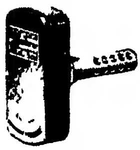
Barber-Colman Company

Rockford, Illinois

AUTOMATIC *Electric* CONTROLS



Barber-Colman Controls are all electric Precision built to insure long, continuous, and dependable service Easy to install in either new or existing installations Ready for instant service at all times, even after long shut down periods

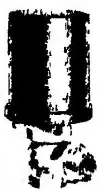


Thermostats. All types—room, duct, immersion and air-stream Single, duplex, two-temperature and heater Range and sensitivity to meet requirements

Hygrostats. Room, and duct types

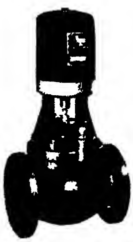
Motor-Operated Valves. Packless, packed, single seat, pilot piston, vee-ported, three-way, four-way, and butterfly

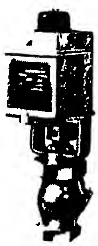
Solenoid Valves. For air, oil, water, gas, and refrigerants



Damper Control Motors. Unidirectional, or reversible, fixed or adjustable speed

Program Switches. Automatic contact-making mechanisms for multi-compressor control or similiar applications

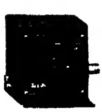


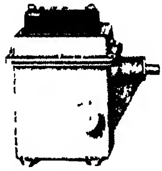


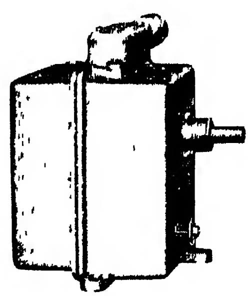
Micro controls give accurate control of modulating valves or dampers by operating them rapidly to a definite position for each different temperature at the thermostat

Literature is available describing complete automatic control of heating, ventilating, and air conditioning systems Consult a Barber-Colman representative, or write the factory

Listed as standard by Underwriters' Laboratories.



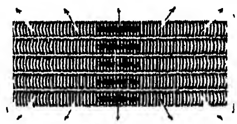




ENGINEERED AIR DISTRIBUTION OUTLETS

Aspiration Diffusion Distribution Directional Flow

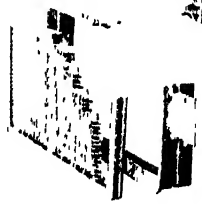
GRILLES Uni-Flo REGISTERS



Cores, of Uni-Flo fin and bar type construction, may be selected to give desired directional flow and throw, and permit placing outlets in most convenient locations. Smaller ducts and openings are possible because higher velocities and lower temperatures may be used without increasing the noise level or causing drafts.

Frames are available for either baseboard, side wall, panel or cabinet mounting.

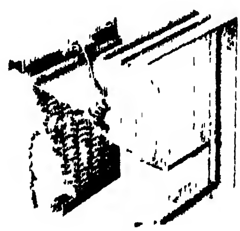
Finish: Plain metal, gray prime coat, gun metal, clear lacquer, brushed bronze, zinc cadmium, satin nickel, satin copper, buffed cadmium, brushed cadmium, polished nickel.



Registers are same construction as grilles, with the addition of spring loaded, positive closing, chain or key operated dampers.

Sizes. Grilles and registers are available in a number of standard sizes or may be made to any desired dimensions.

Special shapes are available to harmonize with any style of architecture.



Volume Control Deflectors control volume of air from a main into branch ducts.

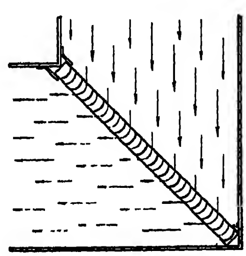
Elbow Turns provide uniform air flow around and beyond square cornered elbows.

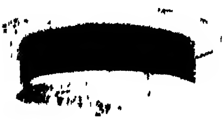
Air-Lites combine an attractive and efficient lighting fixture with either a supply or exhaust outlet, the latter being ideal for residential kitchens.

Ceiling Grilles are available in round, square or rectangular shapes. Used for either supply or exhaust or combination supply and exhaust.

Uni-Flo grilles, registers and accessories offer a solution to the most complex air distribution problems. Consult any Barber-Colman representative, or write the factory for further information.

AIR-LITE (right) *Combination supply (or exhaust) outlet and lighting fixture*





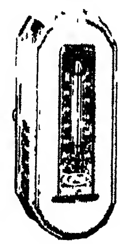
CEILING GRILLE (left) - *Round, square or rectangular Supply only, or combination supply and return*

Cook Electric Company

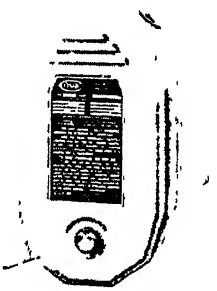
2704 Southport Ave , **Chicago, Ill.**

AUTOMATIC CONTROLS FOR HEATING PLANTS

COOK THERMOSTAT



An accurate low voltage thermostat of ultra modern design, of the heat accelerating or anticipating type It contains no magnets and may be adjusted to any reasonable differential Arranged for mounting on wall or electrical conduit box Base is of Bakelite, cover of brass, statuary bronze finish Size 2½ in x 5½ in x 1⅝ in Weight 12 oz

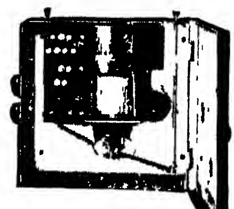


COOK DAMPER CONTROL

A thoroughly reliable damper motor backed by ten years' field and laboratory experience Absolutely quiet Requires no lubrication or attention Equipped with a unique firing clutch The safety feature always checks the fire should a fuse blow and interrupt the electric circuit

COOK No. 218 CONTROL SYSTEM

Complete thermostatic control, summer and winter, for both fire and blower for hand fired, solid fuel, mechanical warm air installations Consists of thermostat, furnace switch and No 21 unit

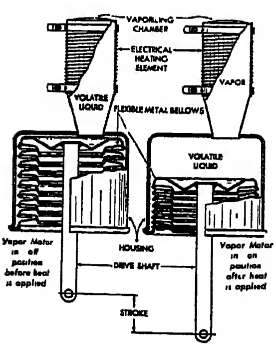


Cook No 21 Unit

COOK ZONE CONTROL

For complete warm air installations, either gravity or mechanical

COOK ELECTRIC VAPOR MOTORS



Cook Vapor Motor

Power exerted by vapor pressure, generated by an electric heating element For controlling dampers, shutters, valves and louvers on heating, ventilating, air conditioning and cooking devices The timing, length of stroke and power developed is variable over a wide range

COOK METAL BELLOWS

Of phosphor bronze, or Monel metal, for air, gas, steam, water, oils or ammonia Cook Bellows are made by crimping and soldering together individual diaphragms The temper, or hardness of the metal is controlled very accurately, as only a slight form of the diaphragm is necessary Cook Bellows are built to a wide range of diameter, width of flange and length to fit almost any application.

COOK ELECTRIC COMPANY—2704 Southport Ave , Chicago, have manufactured electrical equipment for more than thirty years and invite your inquiries

Detroit Lubricator Company

Detroit, Michigan, U. S. A.

New York, N. Y., 40 West 40th Street

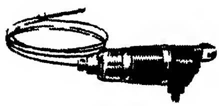
Chicago, Ill., 816 S. Michigan Avenue Los Angeles, Calif., 820 Crocker Street

Canadian Representative

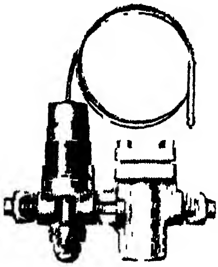
Railway and Engineering Specialties Limited, Montreal, Toronto, Winnipeg

Division of American Radiator & Standard Sanitary Corporation

Detroit Thermostatic Expansion Valve No. 673

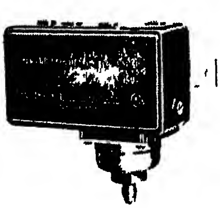


Detroit valves are scientifically designed to keep evaporators completely refrigerated under all conditions. Orifice sizes available from 1/32 in. to 7/32 in. with capacities up to 3½ tons on Freon or 6 tons on Methyl or Sulphur.



Detroit Thermostatic Expansion Valves Nos. 781-783 and 785

Large capacity valves for air conditioning installations. Capacities up to 20 tons on Freon and 35 tons on Methyl. Line Strainer illustrated available for large valves.

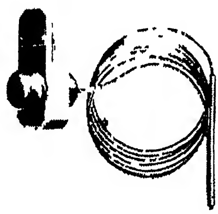


Pressure Control (Model RB-3)

Controls low side pressure. Available with high pressure cut-out to protect against high head pressures. Also available to control temperatures.

Two-Eleven Room Thermostat

A thermostat neat in appearance and of new design for heating, cooling or in combination for both heating and cooling. Supplied with or without adjustable compensator providing uniform control.



Differential Thermostat No. 691

An inexpensive room thermostat for room cooling which modifies indoor temperature in accordance with outdoor temperature to maintain comfort conditions. Provides economy of operation and prevents shock due to over-cooling.

Duct Damper Motor No. 431

An inexpensive means of providing individual temperature control for zones or groups of rooms. Is quiet and can be mounted directly on the duct. Furnished with auxiliary switch to control heating equipment. Neat in appearance and easily installed.

Other Controls

This Company can also supply you with Refrigeration Solenoid Valves, a full line of Boiler and Furnace Limit Controls—Room Thermostats both high and low voltage for both heating and cooling—Fan controls—Humidity and Stoker Controls and Solenoid Valves for the control of water, fuel and gas.

Julien P. Friez & Sons, Inc.,

(A Subsidiary of the Bendix Aviation Corporation)

Baltimore, Maryland, U. S. A.

Established 1876

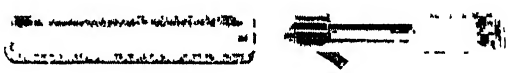
Manufacturers of a Complete Line of Automatic Electric Controls for Industrial and Comfort Applications. Also a Complete Range of Recording and Accurate Measuring Instruments for Indoor and Outdoor Applications

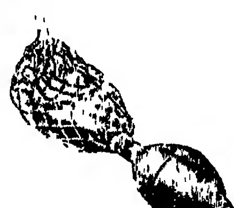
New and Advanced Products for the Coming Year, Ready for Delivery

Humidistat, new model of standard "hair-trigger" control, employing the famous Friez human hair element for controlling humidity for either comfort or industrial purposes

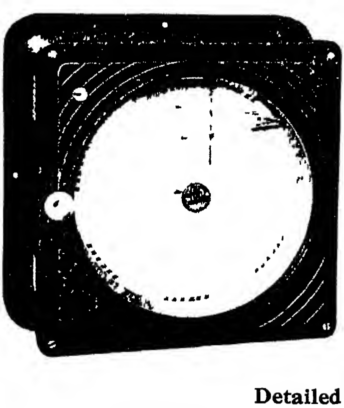
Comfortrol, a temperature control integral with a humidity sensitive element which automatically varies the temperature setting to conform to the needs of changing humidity for human comfort Unique and exclusive effective temperature control

Windowstat, a humidity control device for location indoors at the window It maintains humidity, in any humidity system, just below the condensation point, thus preventing frosting before it begins





Hand Aspirated Psychrometer, wet and dry bulb readings taken quickly, accurately, and inconspicuously, without any whirling Readings are made with instrument in one hand while other hand operates a bellows by which proper ventilation is induced Pocket size, the new Psychrometer for Air Conditioning



Temperature and **Humidity Recorder,** which records conditions either of immediate surroundings, or from distant locations Humidity is read directly from the chart in percentage relative humidity without computations or recourse to tables Unique Air-condition Recorder

Relay-Transformer, in one integral unit at the price of a moderately priced relay Designed for use wherever a relay is required in Air-conditioning, producing low voltage service at high voltage costs

Factory representatives in leading cities, and complete catalogues and engineering service for any project requiring expert counsel and design

Detailed Bulletins on Request

America's Specialist in Air Conditioning Controls

The Fulton Sylphon Company

Manufacturers of Sylphon Automatic Temperature Controlling Instruments and Packless Expansion Joints

Knoxville, Tenn.

Sales Representatives in Principal Cities

PRODUCTS

Temperature Regulators for Storage Water Heaters; Thermostatic Water Mixers; Refrigeration Temperature Regulators; Packless Expansion Joints; Pressure Reducing Valves. (For Sylphon Systems of Temperature Control for heating, ventilating and air conditioning . . . see Page 1123.)

GENERAL INFORMATION

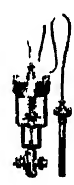
Fulton Sylphon Products depend upon the famous Sylphon Metal Bellows for their trouble-free service and long life

Over thirty-five years ago, THE FULTON SYLPHON COMPANY originated this bellows a precision-built, seamless, jointless "miracle in metal"

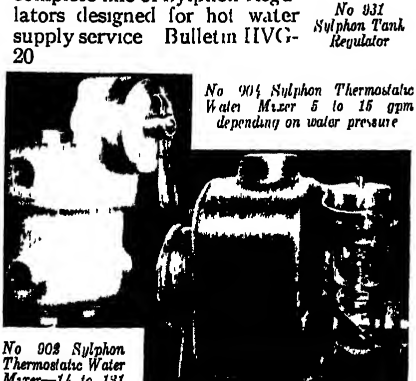
By continuous engineering study and intimate contact with heating and refrigeration problems, this company has used this efficient, practically indestructible bellows in the development of a line of temperature control and heating specialties known everywhere for outstanding service and quality

HOT WATER SUPPLY

No. 931 Sylphon Regulator—Widely used for controlling hot water service heaters, accurate, reliable, long lived, can be installed in any position facilitating use in crowded locations One of a complete line of Sylphon Regulators designed for hot water supply service Bulletin HVG-20



No 931 Sylphon Tank Regulator



No 904 Sylphon Thermostatic Water Mixer 5 to 15 gpm depending on water pressure

No 902 Sylphon Thermostatic Water Mixer—14 to 131 gpm depending on water pressure

Sylphon Thermostatic Water Mixers—Utilize hot water from any storage tank or instantaneous heater, and effectively regulate the amount of cold water required to temper it to the desired degree, actually mixing the hot and cold water together before delivery

Readily adjustable for any desired water temperature, this temperature remains constant in spite of fluctuations in supply water temperatures or pressures

Four sizes with capacities ranging from 5 to 325 gpm Bulletin HVG-40

REFRIGERATION CONTROLS

A complete line of Sylphon Refrigeration Controls, adaptable to the regulation of temperatures down to 40 deg below zero and wherever brine is used as the refrigerant

Latest development of Fulton Sylphon Engineers is a "freeze-proof" valve (illustrated at right on the popular Sylphon No 945-Z Regulator) This valve will work freely even when subjected to temperatures as low as 70 deg below zero Bulletin HVG-20

No 945-Z Regulator Showing detail of "freeze-proof" valve

PACKLESS EXPANSION JOINTS

The Sylphon Packless Expansion Joint eliminates useless building height, expensive construction and non-revenue producing space It prevents costly leaks and repairs, requires no repacking, and because it is always tight, it allows the heating system to operate at full efficiency Thousands are in use Write for Bulletin HVG-140

No 110 Sylphon Expansion Joint

PRESSURE REGULATORS

No. 952—This regulator of all-metal construction may be used as a pressure reducing valve as well as a uniform pressure regulator It may be installed in any position, easily adjusted for different pressures.

No 952 Sylphon Pressure Regulator

Johnson Service Company

TEMPERATURE AND HUMIDITY CONTROL

General Offices and Factory

Milwaukee, Wis.

Branch Offices in all Large Cities

Johnson Temperature Regulating Co of Canada Ltd , 97 Jarvis Street, Toronto, Ont
Montreal, Que Winnipeg, Man Calgary, Alta Vancouver B C

Products and Services

Manufacturers, engineers, and contractors for Automatic Temperature and Humidity Control Systems applied to all types of *heating, cooling, ventilating, and air conditioning installations* A single nation-wide organization devoted to Design, Manufacture and Installation for more than 50 years

Temperature and Humidity Control for every range required in *manufacturing and industrial processes*

Room temperature control applied to radiators, unit ventilators, and heat delivery ducts *Johnson "Duo-Stats"* to maintain the proper relationship between outdoor and radiator temperatures for groups of radiators or "heating zones "

A complete line of devices for automatic control of *air conditioning systems*, heating, cooling, humidifying, dehumidifying Automatic seasonal shifting of control cycles.

Periodic Flush Systems for intermittent flushing in various sections of a building, reducing load on piping system and insuring economy in use of water

Special bulletins and catalogues on request Sales Engineers at branch offices in all principal cities

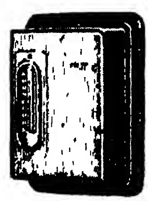
Room Thermostat

Johnson All-Metal Thermostats

Johnson thermostats, room, insertion, and immersion types, are all-metal throughout, no soft or hard rubber parts to deteriorate and become inoperative Every thermostat precise in construction and thoroughly tested for accuracy, efficiency, and durability

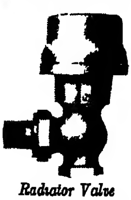
The Johnson *intermediate action thermostat* gives a true graduated action to mixing dampers and valves It holds them in an intermediate position to maintain the temperature of the room accurately within one deg above or below the setting of the thermostat, if desired

Johnson "Dual" or Two-Temperature Thermostats



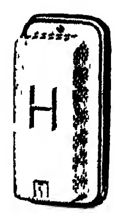
Dual Thermostat

The *Dual, two-temperature,* room thermostat especially adapted for use where various rooms or groups of rooms are occupied when the remainder of the building is not in use Separate steam mains avoided The shifting from "day" or occupancy temperature to an economy temperature for "non-occupancy" conditions, accomplished by a switch or Johnson program clock at a central point Push buttons on the thermostat are provided when "occupant control" is desired Dual thermostats are all-metal and operate valves and dampers gradually to maintain temperature accurately within one degree



Radiator Valve

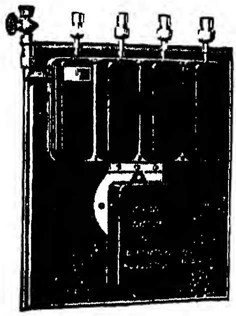
Johnson Valves



Room Humidostat

Johnson *diaphragm valves* are simple and rugged. Seamless metal bellows and heavy spring operate the valve stem No complicated moving parts Made in all standard sizes and patterns Direct acting (normally open) or reverse acting (normally closed) Three-way mixing and three-way bypass valves

For steam, water, brine, and Freon service Johnson valves are available, if desired, with diaphragms of special moulded rubber, super-aged and heat-resistant



Four-Point Insertion Thermostat

Humidostats and Humidifiers

The *Johnson Humidostat* automatically controls the supply of moisture delivered to the air by a humidifier or air washer and maintains a constant percentage of relative humidity Available in both room and insertion patterns, and with various types of elements as determined by requirements, controlling within 1 per cent at relative humidity of 95 per cent and 100 deg Fahr if desired

Johnson humidifiers are furnished in steam "grid" type or pan type with copper evaporating pan, brass heating coil, and float control

Air Conditioning Control

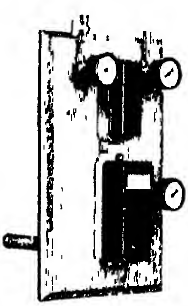
Summer-Winter room thermostats for operation of valves and dampers in reverse sequence for cooling and heating *Insertion* and *immersion* thermostats in one, two, three, and four-point patterns for operating valves and dampers successively at different temperatures

Remote readjustable thermostats, reset from a distant point by pilot or differential thermostat or by pressure switch *Differential thermostats* to maintain desired temperature differences between two points, such as outdoors and treated space

Solenoid air switches, *manual switches*, *static pressure regulators*, *velocity regulators*, operating dampers to regulate flow of air in ducts

The *Johnson sensitivity adjustment* is an important development in the field of automatic temperature and humidity control for air conditioning A unique and convenient means of adjusting the sensitivity of *Johnson* thermostats and humidostats *on the job*, with respect to the capacity of the conditioning apparatus

Summer-Winter Thermostat



Remote Readjustable Thermostat

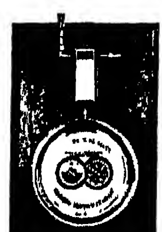
Zone Control

Johnson "Duo-Stats" to regulate the flow of heat in a group of radiators constituting a "heating zone" by maintaining the proper relationship between outdoor and radiator temperatures

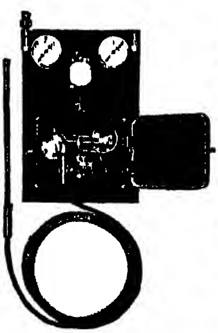
Process Control

Calibrated insertion thermostats for controlling temperature of liquids, air and gases Mercury *extended tube thermostat* for remote location of sensitive element

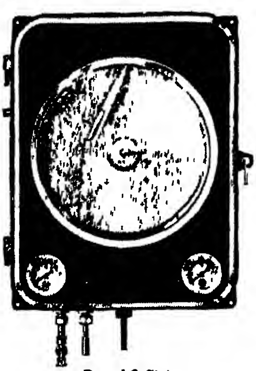
Wet-bulb thermostats for close regulation of humidity "Record-O-Stats," combination instruments to record and control temperatures



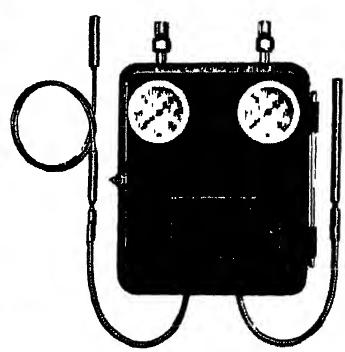
Program Clock



Mercury Extended Tube Thermostat



Record-O-Stat



Johnson "Duo-Stat"

GENERAL CONTROLS

1505 BROADWAY
CLEVELAND, OHIO

1368 HARRISON ST.
SAN FRANCISCO, CALIF.

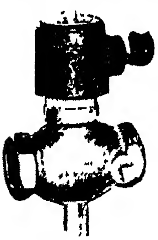
DEPENDABLE CONTROL OF TEMPERATURE-PRESSURE-FLOW

There is a GENERAL magnetic valve specifically designed to control each fluid and gas For example, the GENERAL K-10 lever action gives it the power to operate dependably on oils as heavy as No 6 The K-15 handles refrigerants without corrosion, the B-6 is particularly adapted to zone control, the K-3B is characteristically suited to gas, the K-12 to ammonia, etc Please write for complete catalog

MAGNETIC GAS VALVE TYPE K-3B

Primary Applications—Controlling gas to hot-air furnaces and industrial boilers, safety shut-off

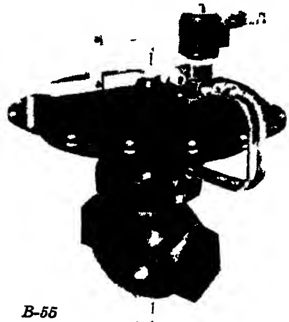
The GENERAL K-3B is a two-wire, straight-magnetic, current-failure valve of packless construction. Closing with the flow, the pressure is on top of the seat, insuring a tight shut-off, indefinitely Solenoid is absolutely humless—a feature originated by GENERAL CONTROLS Manual by-pass allows for opening in case of prolonged current failure Terminal boxes may be turned in any direction to facilitate wiring



K-3B

ELECTRIC SLOW OPENING GAS VALVE—TYPE B-55

Primary Application—Simultaneous control of gas flow and damper opening

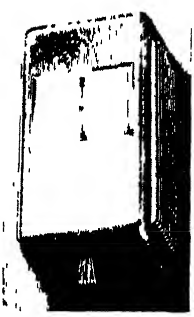


B-55

The B-55 is a diaphragm valve with an adjustable opening time of from 5 to 60 sec, using the GENERAL current-failure solenoid as the pilot valve

THERMOSTAT—THE METROTHERM

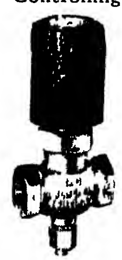
Adjustable heat acceleration is featured in this beautiful new METROTHERM Low thermal inertia guarantees instant response to small temperature change and eliminates under- and over-shooting of set temperature Luxuriously finished in sprayed silver and chrome



REFRIGERANT CONTROL TYPE K-15

Primary Applications—Controlling the flow of methyl, SO Freon or Brine, water, air, gases

The GENERAL K-15 is a full-ported, current-failure, pilot-operated solenoid valve of packless construction Closes with the pressure on top of the seat, giving an absolutely tight shut-off All materials are adapted to refrigerants and are non-corrosive



K-15

SEMI-BALANCED—TYPE B-6

Primary Applications—For water, air, gas, oil, zone control

The GENERAL B-6 is a semi-balanced, current-failure solenoid valve valuable in the automatic regulation of large flows Specifically designed for zone control in hot water heating systems Humless and quiet in operation

B-6

Minneapolis-Honeywell Regulator Company

Factories MINNEAPOLIS, MINN, PHILADELPHIA, PENNA, AND WABASH, IND

Branch Offices

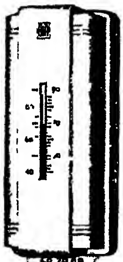
Akron
Allentown
Atlanta*
Austin, Minn
Baltimore
Birmingham
Bismarck, N D
Boston*
Buffalo*
Butte
Charlotte N C
Chicago*
Cincinnati
Cleveland*
Columbus
Dallas*
Dayton
Denver*
Des Moines
Detroit*
Duluth
East Orange
Eau Claire Wis
Evansville
Fargo, N D
Fort Wayne
Grand Rapids
Hartford*
Haverhill, Mass
Houston
Indianapolis*
Jackson, Mich
Kalamazoo
Kansas City*
Little Rock
Los Angeles*
Louisville, Ky
Memphis
Milwaukee*
Minneapolis—
St Paul*
Mitchell, S D
New Orleans
New York*
Oklahoma City
Omaha*
Peoria
Philadelphia*
Pittsburgh*
Portland*
Providence*
Reading
Rock Island, Ill
St Louis*
Salt Lake City*
San Antonio*
San Francisco*
Scranton
Seattle
Sioux Falls, S D
Springfield, Mass
Syracuse*
Toledo
Tulsa
Washington, D C*
Wichita*
Worcester, Mass
York, Pa
Youngstown

In Canada Montreal, Toronto,* Calgary, Winnipeg **In Europe** Amsterdam, Holland,* London, England*

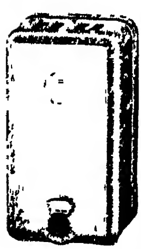
*Stock carried at these offices

THE MODUTROL SYSTEM OF AUTOMATIC CONTROL

for Heating, Ventilating and Air Conditioning



Room Thermostat



Humidity Controller

Electric Motor Radiator Valve

The Modutrol System designation is applied to any combination of Minneapolis-Honeywell Automatic Electric or Pneumatic Controls or Self-contained Automatic Valves used to govern the operation of air conditioning or heating systems other than the small domestic installations.

INCLUDES SUPPLEMENTARY EQUIPMENT

Supplementing the modulating controls is a wide variety of on-and-off or two-piston motors, controllers and valves, both electric and air actuated, thus making the **Modutrol System** extremely flexible as to the selection of control equipment to produce the desired results

EMPLOYS BOTH ELECTRIC AND PNEUMATIC CONTROLS

The **Modutrol System** provides both types of control, thus enabling us to make a real contribution to the heating, ventilating and air conditioning industry With the **Modutrol System** Minneapolis-Honeywell is prepared to offer comprehensive and unbiased recommendations for installation in all types of buildings, existing or new, large or small, at prices commensurate with the needs of each project These recommendations include

1. Pneumatic control only, where that type of control will give satisfactory service and where low first cost is of special importance.
2. Combination pneumatic and electric control where a somewhat higher degree of accuracy is needed
3. Electric control in those installations where precise, flexible and dependable results are required

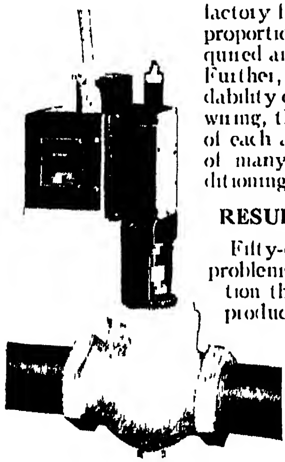
REDUCES COST OF HEATING OR AIR CONDITIONING SYSTEM OPERATION

In new buildings or old, a lowering of operating costs, and an immense improvement in the comfort delivered, can be accomplished by considering the control system as an integral part of the heating system In fact, automatic control systems today cannot be considered as accessory to the heating systems.

PROVIDES TRUE MODULATION

Viewed from the performance standpoint, the electric controls of the **Modutrol System** offer a dependable means of effecting true modulation of the temperature, air flow and humidity, though our pneumatic controls provide a degree of modulation which is satis-

See also Page 983



Motorized Valve

factory for many installations True modulation means automatic proportioning of heat, cold air flow, or humidity in exactly the required amount to offset changes in the heat loss or relative humidity Further, in the Electric **Modutrol System** the well known dependability of electric switches and motors, the permanency of electric wiring, the flexibility of self-contained units and the accessibility of each and every portion of the equipment represent just a few of many advantages offered in this modern means of air conditioning control

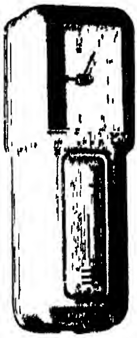
RESULT OF OVER A HALF CENTURY'S EXPERIENCE

Fifty-one years of concentration upon temperature control problems has built up within the Minneapolis-Honeywell organization the knowledge, technique and engineering ability which have produced the **Modutrol System** as well as the comparatively simple domestic control systems

ENGINEERING SERVICE

The Minneapolis-Honeywell Automatic Control engineer is at your service at all times He will be glad to furnish you with recommended control layouts and cost estimates

He is trained to recommend control results before installation of equipment and to produce control results after the installation of equipment has been completed



Electric Clock Thermostat

COMPLETE INSTALLATION SERVICE

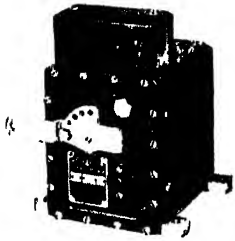
Minneapolis-Honeywell Branch Offices are equipped to make the complete installation of the **Modutrol System** for control of Air Conditioning or Heating installations Thoroughly trained men are also available to supervise, adjust, or service the control equipment A stock of standard control equipment is carried at more than twenty points throughout the country for quick delivery

RESPONSIBILITY FOR ENTIRE CONTROL SYSTEM

The **Modutrol System** and its supplementary equipment is so complete that the Minneapolis-Honeywell Regulator Co is equipped to assume the entire responsibility for any control installation, thereby eliminating the difficulties and misunderstandings which the division of responsibility may create

TYPICAL SPECIFICATIONS

A complete set of typical specifications covering automatic control systems for use in heating, ventilating, and air conditioning is available for use by engineers and architects



Modutrol Motor

DESCRIPTIVE LITERATURE

Catalogs arranged according to the following subdivisions are available upon request

1. **The Modutrol System.**
2. Air Conditioning Controls
3. Oil Burner Controls
4. Gas Heating Controls
5. Stoker Controls
6. Industrial Regulator Controls
7. Refrigeration Controls.
8. Complete Condensed Catalog
9. Brown Instruments for Heating, Ventilating and Air Conditioning

In addition to this literature, complete data sheets are available, including technical information on all equipment in the **Modutrol System** which is of prime interest to the engineer. These pages will be furnished on request.

National Regulator Co.

Manufacturers and Contractors

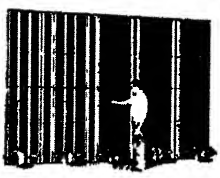
Factory and General Offices

2311 Knox Avenue, **Chicago, Ill.**

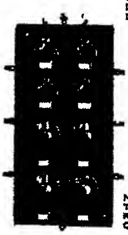
Complete Systems for Control of Temperature, Humidity, Ventilation and Air Conditioning. Metaphram Damper Regulators for Domestic Heating Boilers and Tank Heaters. A-Jacks High Pressure Steam Damper and Combustion Control.

NATIONAL THERMOSTATS FOR ALL TYPES OF OPERATION

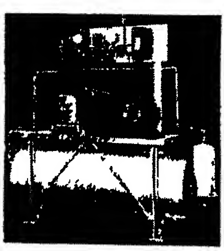
METAPHRAM VALVES IN SIZES FROM ¾" TO 10"



METAPHRAM DAMPERS FOR ALL DUCT WORK



NATIONAL CONTROL BOARDS FOR REMOTE AND ZONE CONTROL



NATIONAL AIR COMPRESSORS FOR UNFAILING OPERATION OF COMPLETE NATIONAL SYSTEMS

A National Control System for temperature, humidity, ventilation or air conditioning comprises coordinated equipment easily understood by local operating engineers Each system is planned to meet the requirements of the individual building to assure proper regulation

National Thermostats (Room Type)—Air operated, simple, two-temperature and compound types for direct radiation, direct radiation and unit ventilating machines, straight blast heating, ventilating fan units and dampers

National Thermostats (Duct Type) – Air operated, for ventilating duct or blast heating control and hot water tanks

National Thermostats (Industrial) for control of industrial and process temperatures

Metaphram Valves—Air operated, for direct radiation, steam lines, hot water tanks, humidifiers and accumulator control.

Metaphram Dampers—Air operated by Metaphram motors, for accurate automatic control of ventilation and blast heating Built in round, double or louvre types, of black or galvanized steel or special metals

National Control Boards—Located in engineers room or central place for remote or zone control of pneumatic or electric-pneumatic switches in connection with two-temperature and zone control systems under manual or time-clock operation, operating valves or steam lines, radiation, or ventilating fan units and dampers

National Air Compressors—Self-contained, automatic units for unfailing operation of National apparatus and equipment

Metaphram Damper Regulators for all domestic hot water or low pressure heating boilers, gas, oil or coal fired.

A-Jacks Control for high pressure boilers (15 lb to 300 lb pressure) giving synchronized control to boiler pressure and fuel consumption

Catalogs and Bulletins—Thoroughly illustrated bulletins are available on all products Engineering assistance will be rendered without obligation

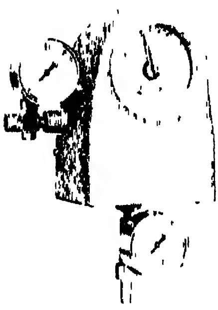
National Regulator Co.

Manufacturers and Contractors

Factory and General Offices

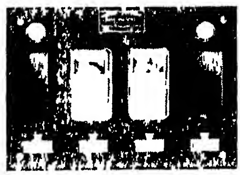
2311 Knox Avenue, **Chicago, Ill.**

IMPROVED INSTRUMENTS FOR AIR CONDITIONING



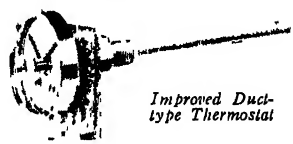
Extension duct type Submaster Thermostat

The duct and room type of Submaster Thermostat (*left*) are readjustable by a master or pilot thermostat measuring the temperature at a distant point or by a graduated switch or a humidity controller Pressure changes in the branch line leading from master stat or switch cause a readjustment of the controlling temperature point of the Submaster The instrument has simple adjustment mechanism so that the ratio of change may be modified



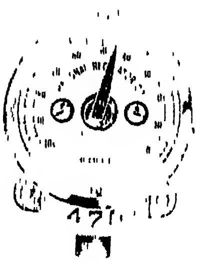
National Gauge Board

Temperature Control System directed by program clocks



Improved Duct-type Thermostat

Improved duct or insertion type stat of brass and Invar steel, closely calibrated Made in straight and reverse acting, gradual or positive Dewpoint and wet-bulb stats and air stream stats



Humidity Controller

Humidity Controller (*left*) with indicating dial graduated in percent of relative humidity and accurately calibrated Redesigned to make a sturdy, dependable instrument and one easily serviced

Summer-winter Thermostats (*right*) are made direct acting for winter use and reverse acting for summer use The summer control point may be at a higher temperature than in winter The switch-over from winter to summer may be made by manual switch or by pilot stat feeling outdoor condition

Room Type Submaster Summer-winter Thermostat

Metaphram remotely controlled Zone Valves (*left*) The unit illustrated is a 12 in valve Valves of this type have been manufactured up to 16 in in size

Zone Control Valve

Sensitivity Control. Many National Thermostats are provided with mechanism so that sensitivity may be adjusted on the job. This is a desirable feature in that it fits the thermostat to various conditions of time lag or load change.

Also static pressure regulators, positive and gradual switches, positive and gradual relays, electric-pneumatic valves.

THE POWERS REGULATOR CO.

45 Years of Temperature and Humidity Control

Offices in 45 Cities—See Your Phone Directory

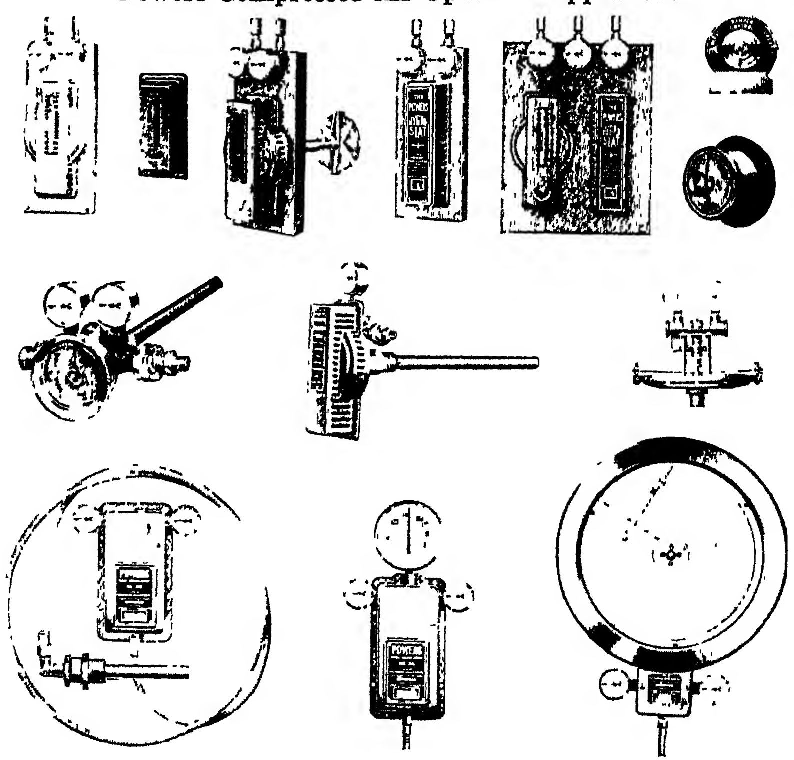
General Offices and Factory: 2719 Greenview Ave., CHICAGO;
General Eastern Office: 231 East 46th Street, NEW YORK;
1808 W. Eighth Street, LOS ANGELES;
The Canadian Powers Regulator Co., 106 Lombard St., TORONTO, ONT.

PRODUCTS—A very complete line of compressed air operated and self-operating temperature, humidity and air flow controls for automatically regulating heating, cooling, ventilating and air conditioning systems and industrial processes.

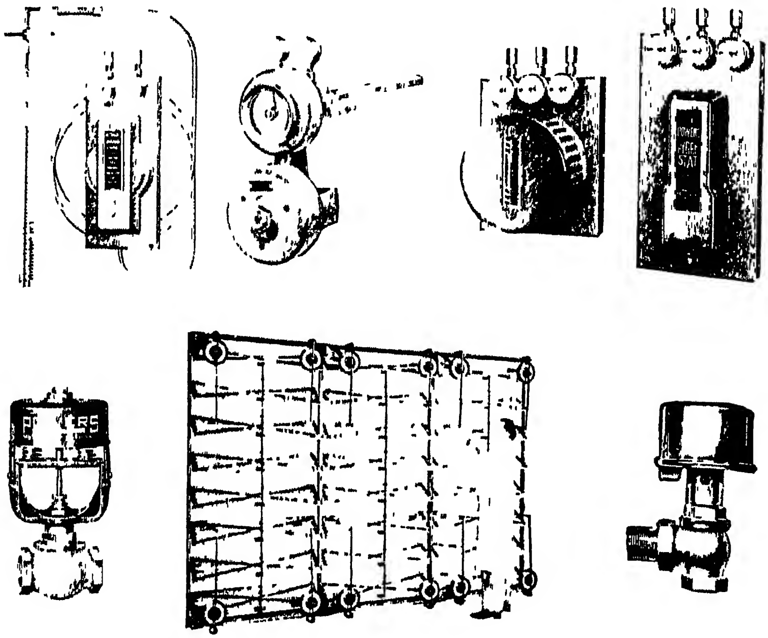
A complete line of self-operating and compressed air operated valves and regulators made for: Controlling steam heated hot water heaters, and submerged type heaters; and for automatically mixing hot and cold water or steam and cold water delivering a mixture at a predetermined temperature.

Dial Indicating and Recording Thermometers. High pressure steam traps and pressure reducing valves.

Powers Compressed Air Operated Apparatus



Powers Compressed Air Operated Apparatus



POWERS ENGINEERING SERVICE

As the accurate performance of a heating, cooling, ventilating or an air conditioning system, or an industrial process is so dependent upon its automatic control equipment, and as the cost of such control is but a fraction of the entire system, the use of the proper type of regulation is always sound economy

To secure the maximum return on the investment in automatic control equipment, it is exceedingly important that proper selection of control apparatus be made when each installation is being planned

Forty-five years of experience in furnishing and installing temperature and humidity control for every conceivable purpose in all types of buildings has given us a wealth of experience from which you can draw in selecting the proper type of control for any purpose

CATALOGS AND BULLETINS describing any or all of our products furnished upon request Phone or write our nearest office See your phone directory.

Penn Electric Switch Co.

Des Moines, Iowa

Factory and Executive Offices to be located at

Goshen, Indiana

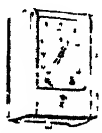
after April 15th, 1937

Offices
NEW YORK, BOSTON, DETROIT, CHICAGO

Export
100 Varick Street, NEW YORK CITY

Distributors in principal cities

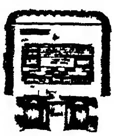
Representatives
GARLAND-AFFOLTER ENGR CORP, San Francisco Seattle, Los Angeles, FORSLUND PUMP AND MACHINERY CO, Kansas City, THE UHL CO, Minneapolis, JULES BENEKE, St Louis, MONARCH SALES, Denver

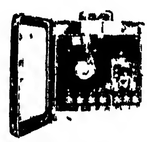

Type 770 Tem-Clock


Type 860 Temtro


Type B Thermostat


Type 703 Relay


Type 900-D Gas Valve


Type 550 Stoker Control

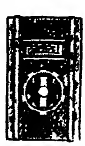
Many years experience in automatic control problems have equipped the Penn Electric Switch Company to produce controls to meet any and every condition encountered in temperature and air conditioning control

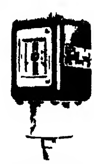
Each device illustrated, and all others produced by Penn have the vital differences which have won Penn it's enviable position in the field of control

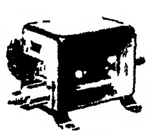
Penn engineers and field representatives are available for consultation with manufacturers having problems involving the use of control mechanisms We offer this service to manufacturers without cost A request on your letter head will place the entire facilities of the Penn organization at your disposal

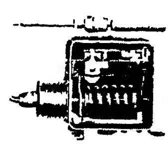

Type 430 Immersion Limit Switch

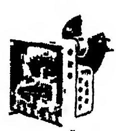

Type 250 Water Valve—1¼" Size

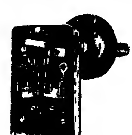

Type 724 Timetrol


Type 400 Steam Pressure Limit Switch


Type 207 Refrigeration Control


Model LRT Unit Heater Control


Type 632 Saftrol


Type 503-G Fan and Limit Switch

Penn Consulting Engineering Service is available on request

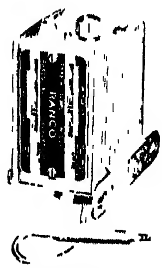
Send for literature on any control listed or for complete catalog.

Ranco, Inc.

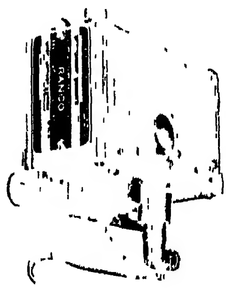
601 West Fifth Avenue, **Columbus, Ohio**

For 23 years makers of circuit breakers, precision domestic, commercial and air conditioning controls.

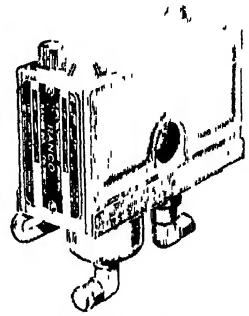
Representatives and Jobbers in Principal Cities



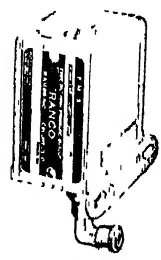
Temperature Control



Temperature and High Pressure Cut-Out Control



High Pressure Cut-Out and Pressure Control



High Pressure Cut-Out

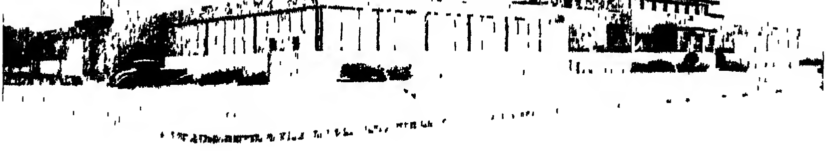
Specifications of RANCO Type "RL" Commercial Controls

	Code	Setting Out	Setting In	Range	Differential	Bellows Fitting	
HIGH PRESSURE CUT-OUT	RLH 850	135 lb 175 190		110 lb to 200 lb	Non-Adjustable 30 lb	Right angle fitting may be turned in any direction 1/4" SAE MALE THREAD	Opens circuit on high pressure, closes circuit on return to normal
PRESSURE CONTROL	RLP 860 Constant Cut-Out	0 lb	5 lb	20 to 40 lb Cut-out Pressure	Cut-in can be adjusted 5 lb to 45 lb above a constant cut-out pressure	Right angle fitting may be turned in any direction 1/4" SAE MALE THREAD	Closes circuit on increase of pressure, opens on decrease of pressure
	RLP 861 Constant Cut-In	20 lb	25 lb	12" to 50 lb Cut-in Pressure	Cut-out can be adjusted 5 lb to 40 lb below a constant cut-in pressure		
PRESSURE CONTROL WITH HIGH PRESSURE CUT-OUT	RLCP 1000	RLP 860 and RLH 850 combined in one case					
	RLCP 1001	RLP 861 and RLH 850 combined in one case					

ELECTRICAL RATING—1½ HP 110-220 V AC—¾ HP 115-230 V DC—1 Amp 550 V AC

Ranco Controls possess many exclusive and practical features, besides the fundamental characteristics of absolute dependability, maintained accuracy of calibration and differential accessability and long life

The above listed items are but a portion of the instruments we manufacture Our services are at your disposal in meeting your control problems



Spence Engineering Company, Inc.

28 Grant Street, **Walden, N. Y.**

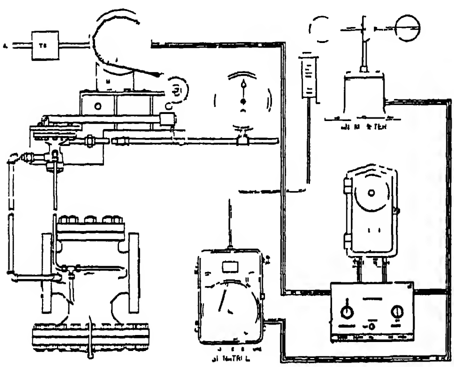
Manufacturers of Pressure Regulators, Pump Governors, Weather Compensators, Self-Cleaning Strainers, Seco Metal.

Advantages of Spence Regulators

Accurate Regulation—Regardless of fluctuations in either load or initial pressure Spence Pilots fit any size main valve, are connected to main valve with unions, and are guaranteed to hold a dead-end All main valves and most pilots are packless

A balanced *single seat* is used even in large sizes, of SECO Metal, guaranteed to resist wiredrawing action of steam The metal diaphragms, under normal conditions, never require replacement

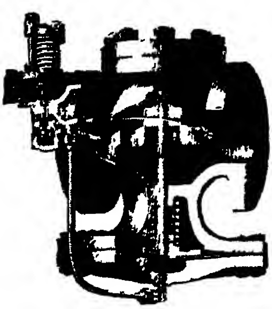
Spence Weather Compensator—Type EWM3T



This simple, dependable Control, when installed on a properly designed orificed heating system, will show a substantial degree-day steam saving, at a low maintenance cost

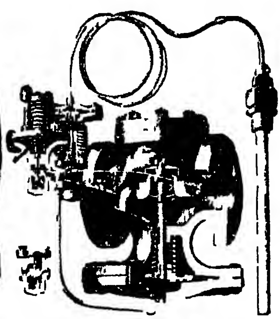
The delivery pressure of the Regulator is automatically adjusted in direct proportion to the building heat losses In other words, as the losses become greater, steam pressure on the system is automatically increased

Any number of zones can be controlled by one automatic Signatrol, automatic Wind Loss Compensator (Anemometer) Time Switch and Master Control Panel equipped with Manual and Automatic Dials for each zone In this way each zone can be set individually and at the same time be under the Master Control When a Manual Dial is used, the automatic controls are cut off from that zone without affecting other zones



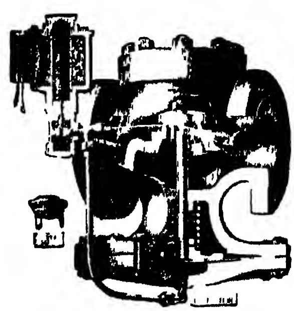
Pressure Regulator—Type ED

Designed to regulate a steady or varying initial pressure so as to maintain a constant, adjustable, delivery pressure Applicable to heating systems, power plant operations, or manufacturing processes



Combined Temperature and Pressure Regulator —Type ETD

Self-contained, pilot operated, dead-end Designed to control flow of fluid to a heating or cooling element, so as to maintain a constant, adjustable temperature, and protect the element against excessive pressure



Electrically Operated Valve—Type EM

Can be opened or closed independently by an electrical switch

Type ET—Same as ETD except pressure control is omitted

Order a SPENCE Regulator for 40 days' free trial.

Sylphon Control Systems, Inc.

(Subsidiary of Fulton Sylphon Company)

AUTOMATIC CONTROL SYSTEMS FOR HEATING, VENTILATING AND AIR CONDITIONING

New York, N. Y. Knoxville, Tenn.

Sales Representatives in Principal Cities

PRODUCTS AND SERVICE

Sylphon Control Systems, developed and manufactured by the Fulton Sylphon Co, and engineered, installed and serviced by Sylphon Control Systems, Inc , a nation-wide organization, offers architects and engineers self-actuated and electric systems and combinations thereof for complete control of all types of heating, ventilating and air conditioning installations These systems comprise

1. Self-contained Control Systems utilizing a complete line of Sylphon self-contained, self-powered instruments requiring no auxiliary power, motors, wiring, thermostats, relays or switches Co-ordinated to provide entirely automatic *modulating, non-cycling* control of any or all functions of air conditioning

2. Semi-Self-Contained and Electric Control Systems -which combine the desirable features of the self-contained control system with the advantages of electric operation, where electric thermostats, humidistats, program - clocks, motor - operated valves, dampers and manually operated remote station switches, etc , are desired

Outstanding features of these systems are

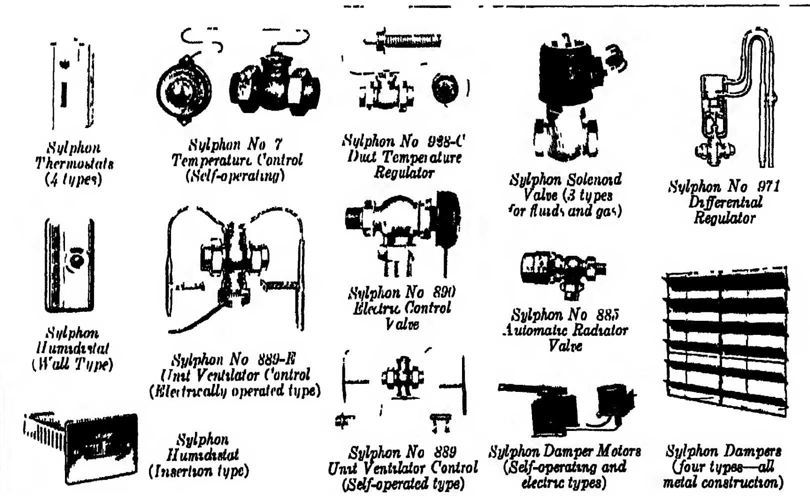
1. They provide *true modulation* by correct automatic proportioning of temperature, air-flow and humidity—with a great reduction in the complication of equipment necessary for the accomplishment of this ideal control

2. Simplicity, ruggedness, reliability—with no short-lived mechanisms to wear out prematurely or get out of order—these systems are a sound investment in years of trouble-free service

3. Interchangeability of instruments in Sylphon Control Systems—offers almost limitless combinations of self-contained and electric instruments, both positive and modulating—permits great flexibility in their application to solve each particular control problem

4. Moderate first cost—extremely low operating cost practically no maintenance

To illustrate the complete line of instruments developed for use in Sylphon Control Systems—a representative few are shown below



Sylphon Thermostats (4 types)

Sylphon No 7 Temperature Control (Self-operating)

Sylphon No 988-C Duct Temperature Regulator

Sylphon Solenoid Valve (3 types for fluids and gas)

Sylphon No 971 Differential Regulator

Sylphon Humidistat (Wall Type)

Sylphon No 889-E Unit Ventilator Control (Electrically operated type)

Sylphon No 890 Electric Control Valve

Sylphon No 885 Automatic Radiator Valve

Sylphon Humidistat (Insertion type)

Sylphon No 889 Unit Ventilator Control (Self-operated type)

Sylphon Damper Motors (Self-operating and electric types)

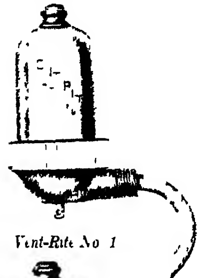
Sylphon Dampers (four types—all metal construction)

See also Page 1109

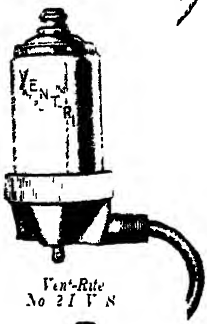
Anderson Products Incorporated

Cambridge, Massachusetts

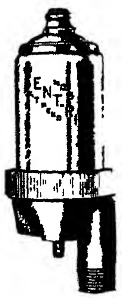
Manufacturers of VENT-RITE Radiator Air Valves—Originators of BALANCED RADIATION by Controlled Venting and the VENT-VAC System



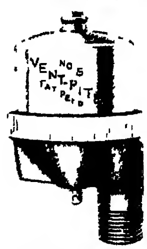
Vent-Rite No 1



Vent-Rite No 2 I V N



Vent-Rite No 3



Vent-Rite No 5

VENT-RITE RADIATOR AIR VALVES

A Complete Line of Eleven Vent-Rite Valves—Moderately Priced, four of which are illustrated—is available to meet every venting requirement Their Wide Venting Range and Tamper Proof Adjustment Improve the Operation of Any One-Pipe Steam Job

Vent-Rite Controlled Venting is Basically "Right"

The correct venting rate for each radiator in a system that will insure Positive Controlled Distribution of steam, CAN NOT BE predetermined It requires actual regulation after installation to "Balance" that particular system on which the installation is made

The venting rate of Vent-Rite Valves therefore, is not limited but expressly provides as many as 540 venting rates by means of a modulating valve construction This venting range, from entirely closed to the full open position of a 1/16 in diameter orifice, is ample to Balance and Control the distribution of steam to any number of Radiators

Thus, Vent-Rite Valves, set by the Heating Contractor, make Controlled Distribution a practical feature regardless of the Number of radiators, their Sizes, Distances from the boiler, or Pressure Carried.

Types and Construction

Vent-Rite Valves are made in both vacuum and non-vacuum types The construction of the vacuum types does not depend upon a ball or disc check to maintain vacuum This is accomplished by a sensitive and positive acting bellows, internally subjected at all times to atmospheric pressure which elongates the bellows, and closes the vent when the internal pressure in the valve is less than the atmospheric pressure, thereby perfecting an inner vacuum seal

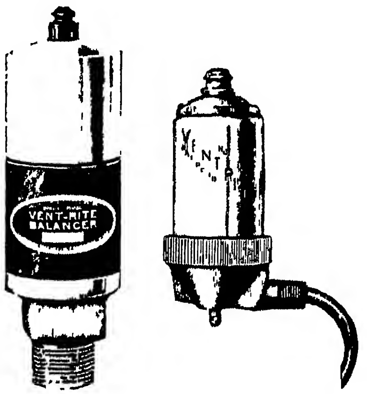
Vent-Rite Valves are Take-apart Construction—Noiseless in Operation—Positive in Action—Seal by Float Action against Water—Will not Leak or Sputter—Close Thermostatically under Temperature—have Tamper-proof Adjustment—Wide Range Venting Control

Vent-Rite Valves are made of fine non-rusting—non-corroding materials to insure years of trouble-free service Bases are Brass Drop Forgings, Valve Pins are Nickel Silver, Finish is Chromium Plate (except Convector Valves, which are Nickel Plated)

Vent-Rite Air Valves and the Vent-Rite Balancer Reach Consumers Through Plumbing and Heating Contractors Only. They are available to the trade Through Selected Recognized Wholesalers Only.

Anderson Products Incorporated

Cambridge, Massachusetts



THE VENT-VAC SYSTEM

The **Vent-Vac System** operates as an Open Atmospheric and Vacuum System combined, retaining only the desirable, practical features of both systems

The system is established on one-pipe steam jobs by installing **the Vent-Rite Balancer, to operate in conjunction with No. 2 Vent-Rite Vacuum Valves.** The Vent-Rite Balancer is a solenoid operated valve which automatically returns a system under vacuum to atmospheric pressure at the beginning of every firing cycle

The Vent Vac System Positively Insures

1. The controlled distribution of steam as it is generated, so that **All Radiators are Heated at the Same Time on Every Firing Cycle.**
2. A further controlled Btu distribution, under vacuum **To All Radiators in Proportion to Their Respective Heat Losses.**

The Vent-Rite Balancer is $6^3/_8$ in in height and $1^{15}/_{16}$ in in diameter It functions Automatically at all times to meet instantly the requirements of Venting, Intake, Thermostatic Closing and Float Seal Action

With the system under vacuum and the room thermostat demanding heat, the firing unit automatically starts and, at the same instant, the Balancer Solenoid is energized, opening the venting orifice of the Balancer to intake air (later to expel air as steam is generated)

The distribution of steam, as it is generated, is so controlled by the use of Vent-Rite Valves that all radiators are heated at the same time on every firing cycle When indicated room temperature is reached, the thermostat shuts off the burner

After the firing unit has stopped, a further controlled Btu distribution is provided by vapor, under vacuum, to all radiators in proportion to their respective heat losses

When the vapor cools to the point where room temperature is no longer satisfied, the thermostat again calls for heat, the firing unit is started, the Solenoid is energized, the Balancer returns the system to atmospheric pressure, and the Vent-Rite Valves return to their original set positions Thus the cycle is completed and controlled distribution on the next cycle is assured

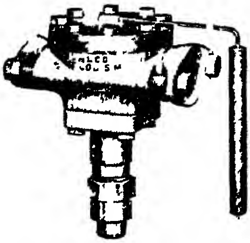
The Vent-Rite Balancer is installed on the boiler or on one of the mains It is electrically connected with the automatic heating unit and operates automatically in conjunction with the heat control or thermostat

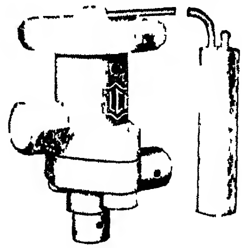
The Vent-Vac System has so improved the operating efficiency of One-Pipe Steam Systems that Architects and Engineers can now specify One-Pipe Steam Jobs for larger residences, moderate size factories, hospitals, schools, etc with confidence that the system will provide adequate, and economical heat

Alco Valve Company, Inc.

AUTOMATIC REFRIGERANT CONTROL VALVES

2626 Big Bend Blvd **St. Louis, Mo.**





THERMO EXPANSION VALVES—They assure positive independent, automatic control of liquid feed in each refrigerated unit of a multiple system

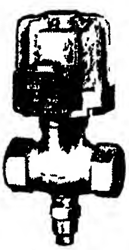
They automatically regulate liquid feed and suction pressure in accord with the refrigerating load in any unit thereby materially increasing the efficiency of the compressor as well as the evaporating coils

They provide a positive method of obtaining a maximum wet interior surface in any type of evaporator

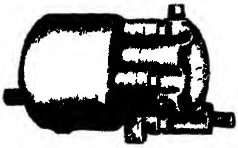
They are sensitive to the slightest change in suction superheat, but are adjustable to control the liquid flow at any required degree of suction superheat without permitting the return of liquid refrigerant to the compressor

There is a size and type for all refrigerants and all capacities from fractional tonnage to 60 tons Ammonia, 100 tons Methyl Chloride, or 50 tons Freon

MAGNETIC LIQUID STOP VALVES—are positive acting, and tight closing They are indispensable wherever instantaneous closing of the liquid supply line is indicated They are used extensively with expansion valves where the temperature difference between the refrigerant and the refrigerated substance or area is very small They are also used extensively with Alco Float Switches to maintain a constant liquid level in flooded evaporators

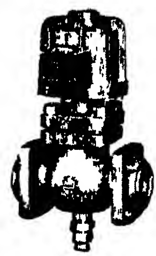


All types are available in all the ordinary pipe sizes up to 1¼ in, and for tonnage capacities ranging from fractional tonnage to 350 tons Ammonia, 115 tons Methyl Chloride, or 55 tons Freon



LIQUID FLOAT VALVES—are provided with a patented vent tube inside the body which prevents gas binding, and permits the valve to be installed at the highest point on a full flooded system even though many feet above the liquid receiver They may also be installed at a low point in the system and will perform their function equally as well Available in a variety of capacities up to 25 tons Ammonia, 10 tons Methyl Chloride, or 5 tons Freon

MAGNETIC SUCTION STOP VALVES—are designed for use as suction line shut-off or as low side by-pass valves They are built without packing so as to operate successfully on heavily frosted lines Makes possible individual control of two or more evaporating units in a multiple system even though there is a wide difference in temperature requirement or load conditions They provide individual temperature control in any number of refrigerated units in a series system either flooded or fed by a constant pressure expansion valve



Built in ½ in, ¾ in, 1 in, 1¼ in, 1½ in, or 2 in, sizes

ELECTRIC FLOAT SWITCHES are highly efficient for the maintaining of a constant liquid level in individual flooded evaporators Particularly desirable for use on tubular coolers There is no liquid flow through the Float Switch By means of its electric switch, it operates a magnetic valve in the liquid line, thereby maintaining the desired liquid level to within a limit of less than one inch

The Dole Valve Company

Main Office and Factory 1901-1941 Carroll Avenue, **Chicago, Ill.**

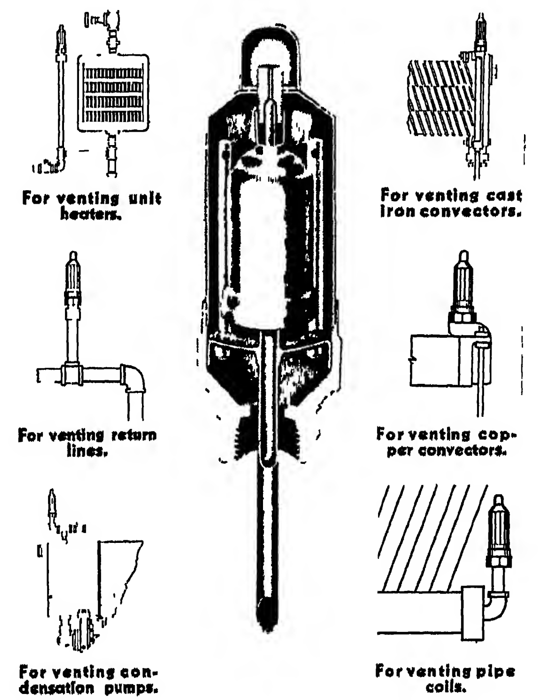
Branch Offices and Representatives in All Principal Cities

Dole

The Dole Valve Company is a progressive manufacturer of air and vacuum valves for one or two pipe steam and hot water heating systems

There is a Dole Air and Vacuum Valve for every purpose The line is complete and it is kept up to date by aggressive designing, engineering and manufacturing policies

Our latest contributions are the new DOLE NO 3A ($^1/_8$ in connection), NO 3B ($^1/_4$ in connection) and NO 3C ($^3/_4$ in male, $^3/_8$ in female connections) AIR VALVES Here is a picture of the No 3C Air Valve



For venting unit heaters.

For venting cast iron convectors.

For venting return lines.

For venting copper convectors.

For venting condensation pumps.

For venting pipe coils.

These Dole Nos 3A, 3B and 3C Air Valves are designed to meet the increasing demand for popularly priced straight shank valves that give satisfactory venting performance on systems where extreme water conditions are encountered They positively do not "stick" or spit water and they insure free venting at any pressure up to 15 lb

Look at the illustrated installation suggestions Notice that a nipple connection is not required when Dole No 3 Air Valves are installed on either cast iron or copper convectors, provided the heater has sufficient space to accommodate the siphon tube Do not cut or bend this tube

Another important addition which rounds out the Dole line is the DOLE NO 7 SIGNAL AIR VALVE for venting concealed hot water convectors or radiators This valve actually signals the operator when all the air is expelled It is very easy to operate (no holding a cup to catch the water overflow) and it is easy to install

Dole Air and Vacuum Valves for other venting purposes on one or two pipe steam and hot water heating systems are the No 2B Vacuum Valve, No 6B Vacuum Valve, No 100 Vacuum Valve, No 103 Vacuum Valve, No 2 Vacuum Valve, No 6 Vacuum Valve, No 1 Air Valve, No 4 Quick Vent Valve, No 5 Quick Vent Float Valve, No 8 Quick Vent Valve, No 9 Quick Vent Vacuum Valve, No 1933 Air Valve, No 10 Hot Water Key Valve and the Dole Compound Gauge

Dole Air and Vacuum Valves that are equipped with the new Vari-Vent feature for "balancing" one pipe steam heating systems are the No 2B Vari-Vent Vacuum Valve, No 100 Vari-Vent Vacuum Valve, No 101 Vari-Vent Vacuum Valve, No 1 Vari-Vent Air Valve and No 3A Vari-Vent Air Valve

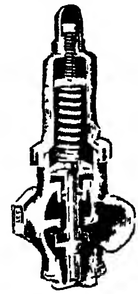
We have prepared folders and illustrated price sheets on the whole Dole Air and Vacuum Valve line These are not only informative but make splendid sales material. They are yours for the asking

Foster Engineering Co.

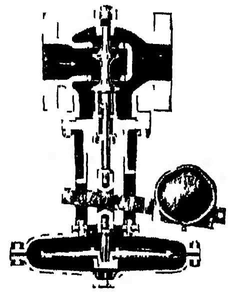
109-113 Monroe Street **Newark, N. J.**

Fifty-seven Years Experience in Designing, Manufacturing and Applying Automatic Valves to meet all Classes of Control Service on Steam, Liquids and Gases. Standard and Special Construction

Class 34G Pressure Regulator



Type 34U Pressure Regulator



Class 35B2 Pressure Regulator

Class 34G Pressure Regulator—Single bevel-seated (strainer protected), pilot operated, for single-stage reduction on steam or air for initial pressures up to 900 lbs, 750°F T T. Maintains a constant delivery regardless of fluctuations of initial pressure or volume of flow and is adapted for deadend service Type 34G8 constructed with large diaphragms for delivery pressure 0 to 15 lbs Sizes ½ in to 12 in Bronze, semi and cast-steel

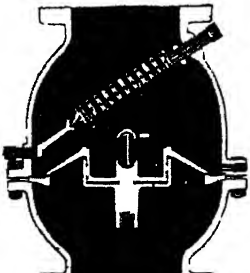
Type 34U Pressure Regulator—Direct Acting Piston Operated for variable initial pressures up to 300 lbs, 160°F T T on water or air, adapted for deadend service Special construction to meet higher pressure Several types of pilot or direct-acting diaphragm actuated for low delivery pressure and intermittent service All types have renewable seat discs Sizes ½ in to 12 in Bronze and semi-steel

Class 35B2 Pressure Regulator—For service on steam in apartments, office buildings, etc Initial pressures up to 250 lbs and constant delivery pressures 0 to 50 lbs Direct acting, double-seated type (not adapted for deadend service), for variable initial pressures and volume Sizes ½ in to 10 in Bronze, semi or cast-steel body

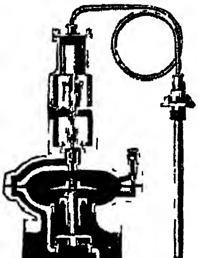
Back Pressure Valve—Type illustrated shows unique yet simple construction with compensating spring and lever movement that maintains constant relation between spring resistance and back pressure A large internal dashpot cushions the valve and eliminates chattering Spring range is 0 to 15 lbs Sizes 2½ in to 20 in Other types such as lever and weight construction available

Temperature Regulator—Pilot operated, self-contained, single-seated regulator for deadend service. Maintains temperatures of gases or liquids within 1°F ± Pressures 10 to 200 lbs Sizes ½ in to 3 in Larger sizes to 8 in., double-seated Type 34T2, direct-acting, double-seated (not adapted for deadend service) for 0 to 125 lbs Sizes ½ in to 6 in Bronze and semi-steel body

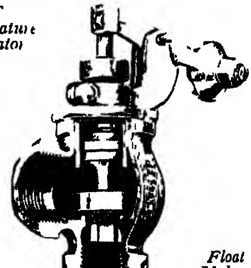
Float Valves—Very sensitive valve for hot or cold water service, being actuated by the pressure in the supply line Sizes range from ½ in. to 8 in Angle and Globe Several types made with piston and stem loosely coupled, adding direct-acting feature Bronze and semi-steel body



Back Pressure Valve



34T Temperature Regulator



Float Valves

Jenkins Bros

BRONZE - IRON - STEEL VALVES - SINCE 1864

Mechanical Rubber Goods

Principal Stores and Offices 80 White Street, NEW YORK, N Y, 524 Atlantic Avenue, BOSTON, MASS, 133 N Seventh Street, PHILADELPHIA, PA, 822 Washington Boulevard, CHICAGO, ILL, 510 Main Street, BRIDGEPORT, CONN, (Office and Factory) JENKINS BROS, LIMITED MONTREAL, QUE, (Works, Head Office); LONDON, W C 2

Fig 106
A Bronze Globe
Renewable Comp Disc

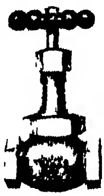


Fig 950
Bronze Globe,
Regrind-Renew

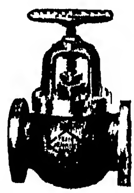


Fig 142
Iron Body Globe

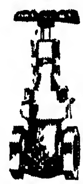


Fig 370
Bronze Gate

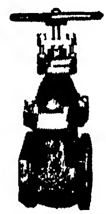


Fig 326
Iron Body Gate

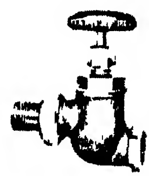


Fig 859
Radiator Offset Globe

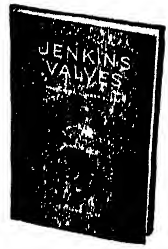
OVER 500 DIFFERENT JENKINS VALVES COVER EVERY HEATING AND AIR CONDITIONING NEED

To adequately describe the complete Jenkins line of valves requires a Catalog of more than 300 pages There are over 500 different types and patterns of valves that bear the trusted "Diamond" trade mark Practically speaking, Jenkins can furnish any valve that you may require for plumbing, heating, air conditioning, general industrial or engineering service

General Classifications of Jenkins Valves Include Bronze Valves fitted with Jenkins renewable composition disc Bronze Regrind-Renew Valves with bevel and plug type seats Bronze Gate Valves Iron Body Valves fitted with Jenkins renewable composition disc Iron Body Regrinding Valves Iron Body Gate Valves with solid wedge and double disc parallel seats All-Iron Valves Cast Steel Gate Valves and Swing Check Valves Electrically and Hydraulically Operated Valves Radiator Valves Fire Line Valves Quick-opening and Self-closing Valves, Needle Valves, Y Valves, Solder-End Valves

Other Jenkins Products Are— Colored Valve Wheels with or without service markings molded in relief letters. Composition Valve Discs exactly suited to service conditions Sheet Packing Gaskets Moncrieff Scotch Gage Glasses.

CONSULT THIS HELPFUL BOOK



This 307 page Jenkins Catalog not only gives complete details on over 500 Jenkins Valves, but also it has a large section of engineering data and practical information about valves and layouts Make sure you have a copy, including the new Supplement "B"

JENKINS VALVES ARE SOLD BY MOST GOOD SUPPLY HOUSES

New York Air Valve Corporation

Since 1898

Nyavco — CHICAGO, BOSTON — DETROIT, ST LOUIS — CLEVELAND, PITTSBURGH — Nyavco

Orifice Control Air Valves — 611-621 Broadway, New York — **Vacuum Orifice Control Air Valves**

NYAVCO ORIFICE "CONTROL BY VENTING" VALVES

Air which fills each unheated radiator must be driven out by entering steam before heat is obtained Using this air as an "air brake" and thus limiting or increasing its evacuation time on each radiator, accordingly slows or increases speed of entering steam and its consequent heating time

The NYAVCO Orifice Control Air Valve incorporates in one valve six gradually ascending vent speeds, which make possible the simultaneous heating of the largest and smallest radiator—or the equalization in steam delivery of the farthest unit from or nearest unit to the boiler Thus balancing the job

The NYAVCO method of metered venting can so "time" each room radiator large or small—near or far—as to heat simultaneously, on coal fired constant heat jobs, or with room containing the automatic control (if an automatic gas, oil or stoker fired job)

NYAVCO venting is fast because of unusually large vent port It can only be set by the heating man—is consequently tamper-proof—because it is "locked in" the "armored cap" by him Has a definite metered disc—Does not depend on thread or needle valve, but actual room or distance equalizing on each setting

Nyavco ORIFICE CONTROL

Air Valve—For one or two pipe gravity steam jobs —Note open cut showing indestructible construction and control disc

Made also in $\frac{1}{8}$ in, $\frac{1}{4}$ in, $\frac{1}{2}$ in, Straight and $\frac{3}{4}$ in Quick Vent

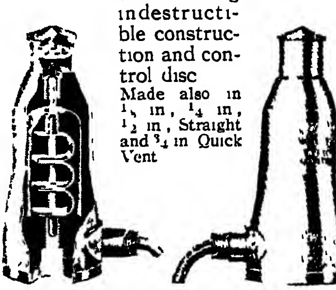


Fig No 1

Nyavco ORIFICE CONTROL

Vacu-Seal Vacuum—A ball check type vacuum valve—for use on intermittent heating jobs where price is a consideration

Made also in $\frac{1}{8}$ in, $\frac{1}{4}$ in, $\frac{1}{2}$ in, Straight Shank and $\frac{1}{4}$ in Quick Vent

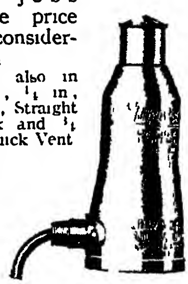


Fig No 2

Nyavco ORIFICE CONTROL

Gold-Seal-Lock-Vacuum The Bellows operated atmospheric pressure locked Vacuum Valve, will maintain vacuum on one pipe gravity steam jobs over very long periods

Made in angle and quick vent types only

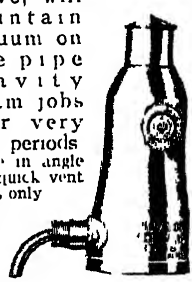


Fig No 20

OPERATION OF VALVE

INDESTRUCTIBILITY

NYAVCO because of its open float and bi-metallic mechanism, depending upon actual steam contact functions immediately and will operate efficiently under all service requirements of one pipe systems

NYAVCO factory adjustment cannot be injured and valve is guaranteed fatigue-proof — rust-proof — shock-proof Write for details

GIANT Three Speed Air ELIMINATOR

For Rapid Air Elimination from Large Mains — Coils — Air Conditioning Units —Unit Heaters, etc.

3 Speed control—$\frac{1}{8}$ in, $\frac{3}{16}$ in, $\frac{1}{4}$ in—Secured by removing screw from speed size desired

$\frac{1}{8}$ in Size for load up to 1500 ft

$\frac{3}{16}$ in Size for load of 1500 ft to 3000 ft

$\frac{1}{4}$ in Size for load of 3000 ft up

Made in regular Venting and Vacuum Valve 6¼ Actual Height

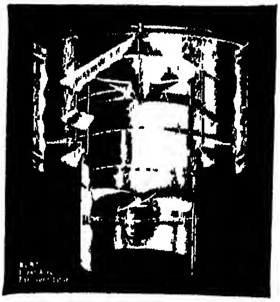
The Burt Mfg. Co.

VENTILATORS - OIL FILTERS - EXHAUST HEADS

AKRON, OHIO, U. S. A.

Representatives in All Principal Cities

Burt "Forst-Aire" Fan Ventilators



Dimensions - Weights — Capacities

Size Inches	Ga Galv	Oz Copper	Net Wt Lbs	Overall Dia Inches	Overall Height In -*	Motor HP	Motor RPM	Free Air Capacity CFM
12	22	16	82	21	31¼	1/10	1725	690
14	22	16	99	25½	35	1/10	1725	1200
16	22	16	104	27½	36	1/8	1725	1850
18	20	18	125	30	38½	1/8	1725	2440
20	20	18	151	34½	43	1/6	1140	2990
24	20	18	170	39	50	1/6	1140	4300
30	18	20	347	52½	58	1/3	860	6350
36	18	24	483	60	64	1/2	860	10600
42	18	24	625	69	72	3/4	675	11130
48	18	24	798	78½	80	3/4	675	14500

*Exclusive of base

Head High Efficiency, Low Resistance
Motor Mounting Rubber Cradled Secure
Types Standard Heat Resisting Acid Resisting Static Pressure

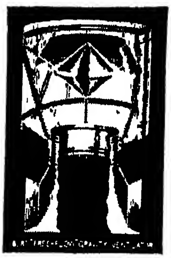
Motors—Totally Enclosed—Fan Duty --Vertical, Ball Bearing—Class 1, Group D optional, Single or Multi-Speed, 110 V, 1 Ph, Standard on 1/10 and 1/6 H P —220 V, 3 Ph on larger sizes

Burt Free-Flow Gravity Ventilator

In the Burt "Free-Flow" Gravity Ventilator, traditional design was disregarded where it was found not to produce desired results Notice that the entire discharge from the ventilator is vertically upward Note also that absence of any internal louvres which would tend to impede the flow of air through the unit

The "Free-Flow" Gravity Ventilator meets a requirement which has long been existent for a well made, reasonably priced super-capacity stationary ventilator Standard material is prime open hearth galvanized steel sheets Optional material if desired

We guarantee this ventilator to have a capacity equal to or greater than any Stationary Ventilator on the market



Dimensions Weights Capacities

Vent Size	Material Ga Galv	Material Oz Copper	Air Shaft Diam In	Wind Band In Width Diam	Wind Band In Height	O A Ht In	Drain Size	Net Wt Galv	Net Wt Copper
8	24	16	8	14½	12	16½	1½	13	11
10	24	16	10	18	14	18½	1½	17	15
12	24	16	12	22	16	25	1½	25	22
14	24	16	14	25½	18	27	1½	31	27
16	24	16	16	28½	21	30	1½	38	33
18	24	18	18	32½	24	36	2	50	48
20	24	18	20	36	27	39	2	63	61
24	24*	18*	24	43½	31	43	2	97	92
30	22*	20*	30	54	36	51	2	153	134

Vent Size	Material Ga Galv	Material Oz Copper	Air Shaft Diam In	Wind Band In Width Diam	Wind Band In Height	O A Height In	Drain Size	Net Wt Galv	Net Wt Copper
36	22	22	36	64½	44	59	2½	237	231
42	20	22*	42	75	53	68	2½	352	300
48	20	24*	48	85½	60	75	2½	460	430
54	20	28*	54	96½	68	86	3	582	620
60	18	28*	60	108	76	94	3	902	778
66	18	28*	66	119	84	102	3	1113	970
72	18	30*	72	130	92	110	4	1352	1238
84	18	32*	84	150	104	125	4	1766	1171
96	18	32*	96	171	120	141	4	2284	2196

Airshaft (*) 2 Gauges (**) 4 Gauges heavier than figures shown

OTHER BURT VENTILATORS

Sliding Sleeve Damper Unit.
Ball Bearing Revolving Ventilator.
Monovent Continuous Ridge Ventilator.
Burt High Efficiency Cone Damper Ventilator.

BURT ROUND LOUVRE DAMPERS

Can be fitted into any Burt Ventilator where an extremely close fitting damper is needed Edges felted if required Easy operating in either hand or damper motor control

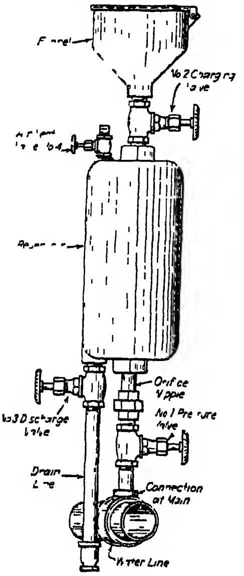
Write for Burt Handbook of Ventilation Data

Ferro-Nil Corporation

500 Fifth Avenue — New York City

WATER CONDITIONING FOR AIR CONDITIONING

FERRO-NIL SERVICE IS USED FOR PREVENTION OF CORROSION



FERRO-NIL INJECTOR FOR AIR CONDITIONING EQUIPMENT

Operation

Close pressure valve No 1, open charging valve No 2, and air vent valve No 4

Now open discharge valve No 3

After contents of injector have been entirely drained out, flush injector by opening pressure valve No 1 for a short time, permitting water to flush out through discharge valve No 3

After closing pressure valve No 1, close discharge valve No 3

Pour into the funnel the required amount of **Ferro-Nil** through charging valve No 2 Flush funnel and charging valve No 2 with 1 pint of water after **Ferro-Nil** has passed into reservoir

Now close charging valve No 2 and air vent valve No 4 Open pressure valve No 1

Pour 1 pint of water into funnel to insure easy reopening of charging valve No 2

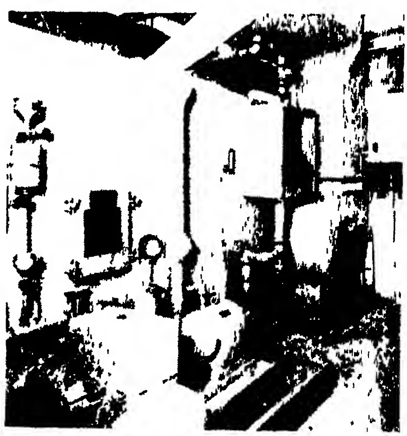
Injector is now in operation

Close valve No 1 when water circulating pump is shut down.

Open valve No 1 at time water circulating pump is started.

All water used for washing air picks up corrosive matter from the air The corrosive matter may be oxygen, carbon dioxide, sulphur trioxide or sulphur dioxide In air conditioning systems where water is recirculated, the amount of acidic material contained in the water may reach very high levels This acid water will corrode metallic surfaces with which it comes in contact In severe cases equipment has been known to fail after three months' operation due to corrosive action of water used **Engineers, Architects and Contractors can avoid responsibility for failure of equipment due to corrosion by specifying Ferro-Nil service.**

Ferro-Nil Service by maintaining water in a non-corrosive condition, prevents corrosion Corrosion causes high rates of depreciation Ferro-Nil Service by maintaining metallic surfaces free from rust and scale affords optimum heat transfer efficiencies at all times



Typical installation of Ferro-Nil chemical feeder on air washer in transformer room—in this particular installation (11) eleven air washers are so equipped

Let us submit estimate on cost of Ferro-Nil Service for your air conditioning equipment

INDEX TO MODERN EQUIPMENT

AMERICAN SOCIETY *of* HEATING *and* VENTILATING ENGINEERS GUIDE, 1937

Numerals following Manufacturers' Names refer to pages in the Catalog Data Section

Numerals following Manufacturers' Names refer to pages in the Catalog Data Section

Numerals following Manufacturers' Names refer to pages in the Catalog Data Section

Lennox Furnace Co, Inc, 864-865
Meyer Furnace Company, The, 866
L J Mueller Furnace Co, 870-871
Trane Company, The, 1094
L J Wing Mfg Co, 954-955

FANS, Portable

Air Devices Corporation, 818-819
American Blower Corporation, 820-821
American Coolair Corp, 942-943
Autovent Fan & Blower Co, 945
Barber-Colman Company, 1104-1105
Bayley Blower Company, 946
Binks Manufacturing Co, 938-939
Buffalo Forge Company, 947
E K Campbell Heating Co, 969
Canadian Sirocco, Ltd, 820-821
Champion Blower & Forge Co, 948
Coppus Engineering Corporation, 886
General Electric Company, 860-861, 1026-1027
Henry Furnace & Foundry Co, 862-863
Ilg Electric Ventilating Company, 949
Westinghouse Electric & Manufacturing Co, 840
L J Wing Mfg Co, 954-955

FANS, Propeller

Air Controls, Inc, 842
Air Devices Corporation, 818-819
American Blower Corporation, 820-821
American Coolair Corp, 942-943
Autovent Fan & Blower Co, 945
Binks Manufacturing Co, 938-939
Buffalo Forge Company, 947
E K Campbell Heating Co, 969
Canadian Sirocco Co, Ltd, 820-821
Champion Blower & Forge Co, 948
Clarage Fan Company, 825
Coppus Engineering Corporation, 886
De Bothezat Division American Machine & Metals, Inc, 944
General Electric Company, 860-861, 1026-1027
Henry Furnace & Foundry Co, 862-863
Ilg Electric Ventilating Company, 949
Marley Co, The, 937
Servel, Inc, 836
B F Sturtevent Co, 950
Thermal Units Manufacturing Company, 837
Torrington Mfg Co, The, 951-953
Trane Company, The, 1094
Westinghouse Electric & Manufacturing Co, 840
L J Wing Mfg Co, 954-955

FANS, Supply and Exhaust

Air Controls, Inc, 842
Air Devices Corporation, 818-819
American Blower Corporation, 820-821
American Coolair Corp, 942-943
Autovent Fan & Blower Co, 945
Bayley Blower Company, 946
Binks Manufacturing Co, 938-939
Buckeye Blower Company, 972-973
Buffalo Forge Company, 947
E K Campbell Heating Co, 969
Canadian Sirocco Co, Ltd, 820-821
Champion Blower & Forge Co, 948
Clarage Fan Company, 825
Coppus Engineering Corporation, 886
De Bothezat Division American Machine & Metals, Inc, 944
Delco-Frigidaire Conditioning Division, General Motors Sales Corporation, 852-854
General Electric Company, 860-861, 1026-1027
Henry Furnace & Foundry Co, 862-863
Ilg Electric Ventilating Company, 949
Meyer Furnace Company, The, 866
L J Mueller Furnace Co, 870-871
John J Nesbitt, Inc, 972-973
Niagara Blower Company, 832-833
B F Sturtevant Co, 950
Thermal Units Manufacturing Company, 837
Trane Company, The, 1094
Westinghouse Electric & Manufacturing Co, 840
L J Wing Mfg Co, 954-955

FEED WATER HEATERS (*See Heaters, Feed Water*)

FEED WATER REGULATORS (*See Regulators, Feed Water*)

FEEDERS, Boiler

Crane Co, 904-905
General Controls, 1112
Kieley & Mueller, Inc, 1084
McDonnell & Miller, 894-895
John H McGowan Co, The, 1058
Milwaukee Valve Co, 1086-1087
Mueller Steam Specialty Co, Inc 1088
Spence Engineering Co, 1122
H A Thrush & Co, 980-981
Warren Webster & Co, (Safety) 1095-1097
Westinghouse Electric & Manufacturing Co, 840
Wright-Austin Co, 1098

FEEDERS, Water

Decatur Pump Co, 1053
General Controls, 1112
Kieley & Mueller, Inc, 1084
McDonnell & Miller, 894-895
Milwaukee Valve Co, 1086-1087
Mueller Steam Specialty Co, Inc, 1088
H A Thrush & Co, 980-981
Wright-Austin Co, 1098

FILTERS, Air (*See Air Cleaning Equipment*)

FITTINGS, Pipe, Flanged

American Rolling Mill Co, The, 1067
Baker Ice Machine Co, Inc, 822
Chase Brass & Copper Co, Inc, 1030-1031
Crane Co, 904-905
Frick Company (Incorporated), 827
Grinnell Company, Inc, 966-968, 1078
Arthur Harris & Co, 1033
Vilter Manufacturing Co, The, 839
York Ice Machinery Corporation 841

FITTINGS, Pipe, Screwed

American Radiator Co, 843, 898-901, 974
Baker Ice Machine Co, Inc, 822
Chase Brass & Copper Co, Inc, 1030-1031
Crane Co, 904-905
Frick Company (Incorporated), 827
Grinnell Company, Inc, 966-968, 1078
Taco Heaters, Inc, 978-979
Vilter Manufacturing Co, The, 839
York Ice Machinery Corporation, 841

FITTINGS, Pipe, Solder

American Brass Co, The, 1028-1029
Mueller Brass Co, 1034-1035
Revere Copper & Brass Inc, 1038

FITTINGS, Pipe, Sweat

American Brass Co, The, 1028-1029
American Radiator Co, 843, 898-901, 971
Chase Brass & Copper Co, Incorporated, 1030-1031
Crane Co, 901-905
Detroit Lubricator Co 1107
Mueller Brass Co, 1034-1035
Revere Copper & Brass Inc, 1038
Streamline Pipe and Fittings Co, 1034-1035
Taco Heaters, Inc, 978-979

FITTINGS, Welding

American Rolling Mill Co, The, 1067
Crane Co, 901-905
Grinnell Co, Inc, 966-968, 1078
York Ice Machinery Corporation, 841

FLOATS, Metal (*See Trap and Valve*)

Arthur Harris & Co, 1033
Spence Engineering Co, 1122
Wright-Austin Co, 1098

FLOOR AND CEILING PLATES

American Radiator Co, 843, 898-901, 971
Beaton & Cadwell Mfg Company, The, 1072-1073
Crane Co, 901-905
Grinnell Company, Inc, 966-968, 1078

FLUE GAS ANALYSIS

Brown Instrument Company, 983
Leeds & Northrup Company, 987
Minneapolis-Honeywell Regulator Company, 1111-1115

FURNACE-BURNER

Anchor Stove & Range Co, Inc, The, 1100
Delco-Frigidaire Conditioning Division, General Motors Sales Corporation, 852-854
Electrol Incorporated, 846-847, 927
Fox Furnace Co, The, 848-851
Gar Wood Industries, Inc, 858-859
Gilbert & Barker Mfg Co, 855-857
Henry Furnace & Foundry Co, 862-863
Lennox Furnace Co, Inc, 864-865
Meyer Furnace Company, The, 866
L J Mueller Furnace Co, 870-871
Herman Nelson Corp, The, 970-971
Norge Division, Borg-Warner Corporation, 873, 932
Williams Oil-O-Matic Heating Corporation, 876-877

FURNACE HEATING SYSTEMS (*See Heating Systems, Furnace*)

FURNACE REGULATORS (*See Regulators, Furnace*)

FURNACES, Electric

General Electric Company, 860-861, 1026-1027
Westinghouse Electric & Manufacturing Co, 840

FURNACES, Warm Air

Airtherm Manufacturing Co, 956
American Gas Products Corp, 897

Numerals following Manufacturers' Names refer to pages in the Catalog Data Section

GAS HEATERS (*See Heaters, Gas*)

GASKETS, Asbestos

Crane Co, 904-905
Ehret Magnesia Manufacturing Co, 1000-1001
Frick Company, Inc, 827
Jenkins Bros, 1129
Johns-Manville, 1006-1007
Ruberoid Co, The, 1012-1013

GASKETS, Cork

Armstrong Cork Products Company, 994-995
Crane Co, 904-905
Johns-Manville, 1006-1007
Mundet Cork Corp, 1008

GASKETS, Rubber

Crane Co, 904-905
Ehret Magnesia Manufacturing Co, 1000-1001
Frick Company, 827
Jenkins Bros, 1129
Johns-Manville, 1006-1007

GOVERNORS, Pump

Crane Co, 904-905
Davis Regulator Co, 1075
C A Dunham Company, 1076-1077
Foster Engineering Co, 1128
Illinois Engineering Company, 1082-1083
Kieley & Mueller, Inc, 1084
Mueller Steam Specialty Co, Inc, 1088
Spence Engineering Co, 1122
Warren Webster & Company, 1095-1097
Wright-Austin Co, 1098

GRATES FOR BOILERS AND FURNACES

American Radiator Company, 843, 898-901, 974
Combustion Engineering Company, Inc, 916
E Keeler Company, 924
Lookout Boiler & Mfg Company, 922-923
L J Mueller Furnace Co, 870-871
Unit Heater and Cooler Co, The, 960

GRILLES, REGISTERS AND ORNAMENTAL METAL WORK

American Blower Corporation, 820-821
American Coolair Corp, 942-943
American Radiator Company, 843, 898-901, 974
Anemostat Corporation of America, 1059
Auer Register Co, The, 1060
Barber-Colman Company, 1104-1105
E K Campbell Heating Co, 969
Canadian Sirocco Co, Ltd, 820-821
Carrier Corporation, 824
Hart & Cooley Manufacturing Co, 1062-1063
Independent Register Co, The, 1061
L J Mueller Furnace Co, 870-871
Trane Company, The, 1094
Tuttle & Bailey, Inc, 1064-1065
Waterloo Register Company, The, 1066

HANGERS, Pipe

American Brass Company, The, 1028-1029
American Radiator Company, 843, 898-901, 974
Baker Ice Machine Co, Inc, 822
Beaton and Cadwell Mfg Company, The, 1072-1073
Chase Brass & Copper Co, Incorporated, 1030-1031
Crane Co, 904-905
Frick Company (Incorporated), 827
Grinnell Company, Inc, 966-968, 1078
Ric-wiL Company, The, 1021
Vilter Manufacturing Company, The, 839

HANGERS, Radiator

American Radiator Company, 843, 898-901, 974
Burnham Boiler Corporation, 902-903
Crane Co, 904-905
Grinnell Company, Inc, 966-968, 1078
National Radiator Corporation, 906-908

HEADS, Exhaust

Burt Mfg Co, The, 1131
Crane Co, 904-905
Illinois Engineering Company, 1082-1083
Kieley & Mueller, Inc, 1084
Wright-Austin Co, 1098

HEADS, Sprinkler

Grinnell Company, Inc, 966-968, 1078
Mueller Brass Co, 1034-1035
Streamline Pipe & Fittings Co, 1034-1035

HEAT CABINETS (*See Heaters, Cabinet*)

HEAT SURFACE, Fan System

Aerofin Corporation, 962-964
Air Devices Corporation, 818-819
American Blower Corporation, 820-821
Buckeye Blower Company, 972-973
Buffalo Forge Company, 947
E K Campbell Heating Co, 969
Canadian Sirocco Co, Ltd, 820-821
Fairbanks Morse & Co, 828-829
Fedders Manufacturing Co, 958
G & O Manufacturing Company, The, 965
General Electric Company, 860-861, 1026-1027
McQuay, Incorporated, 834
Modine Manufacturing Co, 959
John J Nesbitt, Inc, 972-973
Niagara Blower Company, 832-833
B F Sturtevant Co, 950
Thermal Units Manufacturing Company, 837
Trane Company, The, 1094
Unit Heater & Cooler Co, The, 960
Westinghouse Electric & Manufacturing Co, 840
York Ice Machinery Corporation, 841
Young Radiator Company, 961

HEATERS, Air

Aerofin Corporation, 962-964
Air Devices Corporation, 818-819
Airtherm Manufacturing Co, 956
American Blower Corporation, 820-821
American Gas Products Corp, 897
American Radiator Company, 843, 898-901, 974
Autovent Fan & Blower Co, 945
Baker Ice Machine Co, Inc, 822
Buckeye Blower Company, 972-973
Buffalo Forge Company, 947
Burnham Boiler Corporation, 902-903
E K Campbell Heating Co, 969
Canadian Sirocco Co, Ltd, 820-821
Carrier Corporation, 824
Clarage Fan Company, 825
Combustion Engineering Company, Inc, 916
Delco-Frigidaire Conditioning Division, General Motors Sales Corporation, 852-854
Electric Air Heater Co, Div of American Foundry & Equipment Co, 957
Electrol, Incorporated, 816-817, 927
Fairbanks Morse & Co, 828-829
Fedders Manufacturing Co, 958
Gar Wood Industries, Inc, 858-859
General Electric Company, 860-861, 1026-1027
Henry Furnace & Foundry Co, 862-863
Ilg Electric Ventilating Company, 949
Lennox Furnace Co, Inc, 864-865
McQuay, Incorporated, 834
Meyer Furnace Company, The, 866
Modine Manufacturing Co, 959
Herman Nelson Corp, The, 970-971
John J Nesbitt, Inc, 972-973
Niagara Blower Co, 832-833
Norge Division, Borg-Warner Corporation, 873, 932
B F Sturtevant Co, 950
Thermal Units Manufacturing Company, 837
Trane Company, The, 1094
Unit Heater and Cooler Co, The, 960
Westinghouse Electric & Manufacturing Co, 840
L J Wing Mfg Co, 954-955
York Ice Machinery Corporation, 841
Young Radiator Company, 961

HEATERS, Automatic Hot Water, Domestic

American Gas Products Corp, 897
American Radiator Company, 843, 898-901, 974
Crane Co, 904-905
Davis Engineering Corporation, 975
Delco-Frigidaire Conditioning Division, General Motors Sales Corporation, 852-854
Electrol, Incorporated, 816-817, 927
Fitzgibbons Boiler Company, Inc, 918-919
Gar Wood Industries, Inc, 858-859
Henry Furnace & Foundry Co, 862-863
S T Johnson Co, 928-929
Norge Division, Borg-Warner Corporation, 873, 932
Sterling Engineering Company, 1089-1091
Taco Heaters, Inc, 978-979
Williams Oil-O-Matic Heating Corporation, 876-877
Westinghouse Electric & Manufacturing Co, 810
Young Radiator Company, 961

HEATERS, Blast

Aerofin Corporation, 962-964
Air Devices Corporation, 818-819
American Blower Corporation, 820-821
American Radiator Company, 843, 898-901, 974
Autovent Fan & Blower Co, 945
Bayley Blower Company, 946
Buckeye Blower Company, 972-973
Buffalo Forge Company, 947

Numerals following Manufacturers' Names refer to pages in the Catalog Data Section

Numerals following Manufacturers' Names refer to pages in the Catalog Data Section

Unit Heater and Cooler Co, The, 960
United States Radiator Corporation, 912-913
Warren Webster & Company, 1095-1097
Weil-McLain Company, 909
Young Radiator Company, 961

RECEIVERS, Air

Baker Ice Machine Co, Inc, 822
Binks Manufacturing Co, 938-939
Brownell Company, The, 1101
Crane Co, 904-905
Curtis Refrigerating Machine Company, Division Curtis Manufacturing Company, 826
Farrar & Tretts, Incorporated, 917
Illinois Engineering Company, 1082-1083
Ingersoll-Rand Company, 830-831
Kewanee Boiler Corporation, 920-921
Parks-Cramer Company, 835
Trane Company, The, 1094
Warren Webster & Company, 1095-1097

RECEIVERS, Ammonia

Carbondale Div, Worthington Pump & Machinery Co, 823
Frick Company (Incorporated), 827
York Ice Machinery Corporation, 841

RECEIVERS, Condensation

American Steam Pump Company, 1051
Baker Ice Machine Co, Inc, 822
Chicago Pump Company, 1054-1055
Crane Co, 904-905
Illinois Engineering Company, 1082-1083
Micro-Westco, Inc, 931
Nash Engineering Company, The, 1056-1057
Sarco Company, Inc, 1092-1093
Trane Company, The, 1094
Warren Webster & Company 1095-1097

RECEIVERS, Water Vapor

American Blower Corporation, 820-821
Canadian Sirocco Co, Ltd, 820-821
Illinois Engineering Company, 1082-1083
Trane Company, The, 1094
Warren Webster & Company, 1095-1097

RECORDERS, Humidity, Temperature

Bristol Company, The, 982
Brown Instrument Company, 983
Consolidated Ashcroft Hancock Co, Inc, 984
Julien P Friez & Sons, Inc, 1108
H-B Instrument Co, 985
Johnson Service Company, 1110-1111
Minneapolis-Honeywell Regulator Co, 1114-1115
National Regulator Co, 1116-1117
Powers Regulator Co, The, 1118-1119
Taylor Instrument Companies, 988-989

REFRIGERATING EQUIPMENT, Centrifugal

Carrier Corporation, 824
Ingersoll-Rand Company, 830-831

REFRIGERATING EQUIPMENT, Steam Jet

American Blower Corporation, 820-821
Canadian Sirocco Co, Ltd, 820-821
Carbondale Div, Worthington Pump & Machinery Co, 823
Carrier Corporation 824
Ingersoll-Rand Company, 830-831
Westinghouse Electric & Manufacturing Co, 840
Williams Oil-O-Matic Heating Corporation, 876-877

REFRIGERATING MACHINERY

Air Devices Corporation, 818-819
Airtemp Incorporated, 844-845
Baker Ice Machine Co, Inc, 822
Carbondale Div, Worthington Pump & Machinery Co, 823
Carrier Corporation, 824
Curtis Refrigerating Machine Company, Division of Curtis Manufacturing Company, 826
Delco-Frigidaire Conditioning Division, General Motors Sales Corporation, 852-854
Fairbanks Morse & Co, 828-829
Frick Company (Incorporated), 827
General Electric Company, 860-861, 1026-1027
Ingersoll-Rand Company, 830-831
Kelvinator Division of Nash-Kelvinator Corp, 867-869, 930
Norge Division, Borg-Warner Corporation, 873, 932
Servel, Inc, 836
Thermal Units Manufacturing Company 837
Universal Cooler Corporation, 838
Vilter Manufacturing Company, The, 839
Westinghouse Electric & Manufacturing Co, 840
Williams Oil-O-Matic Heating Corporation, 876-877
York Ice Machinery Corporation, 841

REFRIGERATION EQUIPMENT, Water Vapor

Ingersoll-Rand Company, 830-831

REGISTERS (*See Grilles, Registers, etc*)

REGULATORS, Air Volume

Young Regulator Company, The, 881

REGULATORS, Damper

American Radiator Company, 813-898-901, 974
Barber-Colman Company, 1104-1105
Barnes & Jones, Incorporated, 1071
E K Campbell Heating Co, 969
Carrier Corporation, 824
Consolidated Ashcroft Hancock Co, Inc, 984
Cook Electric Company, 1106
Davis Regulator Co, 1075
Detroit Lubricator Company, 1107
C A Dunham Company, 1076-1077
Foster Engineering Co, 1128
Julien P Friez & Sons, Inc, 1108
Fulton Sylphon Co, 1109
General Controls, 1112
General Electric Company, 860-861, 1026-1027
Williams S Haines & Company, 1079
Hart & Cooley Manufacturing Co, 1062-1063
Henry Furnace & Foundry Co, 862-863
Hoffman Specialty Co, Inc, 1080-1081
Illinois Engineering Company, 1082-1083
Johnson-Service Company, 1110-1111
Kieley & Mueller, Inc, 1081
Leeds & Northrup Company, 987
Minneapolis-Honeywell Regulator Co, 1114-1115
National Regulator Co, 1116-1117
Powers Regulator Co, The, 1118-1119
Sarco Company, Inc 1092-1093
Spence Engineering Co, 1122
Sylphon Control Systems, Inc, 1123
Taylor Instrument Companies, 988-989
H A Thrush & Co, 980-981
Trane Company, The, 1094
Tuttle & Bailey, Inc, 1064-1065
Warren Webster & Company, 1095-1097

REGULATORS, Feed Water

Beaton & Cadwell Mfg Company, 1072-1073
Foster Engineering Co, 1128
Kieley & Mueller, Inc, 1081
McDonnell & Miller, 894-895
Mueller Steam Specialty Co, Inc, 1088
Powers Regulator Co, The, 1118-1119
Spence Engineering Co, 1122
H A Thrush & Co, 980-981
Westinghouse Electric & Manufacturing Co, 840
Wright-Austin Co, 1098

REGULATORS, Furnace

Air Controls, Inc, 842
Barber-Colman Company, 1104-1105
Cook Electric Company, 1106
Detroit Lubricator Company, 1107
Fox Furnace Co, The, 848-851
Hart & Cooley Manufacturing Co, 1062-1063
Henry Furnace & Foundry Co, 862-863
National Regulator Co, 1116-1117
Spence Engineering Co, 1122

REGULATORS, Humidity (*See Humidity Control*)

REGULATORS, Pressure

American Radiator Company, 813, 898-901 974
Beaton & Cadwell Mfg Company, 1072-1073
Binks Manufacturing Co, 938-939
Bristol Company, The, 982
Consolidated Ashcroft Hancock Co, Inc 984
Crane Company, 904-905
Davis Regulator Co, 1075
Detroit Lubricator Company, 1107
C A Dunham Company, 1076-1077
Fedders Manufacturing Co, 958
Foster Engineering Co, 1128
Fulton Sylphon Co, 1109
General Controls, 1112
General Electric Company, 860-861, 1026-1027
Henry Furnace & Foundry Co, 862-863
Illinois Engineering Company, 1082-1083

STRAINERS, Oil

Crane Co, 904-905
Davis Regulator Co, 1075
Detroit Lubricator Company, 1107
Foster Engineering Co, 1128
General Controls, 1112
General Electric Company, 860-861, 1026-1027
Illinois Engineering Company, 1082-1083
Kieley & Mueller, Inc, 1084
Milwaukee Valve Co, 1086-1087
Mueller Steam Specialty Co, Inc, 1088
Sarco Company, Inc, 1092-1093
Spence Engineering Co, 1122
Staynew Filter Corporation, 890-891
Wright-Austin Co, 1098

STRAINERS, Steam

Crane Co, 904-905
Davis Regulator Co, 1075
Detroit Lubricator Company, 1107
Foster Engineering Co, 1128
General Controls, 1112
Illinois Engineering Company, 1082-1083
Kieley & Mueller, Inc, 1084
Milwaukee Valve Co, 1086-1087
Mueller Steam Specialty Co, Inc, 1088
Powers Regulator Co, The, 1118-1119
Sarco Company, Inc, 1092-1093
Spence Engineering Co, 1122
Trane Company, The, 1094
Wright-Austin Co, 1098

STRAINERS, Water

Crane Co, 904-905
Davis Regulator Co, 1075
Detroit Lubricator Company, 1107
Foster Engineering Co, 1128
General Controls, 1112
Illinois Engineering Company, 1082-1083
Kieley & Mueller, Inc, 1084
McDonnell & Miller, 894-895
Milwaukee Valve Co, 1086-1087
Mueller Steam Specialty Co, Inc, 1088
Powers Regulator Co, The, 1118-1119
Sarco Company, Inc, 1092-1093
Staynew Filter Corporation, 890-891
Wright-Austin Co, 1098

TANK COILS (*See Coils, Tank*)

TANK COVERING (*See Covering, Pipes and Surfaces*)

TANK HEATERS (*See Heaters, Tank*)

TANKS, Blow-off

Bigelow Company, The, 915
Brownell Company, The, 1101
Farrar & Trefts Incorporated, 917
Kewanee Boiler Corporation, 920-921

TANKS, Pressure

Bell and Gossett Company, 976-977
Bigelow Company, The, 915
Binks Manufacturing Co, 938-939
Brownell Company, The, 1101
Burnham Boiler Corporation, 902-903
Farrar & Trefts Incorporated, 917
Frick Company (Incorporated), 827
Kewanee Boiler Corporation, 920-921
H A Thrush & Co, 980-981

TANKS, Storage

American Radiator Company, 843, 898-901, 974
E B Badger & Sons Co, 941
Bigelow Company, The, 915
Brownell Company, The, 1101
Burnham Boiler Corporation, 902-903
Farrar & Trefts Incorporated, 917
Frick Company (Incorporated), 827
Gilbert & Barker Mfg Co, 855-857
Kewanee Boiler Corporation, 920-921

TEMPERATURE CONTROL

Air Devices Corporation, 818-819
American Radiator Company, 843, 898-901, 974
Barber-Colman Company, 1104-1105
Barnes & Jones, Incorporated, 1074
Bristol Company, The, 982
Brown Instrument Company, 983
Builders Iron Foundry, 1023
Carrier Corporation, 824
Consolidated Ashcroft Hancock Co, Inc, 984
Cook Electric Company, 1106
Davis Regulator Co, 1075
Delco-Frigidaire Conditioning Division, General Motors Sales Corporation, 832-834
Detroit Lubricator Company, 1107
Dole Valve Company, The, 1127
C A Dunham Company, 1076-1077
Foster Engineering Co, 1128
Fox Furnace Co, The, 848-851
Julien P Friez & Sons, Inc, 1108
Fulton Sylphon Co, 1109
General Controls, 1112
General Electric Company, 860-861, 1026-1027
H-B Instrument Co, 985
Illinois Engineering Company, 1082-1083
Illinois Testing Laboratories, Inc, 986
Johnson Service Company, 1110-1111
Kieley & Mueller, Inc, 1084
Leeds & Northrup Company, 987
Mercoid Corporation, The, 1113
Minneapolis-Honeywell Regulator Co, 1114-1115
National Regulator Co, 1116-1117
Penn Electric Switch Co, 1120
Powers Regulator Co, The, 1118-1119
Ranco, Inc, 1121
Sarco Company, Inc, 1092-1093
Spence Engineering Co, 1122
Sterling Engineering Company, 1089-1091
Sylphon Control Systems, Inc, 1123
Taylor Instrument Companies, 988-989
Thermal Units Manufacturing Company, 837
H A Thrush & Co, 980-981
Trane Company, The, 1094
Warren Webster & Company, 1095-1097
Westinghouse Electric & Manufacturing Co, 840
L J Wing Mfg Co, 954-955
Yarnell-Waring Co, 1099

THERMOMETERS, Distance Type

Bristol Company, The, 982
Brown Instrument Company, 983
Builders Iron Foundry, 1023
Consolidated Ashcroft Hancock Co, Inc, 984
Julien P Friez & Sons, Inc, 1108
H-B Instrument Co, 985
Illinois Testing Laboratories, Inc, 986
Johnson Service Company, 1110-1111
Leeds & Northrup Company, 987
Minneapolis-Honeywell Regulator Co, 1114-1115
Powers Regulator Co, The, 1118-1119
Sarco Company, Inc, 1092-1093
Taylor Instrument Companies, 988-989
United States Gauge Co, 991

THERMOMETERS, Indicating

Bell and Gossett Co, 976-977
Bristol Company, The, 982
Brown Instrument Company, 983
Builders Iron Foundry, 1023
Consolidated Ashcroft Hancock Co, Inc, 984
Julien P Friez & Sons, Inc, 1108
H-B Instrument Co, 985
Illinois Testing Laboratories, Inc, 986
Johnson Service Company, 1110-1111
Leeds & Northrup Company, 987
Minneapolis-Honeywell Regulator Co, 1114-1115
Palmer Company, The, 990
Powers Regulator Co, The, 1118-1119
Sarco Company, Inc, 1092-1093
Taylor Instrument Companies, 988-989
H A Thrush & Co, 980-981
United States Gauge Co, 991

THERMOMETERS, Recording

Bristol Company, The, 982
Brown Instrument Company, 983
Builders Iron Foundry, 1023
Consolidated Ashcroft Hancock Co, Inc, 984
Julien P Friez & Sons, Inc, 1108
H-B Instrument Co, 985
Leeds & Northrup Company, 987
Minneapolis-Honeywell Regulator Co, 1114-1115
Powers Regulator Co, The, 1118-1119
Taylor Instrument Companies, 988-989
H A Thrush & Co, 980-981

THERMOSTATS

American Radiator Company, 843, 898-901, 974
Barber-Colman Company, 1104-1105
Bristol Company, The, 982
Carrier Corporation, 824
Consolidated Ashcroft Hancock Co, Inc, 984
Cook Electric Company, 1106
Detroit Lubricator Company, 1107
Julien P Friez & Sons, Inc, 1108
Fulton Sylphon Co, 1109
General Controls, 1112
General Electric Company, 860-861, 1026-1027
H-B Instrument Co, 985
Illinois Engineering Company, 1082-1083
Johnson Service Company, 1110-1111
Mercoid Corporation, The, 1113
Minneapolis-Honeywell Regulator Co, 1114-1115
National Regulator Co, 1116-1117
Penn Electric Switch Co, 1120

Numerals following Manufacturers' Names refer to pages in the Catalog Data Section

UNDERGROUND PIPE CONDUITS (*See Conduits, Underground Pipe*)

UNIT HEATERS (*See Heaters, Unit*)

UNIT VENTILATORS (*See Ventilators, Unit*)

UNITS, Air Conditioning (*See Air Conditioning Units*)

V-BELT DRIVES

American Coolair Corp , 942-943
Binks Manufacturing Co , 938-939
Carbondale Div , Worthington Pump & Machinery Co , 823
Frick Company (Incorporated), 827

VACUUM HEATING SYSTEMS (*See Heating Systems, Vacuum*)

VALVES, Air

American Radiator Company, 843, 898-901, 974
Anderson Products, Inc , 1124-1125
Beaton & Cadwell Mfg Company, The, 1072-1073
Bell & Gossett Company, 976-977
Binks Manufacturing Co , 938-939
Bristol Company, The, 982
Burnham Boiler Corporation, 902-903
Carbondale Div , Worthington Pump & Machinery Co 823
Consolidated Ashcroft Hancock Co , Inc , 984
Crane Co , 904-905
Curtis Refrigerating Machine Company, Division of Curtis Manufacturing Company, 826
Davis Regulator Co , 1075
Detroit Lubricator Company, 1107
Dole Valve Company, The, 1127
C A Dunham Company, 1076-1077
Foster Engineering Co , 1128
Hoffman Specialty Co , Inc , 1080-1081
Jenkins Bros , 1129
Milwaukee Valve Co , 1086-1087
National Regulator Co , 1116-1117
New York Air Valve Corporation, 1130
Spence Engineering Co , 1122
Sterling Engineering Company, 1089-1091
Trane Company, The, 1094
Wright-Austin Co , 1098

VALVES, Angle, Globe and Cross

American Brass Co , The, 1028-1029
American Radiator Company, 843, 898-901, 974
Baker Ice Machine Co , Inc , 822
Burnham Boiler Corporation, 902-903
Carbondale Div , Worthington Pump & Machinery Co , 823
Consolidated Ashcroft Hancock Co , Inc , 984
Crane Co , 904-905
Detroit Lubricator Company, 1107
Frick Company (Incorporated), 827
Grinnell Company, Inc , 966-968, 1078
Illinois Engineering Company, 1082-1083
Jenkins Bros , 1129
Milwaukee Valve Co , 1086-1087
Mueller Brass Co , 1034-1035
Streamline Pipe & Fittings Co , 1034-1035
York Ice Machinery Corporation, 841

VALVES, Automatic

Alco Valve Co , Inc , 1126
American Radiator Company, 843, 898-901, 974
Anderson Products, Inc , 1124-1125
Baker Ice Machine Co , Inc , 822
Barber-Colman Company, 1104-1105
Beaton & Cadwell Mfg Company, The, 1072-1073
Bell & Gossett Company, 976-977
Bristol Company, The, 982
Consolidated Ashcroft Hancock Co , Inc , 984
Crane Co , 904-905
Davis Regulator Co , 1075
Detroit Lubricator Company, 1107
Foster Engineering Co , 1128
Frick Company (Incorporated), 827
Julien P Friez & Sons, Inc , 1108
Fulton Sylphon Co , 1109
General Controls, 1112
Kieley & Mueller, Inc , 1084
Minneapolis-Honeywell Regulator Co , 1114-1115
New York Air Valve Corporation, 1130
Sarco Company, Inc , 1092-1093
Spence Engineering Co , 1122
Sterling Engineering Company, 1089-1091
Sylphon Control Systems, Inc , 1123
H A Thrush & Co , 980-981
Trane Company, The, 1094

VALVES, Back Pressure

Baker Ice Machine Co , Inc , 822
Crane Co , 904-905
Davis Regulator Co , 1075
Fedders Manufacturing Co , 958
Foster Engineering Co , 1128
Illinois Engineering Co , 1082-1083
Jenkins Bros , 1129
Kieley & Mueller, Inc , 1084
Mueller Steam Specialty Co , Inc , 1088
Spence Engineering Co , 1122
Taylor Instrument Companies, 988-989
Warren Webster & Co , 1095-1097
York Ice Machinery Corporation, 841

VALVES, Balanced

Consolidated Ashcroft Hancock Co , Inc , 984
Crane Co , 904-905
Davis Regulator Co , 1075
Foster Engineering Co , 1128
Illinois Engineering Company, 1082-1083
Jenkins Bros , 1129
Kieley & Mueller, Inc , 1084
Mueller Steam Specialty Co , Inc , 1088

VALVES, Blow-off

Consolidated Ashcroft Hancock Co , Inc , 984
Crane Co , 904-905
Jenkins Bros , 1129
Yarnall-Waring Co , 1099

VALVES, By-pass

Consolidated Ashcroft Hancock Co , Inc , 984
Crane Co , 904-905
Jenkins Bros , 1129

VALVES, Check

Consolidated Ashcroft Hancock Co , Inc , 984
Crane Co , 904-905
Davis Regulator Co , 1075
Fedders Manufacturing Co , 958
Foster Engineering Co , 1128
Frick Company (Incorporated), 827
Grinnell Company, Inc , 966-968, 1078
Illinois Engineering Company, 1082-1083
Jenkins Bros , 1129
Milwaukee Valve Co , 1086-1087
Taco Heaters, Inc , 978-979
York Ice Machinery Corporation, 841

VALVES, Diaphragm

Alco Valve Co , Inc , 1126
Consolidated Ashcroft Hancock Co , Inc , 984
Davis Regulator Co , 1075
Foster Engineering Co , 1128
Illinois Engineering Company, 1082-1083
National Regulator Co , 1116-1117
Powers Regulator Co , The, 1118-1119
Taylor Instrument Companies, 988-989
H A Thrush & Co , 980-981

VALVES, Expansion

Alco Valve Co , Inc , 1126
Crane Co , 904-905
Delco-Frigidaire Conditioning Division, General Motors Sales Corporation, 852-854
Detroit Lubricator Company, 1107
Fedders Manufacturing Co , 958
Foster Engineering Co , 1128
Frick Company (Incorporated), 827
Fulton Sylphon Co , 1109
York Ice Machinery Corporation, 841

VALVES, Float

Alco Valve Co , Inc , 1126
Anderson Products, Inc , 1124-1125
Baker Ice Machine Co , Inc , 822
Crane Co , 904-905
Davis Regulator Co , 1075
Detroit Lubricator Company, 1107
Dole Valve Company, The, 1127
C A Dunham Company, 1076-1077
Fedders Manufacturing Co , 958
Foster Engineering Co , 1128
Frick Company (Incorporated), 827
General Electric Company, 860-861, 1026-1027
Arthur Harris & Co , 1033
Illinois Engineering Company, 1082-1083
Kieley & Mueller, Inc , 1084
McDonnell & Miller, 894-895
Mueller Steam Specialty Co , Inc , 1088
Spence Engineering Co , 1122
Sterling Engineering Company, 1089-1091
Trane Company, The, 1094
York Ice Machinery Corporation, 841

VALVES, Flow Control

Bell and Gossett Co , 976-977
Bristol Company, The, 982
Consolidated Ashcroft Hancock Co , Inc , 984
Crane Co , 904-905
Davis Regulator Co , 1075

Numerals following Manufacturers' Names refer to pages in the Catalog Data Section

Roll of Membership

AMERICAN SOCIETY *of*
HEATING *and* VENTILATING ENGINEERS

1937

Contains Lists of Members Arranged Alphabetically and Geographically, also Lists of Officers and Committees, Past Officers and Local Chapter Officers

Corrected to January 1, 1937
Published at the Headquarters of the Society
51 Madison Avenue, New York, N. Y.

Officers and Council

AMERICAN SOCIETY *of* HEATING *and* VENTILATING ENGINEERS

51 Madison Ave., New York, N. Y.

1936-37

President	G. L. LARSON
First Vice-President	D S BOYDEN
Second Vice-President	E. H. GURNEY
Treasurer	A. J. OFFNER
Secretary	A. V. HUTCHINSON

Council

G. L. LARSON, *Chairman*
D. S. BOYDEN, *Vice-Chairman*

One Year	*Two Years*	*Three Years*
M. C. BEMAN	ALBERT BUENGER	R. C. BOLSINGER
JOHN HOWATT	F. E. GIESECKE	S H. DOWNS
L. W. MOON	J. F. MCINTIRE	W. L. FLEISHER
O. W. OTT	W. E. STARK	C. M. HUMPHREYS
W. A RUSSELL		

Committees of the Council

Executive· John Howatt, *Chairman;* L W. Moon, M. C. Beman.
Finance. J F McIntire, *Chairman;* E. H. Gurney, D. S. Boyden.
Meetings: Albert Buenger, *Chairman,* F E. Giesecke, C. M Humphreys.
Membership· W. A. Russell, *Chairman,* R. C. Bolsinger, W. L. Fleisher.

Advisory Council

John Howatt, *Chairman,* Homer Addams, R. P. Bolton, W. H Carrier, S E. Dibble, W. H. Driscoll, H. P Gant, John F. Hale, L. A Harding, H. M. Hart, C. V Haynes, E. Vernon Hill, J D. Hoffman, W. T. Jones, D. D. Kimball, S R. Lewis, Thornton Lewis, J. I. Lyle, D M. Quay, C. L. Riley, F. B. Rowley, F. R. Still and A. C. Willard.

Cooperating Committee

A.S.H.V E. Representative on *National Research Council,* Prof. F. E. Giesecke (1 year).

Special Committees

Committee on Admission and Advancement E N. Sanbern, *Chairman* (*one year*), E J Ritchie (*two years*), and T H Urdahl (*three years*)

Publication Committee M C Beman, *Chairman* (*one year*), A I Brown (*two years*), and F C. McIntosh (*three years*)

Committee on Constitution and By-Laws R H Carpenter, *Chairman*, O W Ott and W. E Stark

Guide Publication Committee J H Walker, *Chairman*, Albert Buenger, S H Downs, C H. B Hotchkiss, W H Severns, and E N McDonnell, *Advisory*

F Paul Anderson Medal Award Committee D S Boyden, *Chairman*, John Howatt, F C McIntosh, F B Rowley, and W E Stark

A S.H V E -A S R E Committee on National Standards for Air Conditioning Applications L A Harding, *Chairman*, Glenn Muffly, *Vice-Chairman*, W L Fleisher, D E French, John Howatt, A P Kratz, H J Macintire, L A Philipp, C W Walton, Jr , and W. E Zieber.

Committees—1936

Nominating Committee for 1936

Chapter	*Representative*	*Alternate*
Cincinnati	H E Sproull	J J Braun
Cleveland	M F Rather	C F Eveleth
Illinois	J H Milliken	J J Hayes
Kansas City	C. A Weiss	L A Stephenson
Manitoba	C H Turland	J E Yates
Massachusetts	W A McPherson	W T Jones
Michigan	Thomas Chester	G. H. Tuttle
Western Michigan	J H Van Alsburg	S H Downs
Minnesota	C E Gausman	N D. Adams
New York	W W Timmis	E. J Ritchie
Western New York	W E Voisinet	B C Candee
Oklahoma City	F X Loeffler	E F Dawson
Ontario	M W Shears	H H Angus
Pacific Northwest	Lincoln Bouillon	E. O Eastwood
Philadelphia	H H Erickson	M F Blankin
Pittsburgh	R. J. J. Tennant	F C McIntosh
St. Louis	R J. Tenkonohy	J W Cooper
Southern California	W E Barnum, Jr	Leo Hungerford
Washington, D. C.	L S Ourusoff	T H Urdahl
Wisconsin	E A. Jones	Ernest Szekely

Committee on Research

W A Danielson, *Chairman*
C. E. Lewis, *Vice-Chairman*
F C Houghten, *Director*
Dr. A. C Willard, *Technical Adviser*
O. P. Hood, *Ex-Officio Member*

One Year	*Two Years*	*Three Years*
C. A. Booth	C A Dunham	W. A Danielson
E. K. Campbell	W. L Fleisher	C E Lewis
John Howatt	Elliott Harrington	D. W. Nelson
A. J. Nesbitt	A P Kratz	Cyril Tasker
J. H Walker	H C Murphy	C -E. A. Winslow

Executive Committee

W A DANIELSON, *Chairman*

C E LEWIS   A P KRATZ

Finance Committee

G D KINGSLAND, *Chairman*

C V HAYNES   E N MCDONNELL

A R HERSKE   T H URDAHL

Technical Advisory Committees, 1936-1937

Committee on Air Cleaning Devices H C. Murphy, *Chairman,* M I. Dorfan, S R Lewis, G W. Penney, D C Simpson, W O Vedder

Committee on Air Conditions in Relation to Comfort C P Yaglou, *Chairman,* J J Aeberly, R R Sayers, C.-E. A Winslow.

Committee on Air Conditioning in the Treatment of Diseases Dr E V. Hill, *Chairman,* N D Adams, J. J Aeberly, John Howatt, A P Kratz, V L Sherman, E L Stammer

Committee on Air Conditioning Requirements of Glass M L Carr, *Chairman,* F. L. Bishop, A. N Finn, E H. Hobbie, R J Lillibridge, R A Miller, F. W. Parkinson, W C Randall, Cyril Tasker, G B Watkins

Committee on Atmospheric Impurities and Resulting Safety and Health Requirements Theodore Hatch, *Chairman* J J Bloomfield, C A Booth, Philip Drinker, Dr Leonard Greenburg, H B Meller

Committee on Climate and Its Relation to Air Conditioning Fundamentals: Dr C A Mills, *Chairman,* Major G. C. Dunham, James Govan, Dr W J. McConnell, C. F Neergaard, Dr F. M Pottenger, Jr, E L Weber, Prof C.-E A Winslow.

Committee on Comfort Requirements for Summer Cooling: W L Fleisher, *Chairman,* A E. Beals, F. R Bichowsky, Thomas Chester, F. E Giesecke, Elliott Harrington, Dr E. V Hill, R E Keyes, Cyril Tasker, C P Yaglou

Committee on Corrosion in Air Conditioning Equipment: A E Stacey, Jr, *Chairman,* M. L Diver, M. S. Kice, F L LaQue, C E Lewis, R M. Palmer, F N. Speller, C. M. Sterne, R T. Thornton, J H Young

Committee on Corrosion in Heating Systems. J. H Walker, *Chairman,* E L Chappell, W H. Driscoll, C. A Dunham, L B Miller, R. R Seeber, C M. Sterne

Committee on Direct and Indirect Radiation with Gravity Air Circulation M K. Fahnestock, *Chairman;* H F Hutzel, J P. Magos, J W McElgin, J F. McIntire, D W. Nelson, R N. Trane, T. A Novotney.

Committee on Effect of Entering Temperature and Velocity on Temperature and Distribution of Air within an Enclosure: Ernest Szekely, *Chairman,* S H. Downs, M. K Fahnestock, D W. Nelson, C H. Randolph, J. E Schoen, G L Tuve, J H. Van Alsburg

Committee on Frictional Resistance to Flow of Air in Small Ducts and Fittings J H. Van Alsburg, *Chairman,* C. A. Booth, S H. Downs, R D Madison, L B. Miller, L E Smith, C. M. Humphreys.

Committee on Heating and Cooling Requirements of Buildings: O. W. Armspach, *Chairman;* P D. Close, W H Driscoll, H M Hart, P E Holcombe, V. W. Hunter, H H Mather, E. C Rack, F B Rowley, R J. J. Tennant, J. H Walker.

Committee on Heat Transfer of Finned Tubes with Forced Air Circulation: F. B. Rowley, *Chairman;* H. F Hutzel, C E. Lewis, E J Lindseth, R. H Norris, W E. Stark, G. L. Tuve.

Committee on Intermittent Heating of Churches, Auditoriums and Similar Buildings: E. K. Campbell, *Chairman;* W L. Cassell, Prof E F. Dawson, N. W. Downs, J M Robertson, J. H Kitchen, Prof. A. H Sluss.

Committee on Insulation. L. A Harding, *Chairman,* E A. Allcut, J. D. Edwards, E. C. Lloyd, R. T Miller, E. R Queer, T. S Rogers, F. B Rowley, W. S. Steele, Cyril Tasker, B. Townshend, G B. Wilkes.

Committee on Psychrometry F R. Bichowsky, *Chairman,* C A Bulkeley, J. A Goff, Dr. E. V. Hill, F. G. Keyes, A. P. Kratz, W M. Sawdon.

Committee on Railroad Air Conditioning: A I Brown, *Chairman,* F. E Giesecke, C S Leopold, S R. Lewis, L W Wallace

Committee on Refrigeration in Relation to Air Treatment: M K. Fahnestock, *Chairman,* E. A. Brandt, John Everetts, Jr., Elliott Harrington, E. D. Milener, K. W. Miller, E B. Newill, F. G. Sedgwick, J. H Walker

Committee on Sound in Relation to Heating and Ventilation J. S. Parkinson, *Chairman;* C M. Ashley, C A Booth, V. O Knudsen, R. F. Norris, C. H Randolph, J. P. Reis, F. R. Watson

Committee on Treatment of Air with Electricity: Prof. C.-E. A. Winslow, *Chairman,* Prof. Vannevar Bush, W. H. Carrier, L W. Chubb, Major W. D. Fleming, R. F. James, L R. Koller, Dr C A. Mills, Prof E. B Phelps, Prof C. R Wait, Prof. W. T Wells.

Officers of Local Chapters, 1936-37

Cleveland
Headquarters, Cleveland, Ohio
Meets. Second Thursday in Month
President, L T Avery
2341 Carnegie Avenue
Secretary, W R Beach
75 Public Square

Cincinnati
Headquarters, Cincinnati, Ohio
Meets· Second Tuesday in Month
President, E B Royer
6635 Iris Avenue
Secretary, K W Schick
561 Reading Road

Illinois
Headquarters, Chicago, Ill
Meets Second Monday in Month
President, J J Hayes
53 W Jackson Boulevard
Secretary, C E Price
6 N Michigan Avenue

Kansas City
Headquarters, Kansas City, Mo
Meets· Second Monday in Month
President, L R Chase
2440 Pennway
Secretary, F J Dean, Jr
6028 Walnut Street

Manitoba
Headquarters, Winnipeg, Man., Canada
Meets Fourth Thursday in Month
President, J B Steele
184 Waterloo Street
Secretary, C H Turland
325 Centennial Street

Massachusetts
Headquarters, Boston, Mass
Meets Second Tuesday in Month
President, James Holt
Massachusetts Institute of Technology, Cambridge, Mass
Secretary, John Turner
285 Columbus Avenue

Michigan
Headquarters, Detroit, Mich
Meets: First Monday after the 10th of the Month
President, R. K Milward
127 Campbell Avenue
Secretary, G H Tuttle
2000 Second Avenue

Western Michigan
Headquarters, Grand Rapids, Mich
Meets Second Monday in Month
President, L G Miller
Mich State College, E Lansing, Mich
Secretary, E I Adams
115 S Pine, Lansing, Mich

Minnesota
Headquarters, Minneapolis, Minn
Meets: Second Monday in Month
President, N D Adams
220 Second Avenue S W , Rochester, Minn
Secretary, J E Swenson
800 Hennepin Avenue

Montreal
Headquarters, Montreal, Que , Canada
Meets: Second Monday in Month
President, G L Wiggs
University Tower
Secretary, C W. Johnson
630 Dorchester Street W

New York
Headquarters, New York, N Y
Meets· Third Monday in Month
President, G E Olsen
101 Park Avenue
Secretary, T W Reynolds
100 Pinecrest Dr., Hastings-on-Hudson, N. Y

Western New York
Headquarters, Buffalo, N Y
Meets Second Monday in Month
President, P S Hedley
Curtiss Bldg
Secretary, C A Gifford
758 Parkside Ave

Oklahoma City
Headquarters, Oklahoma City, Okla
President, F X Loeffler
1604 N W Fifth Street
Secretary, E W Gray
Box 1498

Ontario
Headquarters Toronto, Ont , Canada
Meets· First Monday in Month
President, Thomas McDonald
117 Peter Street
Secretary, H R Roth
57 Bloor Street W

Pacific Northwest
Headquarters, Seattle, Wash
Meets Second Tuesday in Month
President, S D Peterson
473 Colman Bldg
Secretary, D C Griffin
Orpheum Bldg

Philadelphia
Headquarters, Philadelphia, Pa.
Meets. Second Thursday in Month
President, W F Smith
422 Bryn Mawr Avenue, Cynwyd, Pa
Secretary, H H Erickson
804 Architects Bldg

Pittsburgh
Headquarters, Pittsburgh, Pa
Meets: Second Monday in Month
President, M L Carr
P O Box 1646
Secretary, T F Rockwell
Carnegie Inst Tech

St. Louis
Headquarters, St Louis, Mo
Meets· First Tuesday in Month
President, C R Davis
2328 Locust Street
Secretary, R J Tenkonohy
3650 Shaw Boulevard

Southern California
Headquarters, Los Angeles, Calif
Meets Second Tuesday in Month
President, Leo Hungerford
524 Loew's State Bldg
Secretary, H M Hendrickson
5051 Santa Fe Avenue

Texas
Headquarters, College Station, Texas
President, F E Giesecke
Texas A & M College
Secretary, W H Badgett
Texas A & M College

Washington, D. C.
Headquarters, Washington, D C
Meets: Second Wednesday in Month
President, W A Danielson
2236 Munitions Bldg
Secretary, M D Kiczales
3000 Connecticut Avenue N W

Wisconsin
Headquarters, Milwaukee, Wis
Meets: Third Monday in Month
President, M E Erickson
8405 W Greenfield Ave , West Allis, Wis
Secretary, R. G. Koch
626 E Wisconsin Avenue

Roll of Membership

AMERICAN SOCIETY *of* HEATING *and* VENTILATING ENGINEERS

1937

HONORARY MEMBERS

BALDWIN, WM. J. (1915), New York, N Y (Deceased May 7, 1924.)
BILLINGS, DR. J. S. (1896), New York, N Y (Deceased March 10, 1913.)
BOLTON, REGINALD PELHAM (1897), New York, N. Y.
BRECKENRIDGE, L. P. (1920), North Ferrisburg, Vt
GORMLY, JOHN (Charter Member), Norristown, Pa (Deceased January 31, 1929.)
NEWTON, C. W. (Charter Member), Baltimore, Md (Deceased August 6, 1920.)
HOOD, O. P. (1929), Washington, D. C
JELLETT, STEWART A. (Charter Member), (Presidential Member), Philadelphia, Pa (Deceased April 5, 1935)

LIST OF MEMBERS Arranged Alphabetically

(Asterisk indicates authorship of papers)

(*M* 1923; *A* 1918, *J* 1916) indicates, Election as Member 1923; Associate 1918; Junior 1916
(Pres. 1923) indicates, Elected President in 1923 and is now a Presidential Member

A

ABRAMS, Abraham (*M* 1927; *J* 1924), Pres, Abbey Heating Co, Inc, 81 Centre Ave, and (for mail), 100 Clove Rd, New Rochelle, N Y

ACHESON, Albert R. (*M* 1919), Consulting Engr (for mail), 501 Eckel Theatre Bldg, and 852 Ostrom Ave, Syracuse, N. Y

ADAMS, Benjamin (*M* 1919), Dist Mgr (for mail), American Blower Corp, Room 781 Broad Street Station Bldg, and 3006 W Coulter St, Queen Lane Manor, Philadelphia, Pa

ADAMS, Benjamin C., Jr. (*S* 1936), Engrg. Student (for mail), 724 Chautauqua, Norman, Okla, and 5127 Sunset Drive, Kansas City, Mo.

ADAMS, Bruce P. (*A* 1936), Asst Gen Sales Mgr. (for mail), McDonnell & Miller, 400 N. Michigan Ave., and 444 St. James Place, Apt 804, Chicago, Ill

ADAMS, Charles W. (*M* 1920), Salesman, U. S. Radiator Corp, 1405 West 11th St, Kansas City, Mo.

ADAMS, Eugene I. (*M* 1934), Plant Engr, Michigan State College, and (for mail), 115 S. Pine, Lansing, Mich

ADAMS, Harold E. (*M* 1930), Chief Engr (for mail), Nash Engineering Co, Wilson Rd., South Norwalk and Merrill Heights, Norwalk, Conn.

ADAMS, Neil D. (*M* 1929, *A* 1925, *J* 1922), Supt, Franklin Heating Station (for mail), 220 Second Ave, S W., and 836 Eighth Ave, S.W, Rochester, Minn

ADDAMS, Homer (*Charter Member, Life Member*), (*Presidential Member*), (Pres, 1924; 1st Vice-Pres, 1923; Treas., 1915-1922, Council, 1915-1925), Pres, Kewanee Boiler Co, Inc, and Fitzgibbons Boiler Co, Inc, 101 Park Ave, New York, N Y.

ADLAM, T. Napier (*M* 1932), Vice-Pres and Gen Mgr, Sarco Mfg Co, 183 Madison Ave, New York, N Y, and (for mail), 64 Wellington Ave, West Orange, N J

ADLER, Alphonse A.* (*M* 1921), Consulting Engr, 35 Stewart Ave, Arlington, N. J.

ADLER, Jack C. (*J* 1936), Sales Mgr, Air Cond Dept, Frigidaire Corp, 224 West 57th St, New York, and (for mail), c/o B. W Adler, 6952 Groton St, Forest Hills, L I., N Y

ADSHEAD, Bernard (*J* 1936), Tech Director, National Air Conditioning & Humidifying Co, Ltd, 46 Britannic Bldg, Manchester, and (for mail), Elm Bungalow, Brookside Rd, Gatley, Cheshire, England

AEBERLY, John J.* (*M* 1928), Chief of Div. of Htg, Vtg and Ind. Sanitation, Chicago Board of Health, 707 City Hall, and (for mail), 6225 N Newcastle Ave, Norwood Park P. O, Chicago, Ill

AHEARN, William J. (*M* 1929), Htg and Vtg. Engr, 21 Lake Rd., Cochituate, Mass.

AHLBERG, Henry B. (*J* 1933), Chief Engr, Anderson Products Co, 17 Tudor St, Cambridge, and (for mail), 146 Orlando St, Mattapan, Mass

AHLFF, Albert A. (*M* 1923; *A* 1918), Chase Brass & Copper Co, Waterbury, Conn, and (for mail), Walnut Park Plaza, 63rd and Walnut Sts., Apt. 204, Philadelphia, Pa

AIKEN, Jack F. (*J* 1936; *S* 1935), 312 Walnut St., S E., Minneapolis, Minn

AIKMAN, Joseph M. (*M* 1936), Consulting Engr., 128 East 18th St., Minneapolis, Minn.

AITKEN, James (*A* 1935), Supt. (for mail), Imperial Iron Corp, Ltd., Saskatchewan Ave., and 923 Palmerston Ave, Winnipeg, Man, Canada

AKERS, George W. (M 1929), Secy-Treas (for mail), George W Akers Co, 2847 Grand River Ave, Detroit, and 424 Willits St, Birmingham, Mich

ALEXANDER, Samuel W (M 1935), Mgr Htg Div, James Morrison Brass Co, 276 King St, S W, and (for mail), 124 Kingsmount Park Rd, Toronto, Ont, Canada

ALFAGEME, Braulio (M 1935), Engr, Mgr, B Alfageme, Almagro 1, Madrid, Spain

ALFSEN, Nikolai (M 1933), Civil Engr, Alsen & Gunderson, Oslo P O, Box 676, and (for mail), Shabekk near Oslo, Norway

ALGREN, Axel B.* (M 1930), Asst Prof Mech Engr, University of Minnesota, Exp Engrg Lab, and (for mail), 5109-17th Ave., S, Minneapolis, Minn

ALLAN, Norman J. (J 1934), Asst to Pres, Kansas City Pump Co, 1314-22 West 11th St, and (for mail), 3929 State Line Ave, Kansas City, Mo

ALLEN, A. Walter (M 1936), Sales Engr, Pease Foundry Co, Ltd, Toronto, and (for mail), 151 Glen Ave, Ottawa, Ont, Canada

ALLEN, DeWitt M (M 1936, J 1922), Dist Mgr (for mail), Ilg Electric Ventilating Co, 1004 Baltimore Ave, and 5700 Olive St, Kansas City, Mo

ALLEN, William W. (A 1936), Vice-Pres (for mail), American Coolair Corp, Box 2300, Jacksonville, and DaVinci St, Venetia, Fla

ALLMEN, Norman S. (J 1936, S 1934), 8420 Lake Ave, Cleveland, Ohio

ALLONIER, Howard R (A 1936), Dist Mgr (for mail), Buckeye Blower Co, Box 195 (425 W Town St), Columbus, and R F D No 1, Powell, Ohio

ALLSOP, Rowland P. (J 1934), Engr (for mail), Mathers & Haldenby, Archts, 96 Bloor St, W, and 89 Neville Park Blvd, Toronto, Ont, Canada

ALT, Harold L.* (M 1913), Bldg Equip Engr, Gibbs & Hill, Penn Station, New York, N Y, and (for mail), 18-C Kearny St., Newark, N. J

AMES, Charles F. (A 1928), Vice-Pres and Sales Mgr, Ames Pump Co, Inc, 30 Church St, and (for mail), 205 West 54th St, New York, N Y

AMMERMAN, Charles R. (M 1916), Consulting Engr (for mail), 772 Century Bldg, and 3908 Guilford Ave, Indianapolis, Ind

ANDEREGG, R. H. (M 1920), Vice-Pres, The Trane Co., and (for mail), 324 North 24th St, LaCrosse, Wis.

ANDERSON, Carroll S. (M 1920), Mgr (for mail), American Blower Corp, 429 Shell Bldg, 1008 W. Sixth St, and 4267 Holly Knoll Drive, Los Angeles, Calif

ANDERSON, David B. (J 1936, S 1933), Engr, Wood Conversion Co, 1981 First National Bank Bldg, and (for mail), 770 Holly Ave, St Paul, Minn

ANDERSON, George A M. (J 1936), 409 E Main St, Owatonna, Minn, and (for mail), c/o M E Anderson, 115 East 60th St, New York, N Y

ANDERSON, Sigurd H. (J 1936, S 1935), 3921 Bloomington Ave, Minneapolis, Minn.

ANDES, William (A 1934), Secy-Treas, Andico Co, 565 Stones Levee, and (for mail), 3947 West 162nd St., Cleveland, Ohio

ANDRESEN, Garwood C. (S 1936), Sales Engr., York Ice Machinery Corp, 42nd and Second Ave, Brooklyn, N Y.

ANDREWS, George H. (A 1934), Partner and Supt, Frank P. Andrews & Son, 354 Neshanock Ave, and (for mail), 213 Meyer Ave, New Castle, Pa

ANGERMEYER, Albert H. (A 1936), Owner (for mail), 119 N. Commercial St, and 705 E. Forest Ave, Neenah, Wis

ANGUS, Harry H.* (M 1918), (Council, 1927-1929), Consulting Engr, 1221 Bay St, and (for mail), 34 Farnham Ave, Toronto, Ont., Canada.

ANSPACHER, Thomas H. (J 1936), Mgr, Buffalo Forge Co, 315 Dwight Bldg, and (for mail), 611 East 61st St, Kansas City, Mo

ANTHES, Lawrence L (A 1935), Pres, Imperial Iron Corp, Ltd, 30 Jefferson Ave, and (for mail), Anthes Foundry, Ltd, 64 Jefferson Ave, and 119 Dowling Ave, Toronto, Ont, Canada

APT, Sanford R. (M 1935), Mech Engr, New York World's Fair 1939, Inc, 350 Fifth Ave, New York, and (for mail), 36-20-168th St, Flushing, N Y

ARCHER, David M. (M 1934), Sales Repr (for mail), Sarco Co, Inc, 143 Federal St, Boston, and 10 Thurlow St, West Roxbury, Mass

ARENBERG, Milton K. (A 1920), Pres (for mail), Robert Barclay, Inc, 122 N Peoria St, Chicago, and 1033 S Linden Ave, Highland Park, Ill

ARGUE, Edgar J. (A 1935), Sales Engr (for mail), Lennox Furnace Co of Canada, Ltd, Saskatchewan Ave, and Ste 23, Estelle Apts, Winnipeg, Man, Canada

ARMBRUSTER, Frank T. W. (M 1936), Sales Engr, Portsmouth Supply Co, 1532-34 Gallia St, Portsmouth, and (for mail), 105 First Ave, Waverly, Ohio

ARMSPACH, Otto W.* (M 1919), Chief Engr, Kroeschell Engineering Co, 215 W Ontario St, Chicago, and (for mail), 205 S Summit Ave, Villa Park, Ill

ARMSTRONG, Robert W. (S 1935), 2809 E Lake of the Isles Blvd, Minneapolis, Minn

ARNDT, Heinrich W (A 1935), Div Head, Plbg and Htg Dept, Sears Roebuck & Co, 732 Broad St, and (for mail), 214 Third St, Augusta, Ga

ARNOLD, Robert S (A 1926, J 1922), Pres, Lowell Air Conditioning Corp, Architects Bldg, Philadelphia, and (for mail), Wallingford, Pa

ARNOLDY, William F. (A 1930), Branch Mgr, Minneapolis-Honeywell Regulator Co, and (for mail), 415 Brainard St, Detroit, Mich

ARROWSMITH, John O. (M 1934), Plant Engr (for mail), Canadian Kodak Co, Ltd, and 9 Humberview Rd, Toronto 9, Ont, Canada

ARTHUR, John M., Jr. (M 1923), Supt Commercial Light & Steam Sales (for mail), Kansas City Power & Light Co, 1330 Baltimore, Kansas City, Mo, and 3311 State Ave, Kansas City, Kan.

ASHLEY, Carlyle M.* (M 1931), Dir of Development (for mail), Carrier Corp, 750 Frelinghuysen Ave, Newark, and 27 Oakley Ave, Summit, N J

ASHLEY, Edward E. (M 1912), Consulting Engr (for mail), 10 East 40th St, New York, N Y, and Middlesex Rd., Noroton Heights, Conn

ASTON, James (M 1919), Consulting Metallurgist (for mail), A M Byers Co, Clark Bldg, Pittsburgh, and 7315 Perrysville Ave., Ben Avon, Pa

ATHERTON, G. R. (M 1930), Air Cond Div (for mail), American Radiator Co, 40 West 40th St, New York, N Y

ATKINS, Thomas J. (M 1931), Mgr Air Cond Div, Carbondale Machine Corp, Harrison, and (for mail), 22 S. Munn Ave, East Orange, N J.

ATKINSON, Kenneth B. (J 1930), 5 Rose Valley Rd, Moylan, Rose Valley, Pa

AUGHENBAUGH, Harry E. (M 1935), Branch Mgr, York Ice Machinery Corp, 1238-46 North 44th St, Philadelphia, and (for mail), 7105 Penarth Ave, Upper Darby, Pa

AVERY, Lester T (M 1934), Pres (for mail), Avery Engineering Co, 2341 Carnegie Ave, Cleveland, and 21149 Colby Rd., Shaker Heights, Ohio

AXEMAN, James E. (M 1932; A 1931; J 1925), Branch Mgr (for mail), Spencer Heater Co., 1205 Court Square Bldg, and 908 Old Oak Rd, Stoneleigh, Baltimore, Md

AYERS, Earl H. (J 1935), Supt, D W Hickey & Co, 1631 University Ave, and (for mail), 543 S Warwick Ave, St Paul, Minn

B

BACHMAN, Fred (*M* 1936), Proprietor (for mail), 3001 North 21st St, Philadelphia, and 906 Bell Ave, Yeadon, Pa

BACKSTROM, Russell E (*A* 1931, *J* 1928), (for mail), Wood Conversion Co, First National Bank Bldg, and 513 S Snelling Ave, St Paul, Minn

BACKUS, Theodore H L (*M* 1916), Htg and Vtg (for mail), 200-208 Hill St, and 1018 Vaughn St, Ann Arbor, Mich

BADGETT, W Howard (*J* 1932), Research Asst, Texas Engineering Experiment Station, Agricultural and Mechanical College of Texas, P O Box 213 Faculty Exchange, College Station, Texas

BAHNSON, Frederic F. (*M* 1917), Vice-Pres and Chief Engr (for mail), The Bahnson Co, 1001 S Marshall St, Pres, Southern Steel Stampings, Inc, P O Box 1912, and 28 Cascade Ave, Winston Salem, N. C

BAILEY, Edward P., Jr (*M* 1925), Dist Repr, Iron Fireman Corp, 3170 West 106th St, Cleveland, Ohio, and (for mail), 151 Crocker Blvd, Mt Clemens, Mich

BAILEY, W Mumford (*M* 1930), Managing Dir, Mumford Bailey & Preston, Ltd, and Joint Managing Dir, British Trane Co, Ltd (for mail), "Newcastle House" Clerkenwell Close, London, E C 1, and "Oldbury Court," Dainesway, Thorpe Bay, Essex, England

BAIRD, S. Alan (*M* 1935), Consulting Engr (for mail), 621 Commercial Merchants National Bank Bldg, and 911 E Virginia Ave., Peoria, Ill

BAKER, C. T (*M* 1935), Consulting Engr, 713 Glenn St, S W, Atlanta, Ga

BAKER, George R. (*M* 1936), Pres. (for mail), G. R Baker Co, Ltd, 221 Adelaide St, W, and 37 Lappin Ave, Toronto, Canada.

BAKER, Harry L, Jr (*J* 1935), Sales Engr (for mail), American Blower Corp., 40 West 40th St, New York and 9935 Third Ave, Brooklyn, N. Y

BAKER, Howard C. (*M* 1921), Pres, The Howard C Baker Co, 128 S St Clair St, Toledo, Ohio

BAKER, Irving C. (*M* 1921), Gen Sales Mgr (for mail), Airtemp, Inc, 8021 Conant Rd, Detroit, and 1428 Yorkshire Rd, Grosse Pointe, Mich

BAKER, Richard H. (*S* 1936), 1353 West 93rd St, Cleveland, Ohio

BAKER, Roland H. (*M* 1928; *A* 1924), Pres (for mail), R. H Baker Co, 10 State St, Boston, and 420 Memorial Drive, Cambridge, Mass

BAKER, William H., Jr (*A* 1935), Air Cond Div. (for mail), American Radiator Co, 816 S. Michigan Ave., and 1211 N State St, Chicago, Ill

BALDI, Giuseppe (*A* 1936), Engr (for mail), Compagnia Italiana Westinghouse, via Pier Carlo Boggio 20 and Corso Racconigi, 39, Torino, Italy

BALDWIN, William H. (*M* 1921), Branch Mgr (for mail), C A Dunham Co, 2988 East Grand Blvd, and 2939 Calvert, Detroit, Mich.

BALL, William (*A* 1936), Pres (for mail), Interstate Heating & Plumbing Co, 521 Southwest Blvd, Kansas City, Mo, and 1700 West 37th Ave, Kansas City, Kan.

BALLANTYNE, George L. (*A* 1936), Mgr., Htg Sales Dept. (for mail), Crane, Ltd, P. O Box 840, and 4872 Cote Des Neiges, Montreal, P Q, Canada

BALSAM, Charles P. (*M* 1932), 324 Fourth St., Brooklyn, N Y

BAMOND, Manuel J. (*M* 1936), Engr, Reynolds Corp, 609 N LaSalle St, and (for mail), 4715 Magnolia Ave., Chicago, Ill.

BANNON, Lucas E. (*A* 1935), Archt, 18 Church St, and (for mail), 16 Church St, Paterson, N J

BARBIERI, Patrick J. (*J* 1936, *S* 1933), Asst Engr., Armo Cooling & Ventilating Co, 30 West 15th St., and (for mail), 2166 Belmont Ave, New York, N Y

BARNARD, M. Everett (*A* 1931, *J* 1929), Sales Engr (for mail), Carrier Corp, 12 South 12th St, and 341 Vernon Rd, Philadelphia, Pa.

BARNES, Arthur R (*M* 1921), Chief Engr (for mail), U S Supply Co, 1315 West 12th St, and 326 East 70th Terrace, Kansas City, Mo

BARNES, Harry P. (*A* 1936), Mgr, Construction Dept (for mail), Johns-Manville Sales Corp, 2030 Walnut St, and 6101 Walnut St, Kansas City, Mo

BARNES, Herbert (*M* 1936), Mgr (for mail), Herbert Barnes Plumbing & Heating, Delta Block, and 111 Grosvenor Ave, S, Hamilton, Ont, Canada

BARNES, Walter E. (*M* 1933), Pres, Barnes & Jones, Inc, 128 Brookside Ave, Jamaica Plain (Boston), and (for mail), 7 Woodlawn Ave, Wellesley Hills, Mass

BARNEY, William E. (*M* 1936), Mgr and Research Engr (for mail), Hydraulic-Press Brick Co, South Park, and 4929 East 108th St, Cleveland, Ohio

BARNS, Amos A. (*M* 1933), Owner (for mail), 410 W State St, and 318 W. State St, Ithaca, N Y

BARNSLEY, Frank R. (*A* 1936), Mgr, Air Cond Div (for mail), Canadian General Electric Co, Ltd, 1000 Beaver Hall Hill, and 5245 Byron Ave, Montreal, Que, Canada

BARNUM, Charles R. (*S* 1935), 1194 Capitol Ave, St Paul, Minn

BARNUM, Marvin C (*M* 1930; *A* 1928), Eastern Repr (for mail), Waterman-Waterbury Co, 1133 Broadway, New York, and Cherry Lane, Monsey, N Y

BARNUM, Willis E, Jr. (*M* 1933; *A* 1933, *J* 1930), Sales Engr, York Ice Machinery Co., 5051 Santa Fe Ave, Los Angeles, and (for mail), 3507 Santa Ana, South Gate, Calif

BARR, George W. (*M* 1905), (Board of Governors, 1910), Dist Mgr, Aerofin Corp, Land Title Bldg., Philadelphia, and (for mail), Woods End, Villa Nova, Pa

BARRY, James G., Jr. (*M* 1933), Vice-Pres (for mail), Elliott & Barry Engineering Co, 4060 W Pine Blvd, and 5051 Queens Ave, St Louis, Mo

BARRY, Patrick I. (*M* 1920), (Peace Commissioner), MIHVE (for mail), M Barry, Ltd, 4 Marlboro St, and Budaka, Sidney Park, Cork, Ireland

BARTH, Herbert E. (*M* 1920), Sales Mgr., American Blower Corp, 6000 Russell St., Detroit, Mich

BARTLETT, Amos C. (*M* 1919), Mgr., Htg and Vtg Dept (for mail), B F Sturtevant Co, Damon St., Hyde Park, Boston, and 30 Hollingsworth Ave, Braintree, Mass

BARTLETT, C Edwin (*M* 1922), Pres (for mail), Bartlett & Co., Inc, 3223 Arch St., and 37 W Phil Ellena St, Philadelphia, Pa

BARTON, Delbert H. (*J* 1936, *S* 1935), Box 44-B, Somerville, Texas

BASTEDO, Albert E. (*M* 1919), Vice-Pres-Treas-Mgr (for mail), Burnham Boiler Corp., Irvington-on-Hudson, and Burnside Drive, Hastings-on-Hudson, N Y

BAUER, Albert E. (*M* 1935), U S Air Conditioning Corp, N. W Terminal, Minneapolis, and (for mail), 59 S. Victoria St, St Paul, Minn.

BAUGHMAN, L. R. (*M* 1935), Htg. Engr, Helms & Baughman, 103 N Sheridan Rd, Waukegan, and (for mail), 2706 Eschol Ave, Zion, Ill.

BAUM, Albert L. (*M* 1916), Member of Firm, (for mail), Jaros, Baum & Bolles, 415 Lexington Ave, and 601 West 113th St, New York, N. Y

BAUMGARDNER, Carroll M. (*M* 1928), Branch Mgr (for mail), U S Radiator Corp, 3254 N. Kilbourn Ave, Chicago, and 602 Michigan Ave, Evanston, Ill

BAUR, John W. (*J* 1936; *S* 1935), 2517 Leland Ave, Chicago, Ill

BAYSE, Harry V. (*M* 1923), American Furnace Co, 2725 Morgan St, St. Louis, Mo

BEACH, Walter R. (*A* 1936), Sales Engr. (for mail), Cleveland Electric Illuminating Co, 75 Public Square, Cleveland, and 1185 Yellowstone Rd., Cleveland Heights, Ohio

BEAN, George S. (*A* 1935), Mgr, Stoker Div (for mail), North Western Fuel Co, E-1203 First National Bank Bldg, St Paul, and 4949-16th Ave, S, Minneapolis, Minn

BEARD, Earl L. (*J* 1936, *S* 1934), 819 Main St, San Antonio, Texas

BEAURRIENNE, Auguste[1] (*M* 1912), Consulting Engr, 25 Rue des Marguettes, Paris, France

BEAVERS, George R. (*M* 1929), Chief Engr, Canadian Blower & Forge Co, Ltd, Woodside Ave, and (for mail), 60 Church St, Apt 'D,' Kitchener, Ont, Canada

BECKER, Walter A (*M* 1935), Sales Engr, Grinnell Co, Inc, 4425 S Western Ave, and (for mail), 5531 N Artesian Ave, Chicago, Ill

BEEBE, Frederick E W (*A* 1915), Johnson Service Co, 28 East 29th St, New York, N Y

BEECHLER, Jack S (*J* 1937), Div Repr, Kelvinator Corp, 1714 Rhodes Haverty Bldg, Atlanta, Ga

BEERY, Clinton E (*M* 1913), Htg and Combustion Engr (for mail), Heat & Fuel Engineering Co, 646 N Michigan Ave, and 4110 N Paulina St, Chicago, Ill

BEGGS, William E. (*M* 1927), Pres, W E Beggs Co, 907 Lloyd Bldg, and (for mail), 3639 Palatine Ave, Seattle, Wash

BEIGHEL, Howard A. (*A* 1927), Sales Repr (for mail), Herman Nelson Corp, 503 Columbia Bldg, Pittsburgh, and 207 Puritan Rd, Rosslyn Farms, Carnegie, Pa

BEITZELL, Albert E. (*A* 1933, *J* 1930), Mgr, Air Cond. Div, Garnett, Eastman & Fleming, Inc, 600 N Second St, Harrisburg, and (for mail), 37 South 27th St, Camp Hill, Pa

BELING, Earl H[+] (*M* 1936, *A* 1930, *J* 1925), Proprietor, Beling Engineering Co, 2428-13th St, Moline, Ill

BELL, E. Floyd (*M* 1933), Branch Mgr (for mail), Buffalo Forge Co, 619 Foshay Tower, and 5337 Girard Ave, S, Minneapolis, Minn

BELT, Newton O. (*M* 1929), Engrg Dept, E I Du Pont de Nemours & Co, and (for mail), 824 West 10th St, Wilmington, Del

BEMAN, Myron C (*M* 1926), (Council, 1934-1936), Consulting Engr, (for mail), Beman & Candee, 374 Delaware Ave, and 699 Richmond Ave, Buffalo, N Y

BENNETT, Charles A. (*M* 1936), 1750 Harvard St, N W, Washington, D C

BENNETT, Edwin A. (*M* 1936, *A* 1936, *J* 1929), Sales Engr (for mail), American Blower Corp, 40 West 40th St, New York, and 45 Pondfield Rd W, Bronxville, N Y

BENNITT, George E. (*M* 1918), Consolidated Gas Co of N Y, 4 Irving Place, New York, N Y

BENOIST, LeRoy L (*M* 1934), Mgr (for mail), Benoist Bros Supply Co, 117 S Tenth St, and 1500 Main St, Mt Vernon, Ill

BENOIST, Raymond E. (*A* 1936), Mgr, Benoist Bros Supply Co, and (for mail), 811 North 12th St, Mt Vernon, Ill

BENSE, William M. (*J* 1936, *S* 1934), Engr, Institute of Thermal Research (for mail), American Radiator Co, 675 Bronx River Rd, Yonkers, and 340 Hayward Ave, Mt Vernon, N Y

BENSEN, Clarence L. (*J* 1935), Engr (for mail), McQuay, Inc, 1600 Broadway, N E, and 2722 Benjamin St, N E, Minneapolis, Minn

BENSINGER, Mark (*J* 1936), Sales Engr, Combustioneer Stoker Corp, 10th and D Sts, S W, and (for mail), 2737 Devonshire Place, Washington, D C

BENTZ, Harry (*M* 1915), Vice-Pres (for mail), Davis Engineering Corp, 1064 E Grand St, Elizabeth, and 18 Holland Terrace, Montclair, N J

BERCHTOLD, Edward W (*M* 1927, *A* 1925), Rate Engr (for mail), Boston Consolidated Gas Co, 100 Arlington St, Boston, and 20 Randolph St, S, Weymouth, Mass.

BERGHOEFER, Victor A (*M* 1936, *J* 1926), Vice-Pres, Sterling Engineering Co, 3738 N Holton St, and (for mail), 5446 N Kent Ave, Milwaukee, Wis

BERGLUND, Niels W (*A* 1936), Draftsman (for mail), York Ice Machinery Corp, 5051 Santa Fe Ave, and 729 S Bonnie Brae St, Los Angeles, Calif

BERMAN, Louis K (*M* 1908), Pres (for mail), Raisler Heating Co, 129 Amsterdam Ave, and 285 Central Park West, New York, N Y

BERMEL, Alfred H (*A* 1933, *J* 1928), Vice-Pres-Treas, O Vogelbach & Associate, Inc, Consulting Engrs, Chamber of Commerce Bldg, Newark, and (for mail), 16 William St, North Arlington, N J

BERNHARD, George (*M* 1935, *A* 1929), Managing Engr, Associated Heating Co, 16 Lafayette Ave, and (for mail), 985 Park Place, Brooklyn, N Y

BERNSTROM, Bert[1] (*M* 1930), Mech Engr, B Bernstrom Air Conditioning Consultant, 811 Rush St, Chicago, Ill

BERZELIUS, Carl E (*M* 1936), 1st Lt, Commanding Officer CCC Camp (for mail), Co 784 CCC, Neodesha, Kansas, and 1017 East 22nd St, Minneapolis, Minn

BEST, Millard W (*A* 1933), Pres (for mail), Kolelectric Underfeed Stoker Co, Ltd, 245 Kenilworth Ave S, and 1750 King St E, Hamilton, Ont, Canada

BETLEM, Henriette T. (*J* 1934), Air Cond Engr (for mail), Betlem Heating Co, 1926 East Ave, and 1293 Park Ave, Rochester, N Y

BETTS, Howard M. (*M* 1927), Senior Mech Engr, Htg & Vtg (for mail), Dept of Buildings, 213 City Hall, and 4923 Russell Ave S, Minneapolis, Minn

BETZ, Harry D (*M* 1928), Pres (for mail), Betz Unit Air Cooler Co, 6 W Ninth St, and 4210 Mercer, Kansas City, Mo

BEVINGTON, Curtis H (*M* 1936), Mfgr Repr (for mail), James P Marsh Corp, 720 N Michigan Ave, Chicago, and 615 Cedar St, Park Ridge, Ill

BIANCULLI, Vincent A (*J* 1937), Draftsman, S A S Patorno, 101 Park Ave, and (for mail), 557 Broome St, New York, N Y

BICHOWSKY, F. Russell (*M* 1935), Engr, Air Cond Div (for mail), Surface Combustion Corp, Toledo, and 3421 Indian Rd, Ottawa Hills, Ohio

BINDER, Charles G (*M* 1920), Mgr Htg Dept, Warren Webster & Co, 17th and Federal Sts, Camden, and (for mail), 115 Oak Terrace, Merchantville, N J

BINFORD, Wilmer M (*A* 1936, *J* 1930), Mgr Contract Dept, So Div (for mail), 2120 East 25th St, and 6215 San Vicente Blvd, Los Angeles, Calif

BIRD, Charles (*A* 1934), Treas-Gen Mgr (for mail), Doermann-Roehrer Co, 450-456 E Pearl St, and Box 179-D, R R 6, Section Rd, Cincinnati, Ohio

BIRRELL, Allan L (*A* 1925), Consulting Engr, 372 Bay St, Toronto 2, and (for mail), 93 Kingsway, Toronto 9, Ont, Canada

BISCH, Bernard J (*M* 1931), Engr, St Mary-of-the-Woods College, St Mary-of-the-Woods, Ind

BISHOP, Charles R. (*Life Member, M* 1901), 413 Locust St, Lockport, N Y

BISHOP, Frederick R (*M* 1921), Mfgrs Agent, 8011 Dexter Blvd, Detroit, Mich

BISHOP, Marion W (*J* 1935), Sales Engr (for mail), American Blower Corp, 228 N LaSalle St, and 6317 N Kenmore Ave, Chicago, Ill

BLACK, Edgar N., 3rd (*M* 1922), Philadelphia Mgr, Fitzgibbons Boiler Co, Inc, 1215-6 Land Title Bldg, Philadelphia, and (for mail), 111 Woodside Rd, Haverford, Montgomery Co, Pa

BLACK, F. C (*M* 1919), Pres (for mail) F C Black Co, 622 W Randolph St, and 4535 N Ashland Ave, Chicago, Ill

BLACK, Harry G (*M* 1917), Prop (for mail), P Gormly Co, 155 N Tenth St, and 927 North 65th St, Philadelphia, Pa

BLACK, William B. (*J* 1932), 1017 W Second St, Oil City, Pa
BLACKBURN, Edwin C, Jr. (*M* 1929), Consulting Engr, Crow, Lewis & Wick, Archts, 200 Fifth Ave, New York, and (for mail), 5 Kenwood Rd, Garden City, L I, N Y
BLACKHALL, Lewis C (*M* 1935), Engr, Gurney Foundry Co, Ltd, 4 Junction Rd, and (for mail), 231 Brock Ave, Toronto, Ont, Canada
BLACKHALL, Wilmot R (*M* 1922), Partner (for mail), McKellar & Blackhall, 1101 Bay St, and 332 Waverly Rd, Toronto, Canada
BLACKMAN, Alfred O (*M* 1911), Consulting Engr, 150 West 24th St, New York, N Y
BLACKMORE, F H (*M* 1923), Mgr, Operating Dept (for mail), U S Radiator Corp, Box 686, Detroit, and 515 Tooting Lane, Birmingham, Mich
BLACKMORE, George C (*Charter Member, Life Member*), Pres (for mail), Automatic Gas Steam Radiator Co, 301 Brushton Ave, and Cathedral Mansions, Pittsburgh, Pa
BLACKMORE, J. J* (*Charter Member, Life Member*), 32 West 40th St, New York, N Y
BLACKMORE, James S (*J* 1931), Factory Sales Engr, H A Thrush & Co, Peru, Ind, and (for mail), 330 E Manoa Rd, Upper Darby, Pa
BLACKSHAW, J L.* (*J* 1929), Engr, Air and Refrigeration Corp, 11 West 42nd St, New York, and (for mail), 59 Joralemon St, Brooklyn, N Y
BLAIR, Howard A (*J* 1935), Air Cond Service Engr, Westinghouse Electric & Mfg Co, 500 E Fifth St, and (for mail), 71 Stewart Ave, Mansfield, Ohio
BLAKELEY, Hugh J (*M* 1935), Consulting Engr (for mail), Hubbard, Rickerd & Blakeley, 1100 Chapel St, New Haven, and 5 Doty Place, Morris Cove, Conn
BLAKESLEE, Donald (*A* 1935), Pres (for mail), Donald Blakeslee, Inc, 89 E Main St, Patchogue, and Main Rd, Bellport, N Y
BLANDING, George H (*M* 1919), Salesman, Johnson Service Co, 1355 W Washington Blvd, Chicago, and (for mail), 729 Hayes Ave, Oak Park, Ill
BLANKIN, Merrill F. (*M* 1927, *A* 1926, *J* 1919), Pres (for mail), Haynes Selling Co, Inc, S E Cor Ridge Ave and Spring Garden St, and 3328 W Penn St, Philadelphia, Pa
BLAS, Romualdo J. (*M* 1936), Chief Engr, Air Cond Dept, Sanchez & Co, Apartado Postal 1006 Caracas, Venezuela
BLESSED, William A. (*A* 1935), Sales Engr (for mail), Mueller Brass Co, 1925 Lapeer Ave, and 1107 Court St, Port Huron, Mich
BLISS, George L (*A* 1933), Engr and Sales (for mail), Allis-Chalmers Mfg Co, 11th and Main St., Room 1410 Waldheim Bldg, and 7641 Brooklyn Ave, Kansas City, Mo
BLOOM, Louis (*M* 1935), Partner, B Bloom & Son, and (for mail), 1450–52nd St, Brooklyn, N Y
BLUM, Herman, Jr (*J* 1936), Junior Engr for Airtemp Distr, Crawford, Inc, 1008 Union Bldg, Gravier and Baronne Sts, and (for mail), 6131½ Hurst St, New Orleans, La
BLUMENTHAL, Moritz I. (*M* 1936), Engrg Instructor, National Schools, 4000 S Figueroa St, Los Angeles, and (for mail), 7964 Fountain Ave, Hollywood, Calif
BOALES, William G. (*M* 1935; *A* 1923), Mgr (for mail), Wm. G. Boales Co, 6439 Hamilton Ave, Detroit, and 195 McMillan Rd, Grosse Pointe Farms, Mich
BOCK, Bernard A. (*A* 1929, *J* 1927), Mech Engrg Draftsman, 57 Elizabeth Ave, Arlington, N J.
BOCK, I. I. (*A* 1934), Sales Engr (for mail), Carrier Corp, 2022 Bryant St, and 2500 South Blvd, Dallas, Texas
BODDINGTON, William P. (*M* 1927), Mgr (for mail), Canadian Powers Regulator Co, Ltd, 106 Lombard St, and 280 Clendenan Ave, Toronto, Ont, Canada
BODINGER, Jacob H. (*M* 1931), Pres (for mail), Bodinger & Co, Inc, 530 Tenth Ave, New York, and 1429 East 10th St, Brooklyn, N Y
BOEHMER, Andrew P. (*S* 1935), Refrigeration Testing Engr, Mills Novelty Co, 4100 Fullerton Ave, and (for mail), 3012 N Kostner Ave, Chicago, Ill
BOESTER, Carl F. (*A* 1936), Air Cond Engr (for mail), City Ice & Fuel Co, 3038 Olive St, St Louis, and 442 R R 13, Kirkwood, Mo
BOGATY, Hermann S. (*M* 1921), 5230 North 15th St, Philadelphia, Pa
BOLSINGER, Raymon C. (*M* 1916), (Council, 1936), Prop, Automatic Florzone Heating Co, Conshohocken, Pa, and (for mail), 238 E Madison, Ave, Collingswood, N. J
BOLTE, E Endicott (*A* 1929), Salesman, National Radiator Corp, 1111 East 83rd St, and (for mail), 6516 Kenwood Ave, Chicago, Ill
BOLTON, Reginald P.* (*Honorary Member; Life Member*), (*Presidential Member*), (Pres, 1911, 1st Vice-Pres, 1905-1910, 2nd Vice-Pres, 1903; Board of Governors, 1901, 1905, 1910, 1911, 1912, 1913), The R P Bolton Co, 116 East 19th St, New York, N Y
BOND, Horace A. (*M* 1930), Dist Mgr, Warren Webster & Co, 152 Washington Ave, and (for mail), 12 Ramsey Place, Albany, N Y
BONTHRON, Robert C. (*A* 1935), Supervisor Air Cond Dept, Westinghouse Electric & Mfg Co, Room 1208–150 Broadway, New York, N Y
BOOTH, Charles A. (*M* 1917), Vice-Pres (for mail), Buffalo Forge Co, 490 Broadway, and 142 Summit Ave, Buffalo, N Y
BOOTH, Harry N (*M* 1924, *A* 1917), Vice-Pres Sales Dept (for mail), U S Radiator Corp, Room 1056 First National Bank Bldg., and 2916 Seminole Ave, Detroit, Mich
BORKAT, Philip (*J* 1936), Sales Engr, Trane Co, and (for mail), 813 East 100th St, Cleveland, Ohio
BORLING, John R. (*A* 1934), Engr-Custodian, Chicago Board of Education, 6520 S Wood St, and (for mail), 953 East 84th Place, Chicago, Ill
BORNEMANN, Walter A (*M* 1924, *J* 1923), Sales Engr (for mail), Carrier Corp, 12 South 12th St, Philadelphia, and 123 W Wharton Ave, Glenside, Pa
BORUCH, Edwin R (*A* 1935), Power Salesman (for mail), Dallas Power & Light Co, 1506 Commerce St, and 835 N Bishop, Dallas, Texas
BOUCHERLE, Henry N (*M* 1931), Secy (for mail), The Scholl-Choffin Co, Mahoning Ave, and Hogue St, and 3412 Hudson Ave, Youngstown, Ohio
BOUEY, Angus J (*J* 1930), Sales Engr (for mail), B F Sturtevant Co, 553 Monadnock Bldg, and 4810 Fulton St, San Francisco, Calif
BOUILLON, Lincoln (*M* 1933), Consulting Engr (for mail), 1411 Fourth Ave Bldg, and 2211–32nd South, Seattle, Wash
BOWDITCH, Robert P. (*J* 1936, *S* 1935), 504 S Race St, Urbana, Ill.
BOWERS, Arthur F (*A* 1919), Pres, Industrial Heating & Engineering Co, 828 N Broadway, Milwaukee, Wis
BOWLES, Potter (*A* 1928), Pres (for mail), Hoffman Specialty Co, Inc, Room 3324–500 Fifth Ave, New York, and 678 Ely Ave, Pelham Manor, N Y
BOWMAN, James W. (*J* 1936, *S* 1934), 210 S Santa Fe, Norman, Okla
BOYDEN, Davis S.* (*M* 1909), (1st Vice-Pres, 1936; Treas, 1933-1934, Council, 1917-1930-1936), Supt, Steam Htg Service Dept (for mail), Edison Electric Illuminating Co of Boston, 39 Boylston St, Boston, and 1496 Commonwealth Ave, Brighton, Mass
BOYKER, Robert O. (*J* 1935), Contractor, Mac Boyker & Son, Kent, Wash
BOYLE, John R. (*M* 1936), Asst Chief Engr, Westerlin & Campbell Co, 1113 Cornelia Ave, and (for mail), 6858 Osceola Ave, Chicago, Ill.

BOZEMAN, Richard W (*M* 1936, *J* 1929), 450 S Ashland Ave, Lexington, Ky
BRAATZ, Chester J.[^] (*M* 1930), Sales Mgr, Temperature Control Dept, Barber-Colman Co, and (for mail), 718 King St, Rockford, Ill
BRABBEE, Dr Charles W [†] (*M* 1925), (for mail), American Radiator Co, 40 West 40th St, New York, and 50 Lincoln Ave, Tuckahoe, N Y
BRACKEN, John H. (*M* 1927), Mgr, Industrial Uses Dept (for mail), Celotex Co, 919 N Michigan Ave, Chicago, Ill
BRADFIELD, William W. (*M* 1926), Consulting Engr (for mail), 901 Michigan Trust Bldg, and 1352 Franklin St S E, Grand Rapids, Mich
BRADFORD, Gilmore G (*M* 1936), China Repr, Frick Co (for mail), c o American Club, and 176 Route Defour, Shanghai, China
BRADLEY, Eugene P. (*M* 1906), Pres (for mail), Hester-Bradley Co, 2835 Washington Ave, and 6935 Pershing Ave, St Louis, Mo
BRANDI, O H (*M* 1930), Dipl Ing, Rud Otto Meyer, Hamburg 23, Pappelallee 23-39, and (for mail), Altona, Gr, Flottbek, Gottorpstr 1, Germany
BRANDT, Ernst H., Jr. (*M* 1928), Pres, Reliance Engineering Co, Inc, 515 N Church St, and (for mail), P O. Box 292, Charlotte, N. C
BRAUER, Roy (*M* 1926), Prop (for mail), Ventilating Equipment Co, 504 Magee Bldg, and 576 Austin Ave, Mt Lebanon, Pittsburgh, Pa
BRAUN, John J. (*M* 1932), Factory Mgr, U S Playing Card Co, Norwood Station, Cincinnati, and (for mail), 4305 Floral Ave, Norwood, Ohio
BRAUN, Louis T. (*M* 1921), Executive Secy (for mail), Chicago Master Steamfitters Association, 228 N LaSalle St, and 1548 Pratt Blvd, Chicago, Ill
BRECKENRIDGE, L. P * (*Honorary Member, Life Member, M* 1920), The Brackens, North Ferrisburg, Vt
BREDESEN, Bernhard P. (*A* 1931), Engr. (for mail), Reese & Bredesen, 410 Essex Bldg, and 3319 Knox Ave N, Minneapolis, Minn
BRENEMAN, Robert B. (*A* 1931, *J* 1927), Sales Engr (for mail), Armstrong Cork & Insulation Co., 191 Orchard Lane, Columbus, Ohio
BRENNAN, John W. (*M* 1935, *A* 1934), Salesman (for mail), American Blower Corp, 632 Fisher Bldg, and 15423 Gilchrist, Detroit, Mich
BREZINA, Edwin A. (*S* 1936), 3731 East 131st St., Cleveland, Ohio.
BRIDE, William T. (*M* 1928, *J* 1925), Supt Engrg, Bride-Grimes & Co, 9 Franklin St (for mail), P O Box 777 Lawrence, and 50 High St., Methuen, Mass
BRIGHAM, Clare M. (*M* 1935), Vice-Pres in charge of sales (for mail), C A Dunham Co, 450 E Ohio St, Chicago, and 420 Maple Ave, Winnetka, Ill
BRIGHAM, Frederick H. (*M* 1930), Sales Engr, G H Gleason & Co, 25 Huntington Ave, Boston, and (for mail), 39 Woodside Rd, Winchester, Mass.
BRIGHTLY, Frederick C, Jr. (*A* 1936), Vice-Pres, Standard Galvanizing Co, 2619 W Van Buren St, Chicago, and (for mail), 1233 S. Home Ave, Berwyn, Ill
BRINKER, Harry A. (*M* 1934), 524 Village St, Kalamazoo, Mich
BRINTON, Joseph W. (*M* 1920), Dist Mgr (for mail), American Blower Corp, 1003 Statler Bldg, Boston, and 42 Gleason St, West Medford, Mass
BRISSETTE, Leo A. (*M* 1930), Treas (for mail), Trask Heating Co, 4 Merrimac St, Boston, and 168 Florence St, Melrose, Mass
BROCHA, John F. (*M* 1936), Buyer of Plbg and Htg, Montgomery Ward & Co., 619 W Chicago Ave, and (for mail), 1433 N Linder Ave, Chicago, Ill
BRODERICK, Edwin L.* (*M* 1933), Research Asst in Mech. Engrg (for mail), University of Illinois, 213 M E Lab, Urbana, and 1203 W Park St, Champaign, Ill
BRONSON, Carlos E [†] (*M* 1919), Mech Engr (for mail), Kewanee Boiler Corp, and 311 McKinley Ave, Kewanee, Ill
BROOKS, Frank W (*S* 1934), 935 N Broadway, Dayton, Ohio
BROOM, Benjamin A (*M* 1914), Sales Promotion Engr., Weil-McLain Co, 641 W Lake St, and (for mail), 1544 Sherwin Ave, Chicago, Ill
BROOME, Joseph H (*A* 1936), Sales Engr, Minneapolis-Honeywell Regulator Co, 801 Second Ave, New York, and (for mail), 1556 Pacific St, Brooklyn, N Y
BROWN, Alfred P (*M* 1927), Vice-Pres (for mail), Reynolds Corp, 609 N LaSalle St, Chicago, and 551 Hill Terrace, Winnetka, Ill
BROWN, Aubrey I [†] (*M* 1923), Prof of Htg & Vtg (for mail), Ohio State University, and 169 Richards Rd, Columbia, Ohio
BROWN, David (*M* 1936), Owner (for mail), 67 Cooper Square, and 54 West 174th St, New York, N Y.
BROWN, Foskett* (*M* 1926), Vice-Pres (for mail), Gray & Dudley Co, 222 Third Ave N, P O Box 722, and 2314 West End Ave, Nashville, Tenn
BROWN, Mack D (*J* 1936), Mech Draftsman (for mail), Northrup & O'Brien, Archts, 602-03 Reynolds Bldg, and 915 East 21st St, Winston-Salem, N C
BROWN, Norman A (*J* 1936, *S* 1935), 1723 West 19th St, Cicero, Ill
BROWN, Tom (*M* 1930), Vice-Pres-Gen Mgr (for mail), Autovent Fan & Blower Co, 1805-27 N Kostner Ave, and 5325 N Laramie, Chicago, Ill
BROWN, William (*A* 1937), Vice-Pres-Gen Mgr (for mail), Carey Co, 6197 Hamilton Ave, and 660 Virginia Park, Detroit, Mich
BROWN, William A (*M* 1930), Mech Engr, U S Government, and (for mail), 5507 Carolina Place, N W, Washington, D C
BROWN, William H. (*A* 1923), Mgr, Brown Bros, Inc, 3015 North 22nd St, Milwaukee, Wis
BROWN, W. Maynard (*A* 1930), Warren Webster & Co, 17th and Federal Sts, Camden, N J
BROWNE, Alfred L. (*M* 1923), 253 Highland Ave, South Orange, N J
BRUEGGEMAN, Arthur R. (*M* 1920), (for mail), Erie Engineering Co, 1740 East 12th St, Cleveland, and 17220 Aldersyde Drive, Shaker Heights, Ohio
BRUNETT, Adrian L. (*M* 1923), Assoc Mech Engr, U S Supervising Architects Office, Treasury Dept, Washington, D C., and (for mail), P O Box 36, Rockville, Md
BRUST, Otto (*M* 1930), Consulting Engr, Lufttechnische Gesellschaft, Ing Broz & Co, Praha 1. Revolucni 13, and (for mail), Praha, VII, Veverkova 3, Cechoslovakia
BRYANT, Dr. Alice G. (*Life Member, M* 1921), 405 Marlborough St., Boston, Mass
BRYANT, Percy J. (*M* 1915), Chief Engr (for mail), Prudential Insurance Co of America, 763 Broad St, Newark, and 754 Belvidere Ave, Westfield, N J
BUCK, David T. (*A* 1936), Pres (for mail), Buck Engineering Co, Inc, 37-41 Marcy St, and 116 W Main St., Freehold, N J.
BUCK, Lucien (*M* 1928), Engr, Proctor & Schwartz, Inc, Seventh St. and Tabor Rd., Philadelphia, and (for mail), 8210 Westminster Rd, Elkins Park, Pa.
BUENGER, Albert* (*M* 1920, *J* 1917), (Council, 1934-1936), Educational Dept. (for mail), Delco-Frigidaire Conditioning Corp, 1420 Wisconsin Blvd, and 224 Schantz Ave, Oakwood, Dayton, Ohio.
BUENSOD, Alfred C. (*M* 1918), Pres., Buensod Stacey Air Conditioning, Inc, 60 East 42nd St., and (for mail), 1 Fifth Ave, New York, N Y
BULLEIT, Charles R (*M* 1932, *J* 1930), 1811 Bayard Park Drive, Evansville, Ind
BULLOCK, Howard H. (*A* 1933), Commercial Engr (for mail), General Electric Co, 5201 Santa Fe Ave, Los Angeles, and 2442 Cudahy St, Huntington Park, Calif.

BULLOCK, Thomas A (*M* 1930), Engr (for mail), Densmore LeClear & Robbins, 31 St James Ave, Boston, and 35 Everett St, Arlington, Mass

BUOT, Antonio V (*A* 1936, *S* 1935), 2606 Humboldt Ave S, Minneapolis, Minn

BUR, Julien R. C (*A* 1936, *J* 1931), Chief Engr (for mail), Bur & Co, 10 rue du Chapeau Rouge, and 1 Place Francois Rude, Dijon, France

BURBAUM, W. Allen (*J* 1933), 180 Clinton Ave, Brooklyn, N Y

BURCH, Laurence A. (*M* 1934), Sales Mgr, R L Deppmann Co, 957 Holden Ave, and (for mail), 17173 Kentucky Ave, Detroit, Mich

BURKE, James (*J* 1930), Engr, Carrier Corp, 12 South 12th St, Philadelphia, Pa

BURKE, William J. (*A* 1934), 2375 S W Ninth St, Miami, Fla

BURKHART, Elder M (*J* 1935), Sheet Metal Estimator, Overly Mfg Co, 571 W Otterman St, and (for mail), 22 Westminster Ave, Greensburg, Pa

BURKS, Roland H (*J* 1936, *S* 1935), 123 Yale Ave, San Antonio, Texas

BURNETT, Earle S (*M* 1920), Mech Engr, Amarillo Helium Plant, U S Bureau of Mines, and (for mail), 1223 West 11th Ave, Amarillo, Texas

BURNS, Edward J. (*M* 1923), Harris Bros Plumbing Co., 217 W Lake St, and (for mail), 1716 Aldrich Ave, Minneapolis, Minn

BURNS, John R. (*J* 1936, *S* 1933), Htg Engr, Delius Co, 13 N Main St, and (for mail), 504 N Main St, Wallingford, Conn

BURNS, Robert (*M* 1934), Sales Engr, Sears Roebuck & Co, 200 E Ohio St, and (for mail), 482 Antenor Ave, Pittsburgh, Pa

BURR, Kimball (*A* 1936), Mgr Air Cond Div (for mail), American Radiator Co., 40 West 40th St, New York, and Ardsley Park, Dobbs Ferry, N Y

BURRITT, Charles G (*A* 1916), Branch Mgr (for mail), Johnson Service Co, 922 Second Ave. S, and 615 Second Ave S, Minneapolis, Minn

BUSHNELL, Carl D. (*A* 1921), Pres (for mail), Bushnell Machinery Co, 311 Ross St, Pittsburgh, and 94 Pilgrim Rd, Rosslyn Farms, Carnegie, Pa.

BUTLER, Peter D (*M* 1922), Salesman, U S Radiator Corp, Detroit, Mich, and (for mail), 127 Edgewater Rd, Cliffside Park, N J

BUTT, Roderick E W (*A* 1936, *J* 1930), Air Cond Engr., Frigidaire, Ltd., The Hyde, Hendon, London, N W 9 and (for mail), "Hive Corner," The Common Rd, Stanmore, Middlesex, England

BUTTARAVOLI, Frank (*S* 1935), 1820 East 19th St, Brooklyn, N Y

BUTTS, Robert L. (*J* 1936, *S* 1935), 64th and Normandale, Minneapolis, Minn

BYRD, Tom (*A* 1936), Salesman (for mail), American Rolling Mill Co, 703 Curtis St., and 2403 Fleming Rd., Middletown, Ohio

C

CAIRNS, John H. (*A* 1936), Asst Sales Engr., Frigidaire Corp, and (for mail), 127 Scarboro Rd., Toronto, Ont, Canada.

CALDWELL, Arthur C. (*M* 1930), Engr and Estimator, P. Gormly Co, 155 N Tenth St., and (for mail), 550 South 48th St, Philadelphia, Pa

CALEB, David (*M* 1923), Eng (for mail), Kansas City Power & Light Co, 1330 Baltimore Ave, and 141 Spruce St., Kansas City, Mo

CALL, Joseph (*J* 1936), Air Cond Engr, Elliott-Lewis Co., 2518 N Broad St., Philadelphia, and (for mail), 669 Jamestown St, Roxborough, Philadelphia, Pa

CALLAHAN, Peter J. (*M* 1934), Inspecting Engr, Central Hanover Bank & Trust Co, 60 Broadway, New York, and (for mail), 4057 Amboy Rd., Great Kills, S I, N Y

CAMERON, William R. (*A* 1936), Dist. Mgr, L J Mueller Furnace Co, Milwaukee, Wis, and (for mail), 3337 Highland Ave, Kansas City, Mo.

CAMPBELL, Alfred Q, Jr (*J* 1933), Field Clerk, Inspection Div, Federal Emergency Administration of Public Works, 938 Poplar Ave, and (for mail), 1083 Meriwether Ave, Memphis, Tenn

CAMPBELL, Everett K.* (*M* 1920), Pres-Treas (for mail), E. K Campbell Heating Co, 2445 Charlotte St, and 3717 Harrison Blvd, Kansas City, Mo

CAMPBELL, E. Kirker, Jr. (*J* 1930), Secy (for mail), E K Campbell Heating Co, 2445 Charlotte St, and 3717 Harrison Blvd, Kansas City, Mo

CAMPBELL, Frank B (*A* 1927), Sales Engr, American Radiator Co, 40 West 40th St, New York, N Y

CAMPBELL, Robert E (*J* 1935, *S* 1934), Htg Engr (for mail), Brooklyn Borough Gas Co, Mermaid Ave. and West 17th St, Coney Island and 1993 East 28th St, Brooklyn, N Y

CAMPBELL, Thomas F. (*M* 1928), Distributor, Minneapolis-Honeywell Regulator Co, 1013 Penn Ave, Wilkinsburg, Pa

CANDEE, Bertram C. (*M* 1933), Partner, Beman & Candee, 374 Delaware Ave, Buffalo, and (for mail), 19 Tremont Ave, Kenmore, N Y

CAREY, James A (*M* 1928), Carrier Corp, Newark, N J, and (for mail), Villa Nova, Pa

CAREY, Paul C. (*M* 1930), Consulting Engr (for mail), Runyon & Carey, 33 Fulton St, Newark, and 31 Claremont Drive, Maplewood, N J.

CARLE, William E (*M* 1926), Pres (for mail), Carle, Boehling Co, Inc, 1611 W Broad St, and 2220 Floyd Ave, Richmond, Va

CARLOCK, Marion F. (*M* 1936), Engr J H. Walters Co, 610 E Broadway, and (for mail), 505 Henry, Alton, Ill

CARLSON, Everett E. (*M* 1932, *A* 1929), Branch Mgr (for mail), Powers Regulator Co, 1010 Louderman Bldg, and 6652 Washington Ave, St Louis, Mo

CARPENTER, R H. (*M* 1921), (Council, 1930-1935), Mgr, New York Office (for mail), Nash Engineering Co, Graybar Bldg, 420 Lexington Ave, New York, and 20 Jefferson Ave, White Plains, N Y

CARR, Maurice L. (*M* 1931), Director (for mail), Pittsburgh Testing Laboratory, P O. Box 1646, and Webster Hall, Pittsburgh, Pa

CARRIER, Earl G. (*M* 1936; *J* 1929), Estimating Engr, Carrier South Africa (Pty), Ltd, 20 Beresford House, Simmonds St, Johannesburg Transvaal, Union of South Africa

CARRIER, Willis H.* (*M* 1913), (*Presidential Member*), (Pres, 1931, 1st Vice-Pres, 1930, 2nd Vice-Pres, 1929, Council, 1923-1932), Chairman of the Board (for mail), Carrier Corp, 850 Frelinghuysen Ave, Newark, and Rensselaer Rd., Essex Fells, N J

CARTER, Alexander W. (*J* 1936), Heating Engr (for mail), Monarch Brass Mfg Co, Ltd, 71 Browns Ave, and 105 Victoria Park Ave, Toronto, Canada

CARTER, Doctor (*M* 1934), Heating and Sanitary Engr (for mail), Room 415, 410 Szechuen Rd. and Lane 175, House 9, Tunsin Rd, Shanghai, China

CARTER, John H. (*M* 1936), Special Repr (for mail), Frick Co, 100 N Broadway, St Louis, and 529 Atalanta Ave, Webster Groves, Mo

CARY, Edward B. (*M* 1935), Vice-Pres (for mail), John Paul Jones, Cary & Millar, Inc, 448 Terminal Tower, Cleveland, and 3549 Daleford Rd, Shaker Heights, Ohio

CASE, Roy H. (*A* 1936), Resident Mgr (for mail), 431 Lyon Bldg., and 3322 Hunter Blvd, Seattle, Wash.

CASE, Walter G. (*A* 1930), Tech Mgr, Ideal Boilers & Radiators, Ltd, Ideal House, Great Marlborough St, London, W I and, (for mail), 66 The Ridgeway, Kenton, Harrow, Middlesex, England

CASEY, Byron L (*M* 1921), Sales Engr (for mail), Ilg Electric Ventilating Co, 182 N LaSalle St, Chicago, and 515 N Park Ave, Park Ridge, Ill

CASEY, Huntley F. (M 1931), Mech Engr, Treasury Dept, Proc Div, Federal Warehouse, and (for mail), 1843 Irving St N W, Washington, D C

CASH, Tidie T. (A 1925), Mgr (for mail), Grinnell Co, Inc, 240 Seventh Ave S, and 617 Kenwood Parkway, Minneapolis, Minn

CASPERD, Henry W. H (J 1930), Engr, Carrier Co, Ltd, 24 Buckingham Gate, London, and (for mail), 21 Robin Hood Lane, Sutton, Surrey, England

CASSELL, John D* (Life Member, M 1913), Retired (for mail), 2008 Walnut St, Philadelphia, Pa, and 740 Garfield Ave, Palmyra, N J

CASSELL, William L (M 1936), Mech Engr (for mail), 2301 Telephone Bldg, Kansas City, and R F D No 6, Independence, Mo

CAWBY, Elmer L. (J 1935), Sales Engr (for mail), Carrier Corp, 748 E Washington Blvd, and 2315 S Flower St, Los Angeles, Calif

CHAMBERS, Fred W (M 1936), Mgr and Chief Engr (for mail), F W Chambers & Co, 96 Bloor St W, Toronto 5, and 122 Garfield Ave, Toronto, Ont, Canada

CHAPIN, C. Graham (M 1933), Treas (for mail), Hopson & Chapin Mfg Co, 231 State St, and 65 Faire Harbour Place, New London, Conn

CHAPIN, Harvey G. (M 1935), Sales Engr (for mail), Westerlin & Campbell Co, 1113-23 Cornelia Ave, and 8136 Ingleside Ave, Chicago, Ill

CHAPMAN, William A, Jr. (M 1936), Dist Engr (for mail), Frigidaire Corp, 3414 Lindell Blvd, St Louis, and 27 Willow Hill Rd, McKnight Village, St Louis County, Mo

CHARLES, Thomas J. (M 1934), 175 Marine Ave, Brooklyn, N Y

CHARLET, Louis W (M 1934), Mgr, N Y Branch (for mail), Kewanee Boiler Corp, 37 West 39th St, New York, and 427 Rich Ave, Mt Vernon, N Y

CHASE, Chauncey L (M 1931), Partner (for mail), Edward E Ashley, Consulting Engr, 10 East 40th St, New York, and 8829 Fort Hamilton Parkway, Brooklyn, N Y.

CHASE, Louis R (J 1931), Dist Supervisor (for mail), Carter-Waters Corp, 2440 Pennway, and 222 West 68th St, Kansas City, Mo

CHEATWOOD, William H (J 1937), Commercial Engr (for mail), Interstate Electric Co, 300 Spring St, and c/o Y M C A, Shreveport, La

CHEESEMAN, Evans W. (S 1934), Carnegie Institute of Technology, Pittsburgh, Pa

CHEN, Sarcey T. (M 1936), Director and Partner (for mail), American Engineering Corp, 989 Bubbling Well Rd, and 122 Route Frelupt, Shanghai, China

CHENOWETH, Dale M. (S 1936), (for mail), 152 S Grant St, West Lafayette, and 415 W Main St, Portland, Ind

CHERNE, Realto E. (J 1929), Engr, Carrier Corp., 850 Frelinghuysen Ave, Newark, and (for mail), 47 Elm St, Elizabeth, N J

CHERRY, Lester A* (M 1921), Consulting Engr (for mail), Industrial Planning Corp, 271 Delaware Ave, and 155 Euclid Ave, Buffalo, N Y

CHESTER, Thomas* (M 1917), Consulting Engr, c/o Davidson & Co, Ltd, Central House, Kingsway, London, England

CHEYNEY, Charles C. (A 1913), Asst Sales Mgr. (for mail), Buffalo Forge Co, 490 Broadway, and 255 Lincoln Parkway, Buffalo, N Y.

CHIPPERFIELD, W. H (A 1934), Service Engr., Walker-Crosweller Co, Ltd, 20 Queen Elizabeth St, S.E 1, and (for mail), 54 Lankers Drive, North Harrow, Middlesex, England

CHOFFIN, C. C. (M 1919), Pres-Treas (for mail), Scholl-Choffin Co, Mahoning Ave and Hogue St, and 560 Tod Lane, Youngstown, Ohio

CHRISTENSON, Harry (A 1931), Secy-Treas. (for mail), Hunter-Prell Co, 38 S Madison St., and R F. D No 1, Battle Creek, Mich

CHRISTIE, Alfred Y (A 1933), Salesman, U S Radiator Corp, 233 Vassar St, Cambridge, and (for mail), 715 LaGrange St, West Roxbury, Mass

CHRISTMAN, William F (A 1932, J 1931), Engr (for mail), Kroeschell Engineering Co, 215 W Ontario St, and 2803 Lunt Ave, Chicago, Ill

CHRISTOPHERSEN, Andrew E. (M 1935), Engr-Custodian (for mail), Board of Education, Franklin & Waller Br High Schools, 226 W Goethe St, and 2923 N Kilpatrick Ave, Chicago, Ill

CHROUCH, Richard B. (J 1936; S 1935), Consumers Power Co, and (for mail), 225 S Butler Blvd, Lansing, Mich

CHURCH, Herbert J (M 1922), Mgr (for mail), Darling Bros, Ltd, 137 Wellington St W, Room 904, Toronto, and 358 Main St N, Weston, Ont, Canada

CLARE, Fulton W. (M 1927), Pres (for mail), Clare & Co, 120 Spring St N W, and 935 Plymouth Rd N E, Atlanta, Ga

CLARK, E. Harold (M 1922), Mfrs Agt, 600 Michigan Theatre Bldg, and (for mail), 2539 Lakewood, Detroit, Mich

CLARKE, Joseph G. (M 1936), Sales Engr, Delco-Frigidaire Conditioning Corp, Wisconsin Blvd, and (for mail), 1233 Carlisle Ave, Dayton, Ohio

CLARKE, Samuel S. (Life Member, M 1909), Pres.-Mgr (for mail), S S Clarke & Co, Ltd, 605 W Second St, and 603 W Second St, Calgary, Alta, Canada

CLARKSON, W. B. (Life Member, M 1919), Vice-Pres, King Ventilating Co, and (for mail), 251 Broadway, Owatonna, Minn.

CLAY, Charles H. (S 1935), Mech Engr, 815 Main St, Cedar Falls, Iowa

CLEGG, Carl (M 1922), Dist Mgr (for mail), American Blower Corp, 311 Mutual Bldg, and 3513 Gillham Rd, Kansas City, Mo.

CLEGG, Robert R (A 1933), Zone Mgr, Owens-Illinois Glass Co, 1538 LaSalle Wacker Bldg, and (for mail), 3270 Lake Shore Drive, Chicago, Ill

CLERICUZIO, Gerald P. (J 1935), 87 Grove St, Bloomfield, N J

CLEVELAND, Clyde C. (A 1936), Htg Engr, Johnson & Cleveland, 192 Main St, and (for mail), 64 E Main St, Bradford, Pa

CLIPPINGER, J. Verne (S 1936), 6611 Dalzell Place, Pittsburgh, Pa

CLODFELTER, John L. (A 1932), Supt (for mail), Carolina Sheet Metal Corp, 17th St and Chelten Ave, Philadelphia, and West Chester Pike and Brief Ave, Elizabeth Manor Apt, Upper Darby, Pa

CLOSE, Paul D.* (M 1928), Chief Engr, Industrial Uses Div (for mail), Celotex Co, 919 N Michigan Ave, Chicago, and 564 Meadow Rd, Winnetka, Ill

CLOUGH, Leslie (M 1922), Htg and Vtg Engr, 203 Pierce Rd, Weymouth, Mass

COCHRAN, Charles C. (A 1935), Branch Mgr. (for mail), Minneapolis-Honeywell Regulator Co, 325 W North Ave, and 1226 W Wisconsin Ave, Milwaukee, Wis

COCHRAN, Lex H. (M 1934), Dist Mgr (for mail), American Blower Corp., 625 Market St, and 130 Camino Del Mar, San Francisco, Calif

COCHRAN, William B. (J 1936; S 1935), Air Cond Engr (for mail), Cochran Air Conditioning Co (Westinghouse), 1303 Lamar Ave., and 3316 Telephone Rd, Box 16, Houston, Texas.

CODY, Henry C. (M 1936), Sales Engr, Pierce, Butler Radiator Corp., 19th and Glenwood Ave, and (for mail), 7336 North 21st St., Philadelphia, Pa

COE, Ralph T. (M 1917), Owner (for mail), R. T Coe Cos., 410 Reynolds Arcade Bldg, and 235 Chili Ave, Rochester, N Y

COHAGEN, Chandler C. (M 1919), Archt (for mail), 211 Hedden Bldg, and 127 Wyoming Ave, Billings, Mont

COHEN, Harry (*S* 1936), 3630 East 110th St, Cleveland, Ohio

COHEN, Philip (*M* 1932), Dist Mgr (for mail), B F. Sturtevant Co, 107 F Ohio Gas Bldg, Cleveland, and 3081 Lynnfield Rd, Shaker Heights, Ohio

COLBY, Clyde W (*M* 1915), Consulting Engr (Air Cond), 1215 Main St, Springfield, Mass, 578 Madison Ave, New York, N Y, and (for mail), 80 N Pleasant St, Holyoke, Mass

COLCLOUGH, O T. (*A* 1933), Custodian, American Legation, American Government Bldg, and (for mail), 180 Lisgar St, Ottawa, Canada

COLE, Grant E (*A* 1925), Vice-Pres (for mail), Trane Co of Canada, Ltd, 139 King St W, Toronto, and Dixie, Ont, Canada

COLEMAN, John B (*M* 1920), Chief Engr (for mail), Grinnell Co, Inc, 260 W Exchange St, and 237 Cole Ave, Providence, R I

COLLIER, William I. (*M* 1921), Pres (for mail), W I Collier & Co, 522 Park Ave, Baltimore, and Ellicott St, Ellicott City, Md

COLLINS, John F S, Jr (*M* 1933), Supervisor of Steam Utilization (for mail), Allegheny County Steam Heating Co, Philadelphia Company Bldg, 135 Sixth Ave, and 827 N Euclid Ave, Pittsburgh, Pa

COLLINS, Max D. (*J* 1936; *S* 1935), Mech Engr, 418 Concord St, St Paul, Minn

COMPTON, Warren E. (*J* 1936, *S* 1935), Engr, 901 W Green St, Urbana, Ill

COMSTOCK, Glen M. (*A* 1926), Dist Repr, L J. Wing Mfg Co, 151 West 14th St, New York, N Y, and (for mail), 601 Chamber of Commerce Bldg., Pittsburgh, and 151 College Ave, Beaver, Pa

CONATY, Bernard M. (*M* 1935), Sales Mgr (for mail), American District Steam Co, N Tonawanda, and 177 Sanders Rd, Buffalo, N Y

CONNELL, Harold (*M* 1935), Engr and Estimator (for mail), Nolan Engineering Co, Rear 1114 18th St N.W, Washington, D C

CONNELL, Richard F. (*M* 1916), Mgr, Capitol Testing Lab (for mail), U S Radiator Corp, 1056 National Bank Bldg, and 2970 Burlingame, Detroit, Mich

CONNER, Raymond M (*M* 1931), Director (for mail), American Gas Association Testing Laboratories, 1032 East 62nd St, Cleveland, and 271 East 216th St, Euclid, Ohio

CONRAD, Roy (*M* 1935), Regional Sales Engr. (for mail), Kelvinator Corp, 708 Sharon Bldg, 55 New Montgomery St, and 1973 17th Ave, San Francisco, Calif.

CONSTANT, Earl S. (*J* 1935), Air Cond Sales Engr, Tom Dolan Heating Co, Inc, 614 W Grand, and (for mail), 1216 N Virginia Ave, Oklahoma City, Okla.

COOK, Benjamin F (*M* 1920), Consulting Engr. (for mail), 114 W Tenth St Bldg, Kansas City, and 1720 Overton Ave, Independence, Mo

COOK, Harris R (*A* 1935), Dist Mgr (for mail), American Foundry & Furnace Co, 2218 N Third St, and 2325 North 50th St, Milwaukee, Wis.

COOK, Ralph P (*M* 1930), Asst Supt, Engrg. and Maintenance Dept in charge of Engrg Div, Eastman Kodak Co (for mail), Kodak Park Works, and 663 Seneca Parkway, Rochester, N Y

COOMBE, James (*A* 1932), Vice-Pres (for mail), William Powell Co, 2525 Spring Grove Ave, and 2363 Grandin Rd, Cincinnati, Ohio.

COON, Thurlow E. (*M* 1916), Pres (for mail), Coon-DeVisser Co, Inc, 2051 W. Lafayette, and 826 Edison Ave., Detroit, Mich.

COOPER, Albert W. (*M* 1935), Branch Mgr. (for mail), Johnson Service Co, 153 W Ave 34, Los Angeles, and 2614 Hermosita Drive, Glendale, Calif

COOPER, John W. (*M* 1932, *A* 1925; *J* 1921), Repr. (for mail), Buffalo Forge Co, 1596 Arcade Bldg, St Louis, and 312 E. Big Bend Rd, Webster Groves, Mo

COOPER, Tom E (*S* 1936), 300 West 50th St, Minneapolis, Minn

COPPERUD, Edmund R (*J* 1933), Asst Mgr (for mail), Minneapolis Plumbing Co, 1420 Nicollet Ave, and 4110 Nicollet Ave, Minneapolis, Minn

COREY, George R. (*M* 1936), 1173 Boulevard, West Hartford, Conn

CORNELL, J. Clarence (*A* 1930), Checker (Mechanical), 12 South 12th St, and (for mail), 2823 W Allegheny Ave, Philadelphia, Pa

CORNISH, Donald F (*J* 1936), Consulting Engr, Dominion Heating Specialty Co, and (for mail), 95 Dinnick Cres, Toronto, Ont, Canada

CORNWALL, Charles C. (*J* 1935), Research Engr, Bahnson Co, 1001 S Marshall St, and (for mail), 473 Carolina Circle, Winston-Salem, N C

CORNWALL, George I. (*M* 1919), Salesman (for mail), Burnham Boiler Corp, 701 Spring St, and 633 Madison Ave, Elizabeth, N J

CORRAO, Joseph (*A* 1936, *J* 1933), Engr, 854-31st Ave, San Francisco, Calif

CORRIGAN, James A. (*J* 1935, *S* 1930), Engr (for mail), Corrigan Co, 2501 St Louis Ave, and 6130 McPherson, St Louis, Mo

COTTER, Robert P. (*S* 1935), Carrier Air Cond Engr, Martin Wright Electric Co, 1001 Navarro St, and (for mail), 935 W Summit Ave, San Antonio, Texas

COVER, Richard R (*A* 1936), Carrier Corp, 301 Tower Bldg, and (for mail), 1302 Gallatin St N W, Washington, D C

COWARD, Charles W. (*M* 1935), Pres (for mail), Coward Engineering Co, 413 Cooper St, Camden, and 812 Lincoln Ave, Palmyra, N J

COX, Harrison F. (*A* 1930), Htg and Air Cond, 243 Carroll St, Paterson, N J

COX, Thomas M., Jr. (*J* 1937), Sales Engr, Neal & Massy Engineering Co, Ltd, Port of Spain, Trinidad, B W I

COX, William W (*M* 1923), Pres-Mgr (for mail), Heating Service Co, 326 Columbia St, and 6232-31st Ave N E, Seattle, Wash

CRANSTON, William E., Jr. (*M* 1931), 2821 E Pico St, Los Angeles, Calif

CRAWFORD, John H, Jr (*A* 1936, *J* 1930), Air Cond Engr, Hitchen Engineering Co, 441 Lexington Ave, New York, N Y, and (for mail), 133 Lincoln Ave, Orange, N J

CRICHTON, Howard C (*S* 1936), 769 Beaver Ave, Midland, Pa

CRIQUI, Albert A.* (*M* 1919), Chief Engr., Htg and Vtg Dept, Buffalo Forge Co, 490 Broadway, and (for mail), 250 Blaine Ave, Buffalo, N. Y

CRONE, Charles E, Jr. (*M* 1922), Secy-Treas (for mail), Wendt & Crone Co, 2124 N Southport Ave, and 1320 N. State St, Chicago, Ill.

CRONE, Thomas E. (*Life Member*, *M* 1920), Salesman, W A Russell & Co, Grand Central Terminal Bldg, New York, and (for mail), 486 Murray St, Pelham Manor, N Y

CROSBY, Edward L. (*M* 1936), Pres (for mail), Henry Adams, Inc, Consulting Engrs, 1263-69 Calvert Bldg, and 5811 Merville Ave, Baltimore, Md

CROSS, Freeman G. (*M* 1936), Sales Mgr (for mail), Fulton Sylphon Co, 19 Rector St, New York, and 102 Ascan Ave, Forest Hills, L. I, N. Y

CROSS, Robert E. (*A* 1931), 95 State St, Springfield, Mass.

CUCCI, Victor J. (*M* 1930), Consulting Engr (for mail), 347 Madison Ave, New York, and 451-55th St, Brooklyn, N Y

CULBERT, William P. (*A* 1929), Secy (for mail), Culbert-Whitby Co, Inc, 2019 Rittenhouse St., Philadelphia, and 929 Alexander Ave, Drexel Hill, Pa.

CULLEN, Augustine G. (*A* 1936), Pres (for mail), Cullen, Inc, 418 Florida Ave N E, and 1301 Massachusetts Ave N W, Washington, D. C.

CUMMING, Ford J (M 1936), Pres (for mail), Beecher-Cumming, Inc, 820 Second Ave S, and 2628 Garfield Ave S, Minneapolis, Minn

CUMMING, Robert W. (M 1928), Engr-Sales Executive, Sarco Co, Inc, 183 Madison Ave, New York, and (for mail), 81 Alkamont Ave, Scarsdale, N Y

CUMMINGS, Carl H. (A 1927, J 1926), Mgr (for mail), Industrial Appliance Co of New England, 250 Stuart St, Boston, and 41 Edgehill Rd, Chestnut Hill, Mass

CUMMINGS, G. J (M 1923), Mgr (for mail), The Scott Co, 113 Tenth St, and 2001 Hoover Ave, Oakland, Calif

CUMMINS, George H. (M 1919), Dist Mgr (for mail), Aerofin Corp, 616 United Artist's Bldg, and 17376 Wisconsin Ave, Detroit, Mich

CUNNINGHAM, John S. (J 1937, S 1935), 5052 S Ellis, Chicago, Ill

CUNNINGHAM, Thomas M. (M 1931, J 1930), Production Mgr, Carrier Corp, 180 N Michigan Ave, Chicago, Ill

CURRIE, Francis J. (A 1935), Prop, Plbg-Htg Contractor, 16 South 39th St, Philadelphia, and (for mail), 9 Montrose Ave, Kirklyn, Upper Darby, Pa

CURRIER, Charles H. (M 1919), Vice-Pres (for mail), Ross Heater & Mfg Co, Inc, 1407 West Ave, and Park Lane Apts, 33 Gates Circle, Buffalo, N Y

CURTICE, Jean M (A 1936), Repr, House Htg Engr, Public Service Co of Colorado, W Mountain Ave, and (for mail), 309 Locust St, Fort Collins, Colo

CURTIS, Herbert F. (A 1934), Chief Engr (for mail), Henry Furnace & Foundry Co, 3471 East 49th St, Cleveland, and 59 Fourth Ave, Berea, Ohio

CURTIS, Walter A. (S 1936), 2139 Abington Rd, Cleveland, Ohio

CUTLER, Joseph A. (M 1916), Vice-Pres (for mail), Johnson Service Co, 1355 Washington Blvd, Chicago, and 849 Hinman Ave, Evanston, Ill

D

DAHLSTROM, Godfrey A (A 1927), Htg Sales Engr, Central Supply Co, 312 S Third St, and (for mail), 3721-47th Ave. S, Minneapolis, Minn

DAILEY, James A. (A 1920), 31-64-30th St, Astoria, L I, N Y.

DAKIN, Harold W (J 1934), General Electric Co, Woodlawn Ave., Pittsfield, and (for mail), 169 Park Ave, Dalton, Mass

DALLAVALLE, J. M * (A 1936, J 1933), Asst Sanitary Engr (for mail), U S Public Health Service, 19th and Constitution Ave, Washington, D C, and 17 Jones Bridge Rd, Chevy Chase, Md

DALY, Charles P. (A 1935), Member of Firm (for mail), Rautman Plumbing & Heating Co, 115 Jackson St., and 2438 Queen Anne Ave, Seattle, Wash

DALY, Robert E (M 1931), Exec Engrg Dept (for mail), American Radiator Co, 40 West 40th St, and 12 East 88th St, New York, N Y

DAMBLY, A. Ernest (M 1924; J 1921), (for mail), c/o H B Hackett, 901 Architects Bldg, Philadelphia, Pa, and Harvey Cedars, N J

DANIELSON, E. B. (A 1936), Owner-Mgr (for mail), Air Conditioning Co, 627 Main St., and 448 E Third St, Russell, Kan

DANIELSON, Lloyd C. (S 1936), Sales Engr (for mail), Air Conditioning Co, 627 Main St, and 448 E Third St., Russell, Kan

DANIELSON, Wilmot A. (M 1935), Lieut-Col, Q M. C, U S Army (for mail), 2236 Munitions Bldg, and 3812 Fulton St N W., Washington, D C

DARBY, Marion H (J 1930), Chief Engr, Air Cond Dept (for mail), General Electric, S A., Avenida Rio, Branco No 114, Rio de Janeiro, Brazil, and c/o Carrier-Brunswick International, Inc, 850 Frelinghuysen Ave., Newark, N J

DARLING, Arthur B. (A 1929), Asst Sales Mgr (for mail), Darling Bros, Ltd, 140 Prince St, Montreal, P Q, and 4216 Dorchester St. W, Westmont, P Q, Canada

DARLINGTON, Allan P. (M 1930), Dist Sales Mgr (for mail), American Blower Corp, 632 Fisher Bldg, and 5200 Haverhill, Detroit, Mich

DARTS, John A. (M 1919), Kewanee Boiler Co, Inc, 101 Park Ave, New York, N Y

DAUCH, Emil O. (M 1921), Secy-Treas (for mail), McCormick Plumbing Supply Co, 1675 Bagley Ave, and The Whittier Hotel, Detroit, Mich

DAVENPORT, R. F. (A 1933), Furnace Sales Mgr., Canada Foundries & Forgings, Ltd, and (for mail), 77 James St E, Brockville, Ont, Canada

DAVIDSON, John C. (J 1936), Gas Htg Inspector, City of Minneapolis, 213 City Hall, and (for mail), 3628 Nicollet Ave, Minneapolis, Minn

DAVIDSON, L. Clifford (M 1927), Associate Dist Mgr (for mail), Buffalo Forge Co, 220 South 16th St, Philadelphia, and 439 Anthwyn Rd, Merion, Pa

DAVIDSON, Philip L (M 1924, J 1921), Consulting Engr (for mail), 1204 Commercial Trust Bldg, Philadelphia, and 14 Radnor Way, Radnor, Pa

DAVIES, George W (M 1918), Mgr, G W Davies & Co (for mail), P O Box 390, Dunedin, C 1N 2, and Colinswood, Macandiew Bay, New Zealand

DAVIS, Arthur C.* (M 1920), Supt of Maintenance, Port of New York Authority, 111 Eighth Ave, New York, N Y, and (for mail), 73 Preston St, Ridgefield Park, N J

DAVIS, Arthur F. (M 1934), Vice-Pres (for mail), Johnson & Davis Plumbing & Heating Co, 2235 Arapahoe St., and 1901 Ivanhoe St, Denver, Colo

DAVIS, Bert C (M 1901), (Council, 1917), Pres-Treas (for mail), American Warming & Ventilating Co, 317-19 Pennsylvania Ave, and 603 W Church St, Elmira, N Y

DAVIS, Calvin R. (M 1927), Branch Mgr (for mail) Johnson Service Co, 2328 Locust St., and 7534 Westmoreland Drive, St Louis, Mo.

DAVIS, George C. (J 1936), Sales Engr (for mail), Northern Public Service Corp, Ltd., 307 Power Bldg, and 516 Devon Court, Broadway, Winnipeg, Man, Canada

DAVIS, James R. (J 1936, S 1934), 9704 Miles Ave, Cleveland, Ohio

DAVIS, Joseph (M 1927, A 1926), Owner-Htg Engr and Contractor (for mail), 607 Root Bldg, 70 W Chippewa, and 166 Huntington Ave., Buffalo, N Y

DAVIS, Otis E. (M 1929, A 1925), Sales Engr (for mail), Hoffman Specialty Co., Waterbury, Conn., and Box 98, Scottsbluff, Nebr.

DAVIS, Rowland G. (A 1921), Sales Repr, 887 Nela View Rd, Cleveland Heights, Ohio

DAVISON, Robert L. (M 1931), Director of Housing Research (for mail), John B Pierce Foundation, 37 West 39th St, New York, and Meadow Glen Rd, Fort Salonga, L I, N. Y.

DAWSON, Eugene F. (M 1934), Asst Prof, Mech Engr (for mail), University of Oklahoma, and 916 S Flood St, Norman, Okla

DAWSON, G. Stewart (A 1935), Mdse Sales Engr (for mail), British Columbia Electric Railway Co, Ltd, 425 Carrall St., and 1860 Barclay St, Vancouver, B. C., Canada.

DAWSON, Thomas L. (M 1930), Pres (for mail), Thomas L Dawson Co, 2035 Washington St., Kansas City, Mo, and 56th and Shawnee Mission Rd, Rosedale Station, Kansas City, Kan

DAY, Harold C. (A 1934), Mgr. (for mail), American Radiator Co, 1807 Elmwood Ave, and Stuyvesant Hotel, Buffalo, N. Y

DAY, Irving M. (A 1930), Sales Engr (for mail), 722 Mills Bldg, Washington, D C., and 405 Cumberland Ave, Chevy Chase, Md.

DAY, V. S * (*M* 1921), Engr (for mail), Carrier Corp, 850 Frelinghuysen Ave, Newark, and 100 Summit Ave, Summit, N J

DEAN, Carl H. (*M* 1930), Htg Engr (for mail), Oklahoma Natural Gas Co, P O 871, and 1007 N Main, Tulsa, Okla

DEAN, Charles L. (*M* 1932), Asst. Prof of Mech Engr, University of Wisconsin, 305 University Extension Bldg, and (for mail), 102 Grand Ave, Madison, Wis

DEAN, Frank J, Jr (*J* 1935, *S* 1934), Sales Engr, Gustin-Bacon Mfg Co, 1412 West 12th St, and (for mail), 6028 Walnut St, Kansas City, Mo

DEAN, Marshall H. (*S* 1936), Princeton University (for mail), 324 1903 Hall, Princeton, N. J., and 1030 West 55th St, Kansas City, Mo

DEANE, Joseph H. (*A* 1937), Sales Engr (for mail), Detroit Lubricator Co, 816 S Michigan Ave, Chicago, and 125 W Hickory St, Lombard, Ill

DeBOOS, F A. (*M* 1935), Dist Mgr, B F Sturtevant Co, 812 Michigan Theatre Bldg, and (for mail), 1083 Field Ave, Detroit, Mich

DEELY, James J. (*J* 1933), Supervisor House Htg Div. (for mail), Brooklyn Union Gas Co, 180 Remsen St, Brooklyn, and 61 College Ave N., Tarrytown, N. Y

DeLAND, Charles W. (*M* 1924; *J* 1923), Secy-Treas. (for mail), C W Johnson, Inc, 211 N Desplaines St, and 2021 Estes Ave, Chicago, Ill

DeMERIT, Robert N. (*S* 1936), 314 Le Roi Rd, Pittsburgh, Pa

DENISE, John R. (*J* 1935), Asst Commercial Engr, Consolidated Gas Co. of New York, 4 Irving Place, and (for mail), 230 Madison Ave, New York, N Y.

DENNY, Harold R. (*A* 1934), Eastern Mdse Mgr. (for mail), American Blower Corp, 40 West 40th St., New York, N Y., and 429 Edgewood Ave, Westfield, N J

DES REIS, John F. (*M* 1930), Field Mgr, West Indies and Caribbean Div., Carrier Corp, and (for mail), P O Box 5157, Puerta de Tierra Station, San Juan, Puerto Rico

DEVER, Henry F (*M* 1936; *A* 1935), Branch Mgr, Minneapolis-Honeywell Regulator Co, Wayne and Roberts Aves., Philadelphia, and (for mail), 502 Merwyn Rd, Narberth, Pa

DEWAR, William G. (*A* 1936), Purchasing Agt, Construction Equipment Co, 1200 Benoit St, and (for mail), Apt 4, 5550 Cote St Luc, Montreal, Que., Canada

DEWEY, Ritchie P. (*M* 1934), Mgr, Temp Control and Uni-Flo Depts., Barber-Colman Co, River and Loomis Sts, and (for mail), 2301 Oxford St, Rockford, Ill

DeWITT, Earl S. (*A* 1936), Branch Mgr (for mail), American Blower Corp, 438 Woodward Bldg, and 5429-41st N.W, Washington, D. C.

DIBBLE, S. E.* (*M* 1917), (*Presidential Member*), (Pres., 1925, 1st Vice-Pres., 1924, 2nd Vice-Pres., 1923, Council, 1921-1926), Supt, Thomas Ranken Patton School, Elizabethtown, Pa

DICK, Andrew V. (*J* 1935), Sales Engr, National Radiator Corp, 141 Jay St., Albany, N Y.

DICKENSON, Frederick R. (*M* 1936; *A* 1934), Asst. Sales Mgr. (for mail), American Blower Corp., 6000 Russell St, Detroit, and 284 Pilgrim Rd., Birmingham, Mich.

DICKENSON, Malcolm E. (*M* 1936), Gen Mgr (for mail), Livingston Stoker Co., Ltd, 78 Catharine St. N., and 964 Cumberland Ave, Hamilton, Canada

DICKEY, Arthur J. (*M* 1921), Vice-Pres.-Gen Mgr., C. A. Dunham Co, Ltd., 1523 Davenport Rd., and (for mail), 9 Mossom Place, Toronto, Ont., Canada.

DICKINSON, Tom (*A* 1936), Draftsman, York Ice Machinery Corp, and (for mail), 419 West 65th St, Los Angeles, Calif

DICKSON, George P. (*M* 1919), Dist Mgr. (for mail), B F. Sturtevant Co, 89 Broad St, Boston, Mass., and P O. Box 22, Canterbury, New Hampshire.

DICKSON, Robert B. (*M* 1919), Pres (for mail), Kewanee Boiler Corp, Franklin St and Q Tracks, and 409 E Prospect St, Kewanee, Ill

D'IMOR, Elton J. (*M* 1933), Engr-Mgr, Power Equipment Co, 829 16th St, Des Moines, Iowa.

DISNEY, Melvin A (*A* 1934), Partner, Disney-Leffel Co, 628 New York Life Bldg, Kansas City, Mo, and (for mail), 1147 S Quaker, P O. Box 1585, Tulsa, Okla

DIVER, M. L. (*M* 1925), Consulting Engr, P O Box 1016, San Antonio, Texas

DIXON, Arthur G (*M* 1928), Sales Mgr (for mail), Modine Mfg Co, and 442 Wolff St, Racine, Wis

DIXON, Meredith F. (*A* 1937), Combustion Engr, Fuel Oil and Oil Burner Div, Imperial Oil, Ltd, P O Box 1440, and (for mail), 2281 Wilson Ave, Montreal, Que, Canada

DOBBS, C. E. (*A* 1921), Repr, Burnham Boiler Corp, 31st and Jefferson Sts, Philadelphia, Pa, and (for mail), 72 Berlin Ave, Haddonfield, N J

DOBIE, Thomas K. (*A* 1936), Supt of Bldgs, London Life Insurance Co, and (for mail), 969 Colborne St, London, Ont, Canada

DODDS, Forrest F. (*M* 1920), Mgr, K C Branch (for mail), American Radiator Co, 1023 Grand Ave, and 229 Ward Parkway, Kansas City, Mo.

DODGE, Harry A. (*M* 1936), Elec Engr, S H. Kress & Co, 114 Fifth Ave, and (for mail), 425 East 86th St, New York, N Y

DOERING, Frank L. (*M* 1919), Sales Engr., American Radiator Co., 238 Boston Ave, Lynchburg, Va

DOHERTY, Russell (*A* 1929), Dist Mgr (for mail), National Radiator Corp, 1111 East 83rd St, Chicago, and 300 Forest Ave, Oak Park, Ill.

DOLAN, Raymond G. (*M* 1926, *J* 1922), Secy-Treas (for mail), Tom Dolan Heating Co, Inc, 614 W Grand, and 2112 West 20th, Oklahoma City, Okla

DOLSON, Charles N. (*A* 1937), Sales Engr, Stokers, R F Knoche & Son, and (for mail), 36 Hunt Ave, Hamburg, N Y.

DOME, Alan G. (*J* 1936), Engr, Bryant Air Conditioning Corp, 152 North 15th St, and (for mail), 7839 Devon St, Philadelphia, Pa

DONNELLY, James A * (*M* 1904), (Treas, 1912-1914), Largent, W Va

DONNELLY, Russell (*M* 1923), Sales Engr., Nash Engrg Co, Graybar Bldg, 420 Lexington Ave, New York, N Y

DONOHOE, John B (*J* 1935), Engr-Estimator (for mail), B F Donohoe Co, 51 Albany St, Boston, and 24 Primrose St., Roslindale, Mass.

DONOVAN, William J. (*A* 1930), 2239 North 27th St, Philadelphia, Pa.

DONZELLI, Enrico (*M* 1933), E. Donzelli & Co, Piazza SS Pietro e Lino No. 4, Milan, Italy

DORFAN, M. I. (*M* 1929), Mgr, Dust Control Div, Blaw-Knox Co, P. O. Box 1198, and (for mail), 1217 Malvern Ave, Pittsburgh, Pa.

DORNHEIM, G. A. (*M* 1912; *J* 1906), 15 Hamilton Ave, Bronxville, N Y.

DORSEY, Francis C. (*M* 1920), Engr-Contractor (for mail), Francis C Dorsey, Inc., 4520 Schenley Rd, Roland Park, and 212 Gittings Ave, Baltimore, Md.

DOSTER, Alexis (*A* 1934), Secy-Director (for mail), Torrington Mfg Co, 70 Franklin St, Torrington and Litchfield, Conn

DOUGHTY, Charles J. (*M* 1925), Pres-Managing Director (for mail), C J Doughty & Co, Fed Inc, U S A., 30 Brenan Rd., and 1920 Ave., Joffre, Shanghai, China

DOUGLAS, Howard H. (*A* 1936), Elec. Htg. Engr. (for mail), Southern California Edison Co, 601 W. Fifth St, and 2317 Kelton Ave., Los Angeles, Calif

DOVOLIS, Nick J. (*J* 1936, *S* 1935), 3403 Chicago Ave., Minneapolis, Minn

DOWNE, Edward R. (*M* 1927), Vice-Pres., American Gas Products Corp, 40 West 40th St, New York, and (for mail), 108 Park Ave., Larchmont, N. Y.

DOWNE, Henry S (Life Member, M 1895), Cie Nationale des Radiateurs, 149 Boulevard Haussman, Paris, France

DOWNES, Henry H (M 1923), Dist Mgr (for mail), American Blower Corp, 438 Woodward Bldg, Washington, D C, and 4621 Chevy Chase Blvd, Chevy Chase, Md

DOWNES, Nate W. (M 1917), (Council, 1928-1930), Chief Engr-Supt of Bldg (for mail), School District of Kansas City, Mo, 317 Finance Bldg, and 2119 East 68th St, Kansas City, Mo

DOWNS, Charles R (M 1936), Vice-Pres-Treas (for mail), Weiss & Downs, Inc, 50 East 41st St, New York, N Y, and Sylvan Lane, Old Greenwich, Conn

DOWNS, Sewell H (M 1931), Chief Engr, Clarage Fan Co, and (for mail), 211 Creston Ave, Kalamazoo, Mich

DOYLE, William J (M 1920), Factory Mgr, Williamson Heater Co, 4558 Marburg Ave, and (for mail), 3766 Hyde Park Ave, Cincinnati, Ohio

DRAKE, George M (J 1936), Vice-Pres (for mail), George H Drake, Inc, 218 Lexington Ave, and 351 Norwood Ave, Buffalo, N Y

DRINKER, Philip+ (M 1922), Prof (for mail), Harvard School of Public Health, 55 Shattuck St, Boston, and 40 Puddingstone Lane, Newton Centre, Mass

DRISCOLL, William H.* (M 1904), (Presidential Member), (Pres, 1926, 1st Vice-Pres, 1925; 2nd Vice-Pres, 1924, Treas, 1923, Council, 1918-1927), Vice-Pres (for mail), Thompson-Starrett Co, Inc, 444 Madison Ave, New York, N Y, and 50 Glenwood Ave, Jersey City, N J

DuBOIS, Louis J. (M 1931), Air Cond Engr, York Ice Machinery Corp, 117 South 11th St, St Louis, and (for mail), 7451 Bland Drive, Clayton, Mo

DUBRY, Ernest E. (M 1924), Asst Supt, Central Htg, Detroit Edison Co, 2000 Second Ave, and (for mail), 9116 Dexter Blvd, Detroit, Mich

DUFAULT, Felix H. (A 1936), Mgr Furnace Div (for mail), General Steel Wares, Ltd, 2355 Delisle St, and 1277 Visitation St, Montreal, Que, Canada

DUFF, Kennedy (M 1915), Mgr Eastern Territory (for mail), Johnson Service Co, 28 East 29th St., New York, N Y, and 9 Park Ave, Maplewood, N J

DUGAN, Thomas M. (M 1920), Sanitary-Htg Engr, National Tube Co, Fourth Ave and Locust St, and (for mail), 1308 Freemont St., McKeesport, Pa

DUGGER, Earl R (J 1936, S 1934), Service-Installation Engr, Oklahoma Refrigerating Co., 18 W Grand, and (for mail), 3409 Classen, Oklahoma City, Okla

DULLE, Willferd L. (J 1936), Asst Secy, E E Souther Iron Co, St Louis, and (for mail), 2910 Lincoln Ave, Normandy, Mo

DUNCAN, James R. (M 1923), Sales Engr., Air Cond, Carrier Corp, Room 408, Chrysler Bldg., New York, N Y

DUNCAN, William A. (A 1930), Dist Service Engr (for mail), Dominion Oxygen Co, Ltd., 92 Adelaide St. W, and 20 Tyrell Ave, Toronto, Ont, Canada

DUNHAM, Clayton A.* (M 1911), Pres (for mail), C A. Dunham Co, 450 E Ohio St, Chicago, and 150 Maple Hill Rd, Glencoe, Ill

DUPUIS, Joseph R. (A 1936), Dist Mgr (for mail), Trane Co of Canada, Ltd, 660 St Catherine W, Montreal, Que, and 331 Clarke Ave, Westmount, Que., Canada.

DURKEE, Merritt E. (A 1930), Sales Engr, C. A Dunham Co, 30 Rockefeller Plaza, New York, and (for mail), 169 Church St, White Plains, N Y

DURNING, Edward H. (A 1936; J 1931), Commercial Sales, Dallas Gas Co, Harwood and Jackson Sts, and (for mail), 1830 Moser St, Dallas, Texas.

DWYER, Thomas F (M 1923), Mech Engr (for mail), Board of Education, 49 Flatbush Ave Ext, Brooklyn, and 1163 Clay Ave, New York, N Y

DYKES, James B. (J 1936), Estimator (for mail), T A Morrison & Co, Ltd, 1070 Bleury St, Montreal, and 37 Glencoe Ave, Outremont, P Q, Canada

E

EADE, Hugh R. (M 1935), Archt (for mail), 5 Imperial Bank Bldg, and Cheriton Ave, North Kildonan, Winnipeg, Canada

EADIE, John G (M 1909), Consulting Engr, Eadie, Freund & Campbell Co, 110 West 40th St, New York, N Y

EAGAR, R. Frank (M 1922), 93 Edward St, Halifax, N S, Canada

EAGLETON, Sterling P (M 1936), Assoc Air Cond Engr, National Park Service (Bldgs Branch), Navy Bldg, and (for mail), 3522 "S" St N W, Washington, D C

EARL, Warren (A 1936), Sales Engr, W C Dance, Inc, 811 N Bldg, and (for mail), 618 N. W 29th St, Oklahoma City, Okla

EARLEY, Thomas J. (A 1935), Sales Engr, Jennison Co, Putnam St, and (for mail), 46 Elizabeth St, Fitchburg, Mass

EASTMAN, Carl B (M 1932, J 1929), Mgr, Philadelphia Sales Office, C A Dunham Co, 1500 Walnut St, Philadelphia, and (for mail), 530 Brookview Lane, Brookline, Upper Darby, Pa

EASTWOOD, E. O. (M 1921), (Council, 1931-1934), Prof of Mech. Engrg (for mail), University of Washington, and 1702 12th Ave, Seattle, Wash

EATON, Byron K. (M 1920), Zone Mgr, Delco-Frigidaire Conditioning Corp, 1420 Wisconsin Blvd, Dayton, Ohio, and (for mail), 240 S Brainard Ave, LaGrange, Ill

EATON, William G M. (A 1934), Sales Engr, Pease Foundry Co, Ltd, 277 Victoria St, Toronto 2, and (for mail), 59 Symington Ave, Toronto 9, Canada

EBERT, William A (M 1920), Mech Contractor (for mail), 1026 W Ashby, and 2151 W Kings Highway, San Antonio, Texas

ECKERT, E. Kendall (J 1935), Engr (for mail), American Blower Corp, 6000 Russell St., and 91 E Kirby, Detroit, Mich

EDELMAN, Bernard P (A 1935), Asst Sales Mgr (for mail), U S Air Conditioning Corp, 2101 Kennedy St N E, and 4233 Nicollet Ave, Minneapolis, Minn

EDWARDS, Arthur W (M 1936), Dist Mgr, The Trane Co, 628 Broadway, and (for mail), 3062 Griest Ave, Cincinnati, Ohio

EDWARDS, Daniel F. (M 1920), 2340-42 Pine St, St Louis, Mo

EDWARDS, Don J. (A 1933), Vice-Pres (for mail), General Heat & Appliance Co, 698 Beacon St, Boston, and 40 Rockledge Rd, Newton, Mass

EDWARDS, Henry B (J 1935), D S. Horrall & Co, 513 W. Van Buren St, Phoenix, Ariz

EDWARDS, Junius D. (M 1936), Asst Director of Research (for mail), Aluminum Company of America, P O Box 772 (Freeport Rd), New Kensington, and 536 Sixth St., Oakmont, Pa

EDWARDS, Paul A. (M 1919), Pres (for mail), G F Higgins Co, 608 Wabash Bldg, and 3074 Pinehurst Ave, Pittsburgh 16, Pa

EHRLICH, M. William* (M 1916), Chief Engr, Commodore Heaters Corp, 11 West 42nd St, New York, N Y, and (for mail), 56 Ridge Rd, Lyndhurst, N J.

EICHER, HuBert C. (M 1922), Chief, Div of School Bldgs, Pennsylvania State Dept of Public Instruction, State Capitol, and (for mail), 103 South St, Harrisburg, Pa

EILS, Lee C. (J 1936), Asst. Supt, George J. Meyer & Son, 3223 Kennett Square, Pittsburgh, P O No 13, Pa

EISS, Robert M (*M* 1933, *J* 1930), Engr, Kimberly-Clark Corp, and (for mail), 215 Hewitt St, Neenah, Wis

ELLINGWOOD, Elliott L (*M* 1900), 351 S Spring St, Los Angeles, Calif

ELLIOT, Edwin (*M* 1929), Edwin Elliott & Co (for mail), 560 North 16th St, Philadelphia, and 403 W Price St, Germantown, Philadelphia, Pa

ELLIOTT, Louis (*M* 1932), Consulting Mech Engr, Ebasco Services, Inc, 2 Rector St, Room 1530, New York, N Y

ELLIOTT, Norton B (*A* 1931), Branch Mgr, American Blower Corp, 1011 Majestic Bldg, Milwaukee, Wis.

ELLIS, Frederick E. (*M* 1923), Sales Mgr (for mail), Imperial Iron Corp, Ltd, 30 Jefferson Ave, Toronto, and 9 Princeton Rd, Kingsway P O, Toronto 3, Ont., Canada

ELLIS, Frederic R. (*M* 1913), Sales Engr, Buerkel & Co, Inc, 18-21 Union Park St., Boston, and (for mail), 131 Beacon St, Hyde Park, Boston, Mass

ELLIS, Gershom P. (*M* 1935), Chief Engr (for mail), Board of Public Education, 311 Bellefield Ave, and 6601 Dalzell Place, Pittsburgh, Pa

ELLIS, Harry W. (*Life Member*, *M* 1923, *A* 1900), Pres-Gen Mgr, Johnson Service Co, 507 E Michigan St, Milwaukee, Wis

ELWOOD, Willis H. (*M* 1936), Branch Mgr, Holland Furnace Co, 209 King St., Ithaca, N Y

EMERSON, Ralph R. (*M* 1922), Pres, Emerson Swan Goodyer Co, 110 Arlington St, Boston, and (for mail), 11 Whitney Rd, Newtonville, Mass

EMERY, Gordon W. (*A* 1935), Service Engr, H H Van Sunn, Inc, 190 Moore St, Hackensack, and (for mail), 65 Birk St, Rochelle Park, N. J

EMMERT, Luther D. (*M* 1919), Repr (for mail), Buffalo Forge Co, Room 1909, 20 N Wacker Drive, Chicago, and 1701 Hinman Ave, Evanston, Ill.

EMMONS, Neal L (*J* 1936; *S* 1934), 1100 East 19th, Oklahoma City, Okla

EMSWILER, John E.* (*M* 1917), Prof of Mech Engrg (for mail), University of Michigan, 231 W. Engrg. Bldg., and 1303 Granger Ave, Ann Arbor, Mich.

ENGEL, Edward (*J* 1933), Chief Engrg Draftsman (for mail), U S. Navy Yard, Hull Div., and 2608 North 30th St, Philadelphia, Pa

ENGLE, Alfred (*A* 1923), Secy (for mail), Jenkins Bros, 80 White St, New York, and 1 Edgewood Rd, Scarsdale, N. Y.

ENGLISH, Harrold (*M* 1935, *A* 1930), Pres. (for mail), English & Lauer, Inc, 309 West 12th St, and 201 S Irving Blvd, Los Angeles, Calif.

ENSIGN, Willis A (*M* 1935), Vice-Pres, Frontier Fuel Oil Corp., 986 Ellicott Square Bldg, Buffalo, and (for mail), Revere Drive, Derby, N. Y.

EPPLE, Arnet B. (*J* 1934), Sales Engr, B. F Sturtevant Co, 812 Michigan Theatre Bldg, Detroit, Mich.

ERICKSON, E. Vincent (*M* 1936), Sales Engr. (for mail), Carrier-Brunswick International, 850 Frelinghuysen Ave, Newark, and 803 Colonia Rd., Elizabeth, N J

ERICKSON, Harry H (*A* 1929), Dist. Mgr (for mail), General Fittings Co, 804 Architects Bldg., and 217 W. Tulpehocken St, Philadelphia, Pa

ERICKSON, Martin E. (*A* 1926), Supt, Maintenance, Board of Education, and (for mail), 1533 South 74th St, West Allis, Wis.

ERICSSON, Eric B. (*M* 1933), Engr, Board of Education, and (for mail), 605 West 116th St, Chicago, Ill

ERISMAN, Percival H., Jr. (*M* 1936), Chief Engr, Washington Refrigeration Co, 1731–14th St. N W, and (for mail), 3204–18th St N W., Washington, D C.

ERRATH, Edward O. (*J* 1936), Htg. and Air Cond. Engr. (for mail), Heil Co, 3000 W Montana St, and 2646 North 45th St, Milwaukee, Wis

ESCHENBACH, Samuel P. (*J* 1935), Sales Engr, American Blower Corp, Builders Exchange, 135 Spring St, Rochester, N Y

ESKIN, Samuel G. (*A* 1936), Engr in charge of Development and Design (for mail), American Thermometer Co, 2917 Clark Ave, St Louis, and 619 Westwood Drive, Clayton, Mo

ESTEP, Leslie G (*M* 1936), Engr in charge of Kelvin House, Kelvinator Corp, 14250 Plymouth Rd, and (for mail), 14909 Marlowe Ave, Detroit, Mich

ESTES, Edwin C. (*A* 1936), Mech Draftsman (for mail), Railroad Transportation, Room 820, Northern Pacific Railway Bldg, and 1758 Stanford Ave, St Paul, Minn

ETLINGER, J. Martin (*J* 1936), 2564 Creston Ave, New York, N Y

EVANS, Edwin C. (*M* 1919), B F Sturtevant Co, Inc, 504 Eckel Bldg, Syracuse, N Y

EVELETH, Charles F.* (*M* 1911), 2030 East 115th St, Cleveland, Ohio

EVEREST, R. Harry (*M* 1935), Sales Engr, Sheldons, Ltd, Galt, and (for mail), 235 Waterloo St, Preston, Ont, Canada

EVERETTS, John, Jr* (*A* 1935, *J* 1929), Engr (for mail), Air & Refrigeration Corp, 11 West 42nd St, New York, N Y, and 55 Sound Beach Ave, Old Greenwich, Conn

EVLETH, Everett B. (*A* 1927), Res Vice-Pres. (for mail), Minneapolis-Honeywell Regulator Co, 43 E Ohio St, Chicago, and 1023 Ashland Ave, Wilmette, Ill

F

FABER, Dr. Oscar (*M* 1934), Consulting Engr (for mail), Romney House, Marsham St, Westminster, London and Hayes Court, Kenley, Surrey, England

FAGIN, Daniel J. (*M* 1932), Htg Engr, Laclede Gas Light Co., 1017 Olive St, and (for mail), 1344 Woodruff Ave, St Louis, Mo

FAHNESTOCK, Maurice K.* (*M* 1927), Research Asst Prof (for mail), University of Illinois, 214 M E Laboratory, and 701 W California St, Urbana, Ill

FAILE, Edward H. (*M* 1934), Designing and Construction Engr (for mail), 608 Fifth Ave, New York, N Y, and R F D. 1, Westport, Conn

FALTENBACHER, Harry J. (*M* 1930), Pres, H J. Faltenbacher, Inc, 235 E. Wister St, Philadelphia, Pa

FALVEY, John D. (*M* 1922), Consulting Engr, 316 N. Eighth St, St Louis, and (for mail), 6636 Pershing Ave, University City, Mo

FAMILETTI, A. Robert (*J* 1930), Chief Engrg Draftsman, Industrial Dept, Navy Yard, and (for mail), 6735 Guyer Ave, Philadelphia, Pa

FARBER, Louis M. (*J* 1936), Engr, Natkin & Co, 1800 Baltimore, and (for mail), 3714 Flora, Kansas City, Mo.

FARBMAN, Leonard X. (*J* 1935), Assoc Engr, (for mail), 331 West 59th St, and 817 West End Ave, New York, N Y

FARBMAN, Saul M. (*S* 1935), (for mail), 5227 Fifth Ave, Pittsburgh, Pa, and 817 West End Ave, New York, N Y

FARLEY, W. F. (*M* 1930), Salesman, American Radiator Co, 40 West 40th St, New York, and (for mail), 28 Elm St, New Rochelle, N. Y

FARLEY, Willoughby S. (*A* 1936, *J* 1933), Partner, Farley & Luther, 120 S Union St, and (for mail), 211 Montague St, Danville, Va

FARNHAM, Roswell (*M* 1920), (Council, 1927-1932), Dist Mgr, Engrg Sales (for mail), Buffalo Forge Co., P O Box 985, and 5 Clarendon Place, Buffalo, N Y.

FARNSWORTH, John G. (*A* 1936; *J* 1931), Gas House Htg Engr (for mail), Central Illinois Light Co, 316 S. Jefferson St., and 216 Dechman Ave, Peoria, Ill.

FARRAR, Cecil W. (*M* 1920; *A* 1918), (Treas, 1930, Council, 1930), Pres (for mail), Excelso Products Corp, 65 Clyde Ave, and 29 Oakland Place, Buffalo, N Y

FATZ, Joseph L. (*M* 1935), Htg-Vtg Engr (for mail), Board of Education, 228 N LaSalle St, Room 530, and 1634 N Mason Ave, Chicago, Ill
FAULKNER, Gordon (*S* 1935), 1116 W California St, Urbana, Ill
FAUST Frank H (*M* 1936, *J* 1930), Air Cond Dept (for mail), General Electric Co, 5 Lawrence St, Bloomfield, and 202 Vreeland Ave, Nutley, N J
FAY, Donald P. (*S* 1936), 91-38-115th St, Richmond Hill, L I, N Y
FAY Frank C. (*M* 1925), Engr (for mail), Raisler Heating Co, 129-31 Amsterdam Ave, New York, and 9217-54th Ave, Elmhurst, L I, N Y
FEBREY, Ernest J. (*M* 1903), E J Febrey & Co (for mail), 616 New York Ave N W, and 2331 Cathedral Ave N W, Washington, D C
FEEHAN, John B (*M* 1923), Pres-Treas (for mail), John B Feehan, Inc, 471 Union St, Lynn, and 4 Ocean View Drive, Marblehead, Mass
FEELY, Frank J. (*M* 1935, *A* 1929), Mgr of Sales, Taylor Supply Co, 700 Monroe Ave, Detroit, and (for mail), 950 Trombley Rd, Grosse Pointe Park, Mich
FEHLIG, John B. (*Life Member, M* 1918), Pres (for mail), Excelsior Heating Supply Co, 528 Delaware St, and 2927 Brooklyn Ave, Kansas City, Mo
FEINBERG, Emanuel (*J* 1937), Dist Sales Engr (for mail), Ilg Electric Ventilating Co, 415 Brainard, and 2935 Tuxedo Ave, Detroit, Mich
FELDMAN, A. M.* (*Life Member, M* 1903), Consulting Engr, 40 West 77th St, New York, N Y.
FELS, Arthur B. (*M* 1919), Pres (for mail), The Fels Co, 42 Union St., Portland and Gilman St, Yarmouth, Maine
FELTWELL, Robert H. (*M* 1905), Htg Engr, U. S. Radiator Corp, 2321 Fourth St N E, and (for mail), 1370 Oak St N W, Washington, D C
FENNER, Everett M. (*M* 1936, *A* 1928), Chief Engr (for mail), Staples Coal Co, 92 Pleasant St, Fall River, Mass., and 719 Pontiac Ave, Cranston, R I.
FENNER, N. Paul (*A* 1928), John G Kelly, Inc, 210 East 45th St, New York, N Y
FENSTERMAKER, Sidney E. (*M* 1909), Pres (for mail), S E Fenstermaker & Co, 937 Architects and Builders Bldg, and 3102 Washington Blvd, Indianapolis, Ind
FERGESTAD, Marvin L. (*J* 1935), Sales Engr (for mail), Insulite Co, 607 Second Ave S, and 3509 S Colfax Ave, Minneapolis, Minn
FERGUSON, Ralph R. (*M* 1934, *A* 1927, *J* 1925), Mgr Air Cond Dept, American Blower Corp, 40 West 40th St, New York, N Y, and (for mail), 160 Prospect St, East Orange, N J
FICNERSKI, Paul (*J* 1936; *S* 1935), 2922 Allen Ave, Chicago, Ill
FIDELIUS, Walter R (*M* 1936), Sales Engr, Fitzgibbons Boiler Co, Inc, 101 Park Ave, New York, and (for mail), 135 Amersfort Place, Brooklyn, N Y.
FIEDLER, Harry W. (*M* 1923), Pres (for mail), Air Conditioning Utilities, Inc, 8 West 40th St, New York, and 77 Hillside Ave, Mt Vernon, N Y.
FIFE, G. Donald (*A* 1931; *J* 1929), Air Cond Engr., Architect of the Capitol, and (for mail), 3221 Connecticut Ave, Washington, D C.
FIGGIS, Thomas G. (*J* 1936), Mgr, Air Cond and Refrigeration Dept. (for mail), Malcolm & Co, Ltd, Wayfoong House, Shanghai, China, and 27 Lovelace Rd, Surbiton, Surrey, England
FILLO, Frank B. (*A* 1934), Dist. Mgr, Minneapolis-Honeywell Regulator Co, 2831 Olive St, St Louis, and (for mail), 252 E. Big Bend Rd, Webster Groves, Mo
FINAN, James J. (*M* 1923), Supervising Engr, Board of Education, 228 N LaSalle St, Room 530, and (for mail), 7149 Euclid Ave, Chicago, Ill.
FINERAN, Edward V (*J* 1935), Asst Engr, Washington Gas Light Co, 411 Tenth St N W, Washington, D C, and (for mail), 821 Dale Drive, Silver Spring, Md
FIRESTONE, James F (*A* 1925, *J* 1914), Exec Vice-Pres, Round Oak Co, Dowagiac, Mich
FISHER, John T. (*J* 1936), Air Cond Engr, National Electrical Supply Co, 1328 New York Ave N W, and (for mail), 3228 Rittenhouse St N W, Washington, D C
FITCH, Howard M. (*J* 1936), Sales Engr, American Air Filter Co, 215 Central Ave, and (for mail), 201 Clare Ave, Louisville, Ky
FITTS, Charles D (*M* 1920), Branch Mgr (for mail), American Radiator Co, 692 Prior Ave, St Paul, and 2807 Dean Blvd, Minneapolis, Minn
FITTS, Joseph C (*M* 1930), Secy, Heating, Piping & Air Conditioning Contractors National Association, 1250 Sixth Ave, New York, N Y., and (for mail), 215 Kenilworth Rd, Ridgewood, N J
FITZGERALD, Matthew J (*M* 1934), Secy-Treas, Standard Asbestos Mfg Co, 820 W Lake St, Chicago, and (for mail), 1117 N Linden Ave, Oak Park, Ill.
FITZGERALD, William E (*J* 1936; *S* 1935), Engr, Fitzgerald Plumbing & Heating Co, Inc, 939-41 Louisiana Ave, and (for mail), 210 Vine St, Shreveport, La
FITZSIMONS, J Patrick (*J* 1934, *S* 1932), Air Cond Engr (for mail), Trane Co of Canada, Ltd, 439 King St. W, and 616 Spadina Ave, Toronto, Canada
FLARSHEIM, Clarence A (*J* 1933), C. A Flarsheim, Inc, 515 Business Mens Assurance Bldg (for mail), P O Box 56, and 3720 Holmes St, Kansas City, Mo
FLEISHER, Walter L.* (*M* 1911), Consulting Engr (for mail), 11 West 42nd St, New York, and New City, N Y
FLEMING, James P. (*M* 1923), Engr-Custodian, Board of Education, 5015 N Kimball Ave, Chicago, Ill
FLEMING, Thomas F. (*J* 1936, *S* 1935), Safety Engr, Liberty Mutual Insurance Co, 122 N. Seventh St, St Louis, Mo, and (for mail), 7004 Eggleston Ave, Chicago, Ill.
FLINK, Carl H. (*M* 1923), Director of Research (for mail), American Gas Products Corp, 108 East 111th St, New York, and 74 Brookside Ave, Mt Vernon, N Y
FLINN, George S. (*J* 1936), Chief Engr, McGregor's, Inc, 680 Union Ave, and (for mail), 100 N Avalon, Memphis, Tenn
FLINT, Coll T (*M* 1919), Sales Mgr (for mail), H B Smith Co, 640 Main St, Cambridge, and 56 Brantwood Rd, Arlington, Mass
FLOYD, Morris (*M* 1933), Vice-Pres., Floyd-Gerdsen Corp., 626 Broadway, and (for mail), The Phelps, 506 E Fourth St, Cincinnati, Ohio
FLYNN, Frank J. (*M* 1936), Secy.-Treas, Missouri Water & Steam Supply Co., 810-20 Sixth St, and (for mail), 920 Ridenbaugh St, St Joseph, Mo.
FOGARTY, Orville A. (*M* 1934), Engr, Rivers Salvage Co, Ltd, 2215 St James St W., Montreal, and (for mail), Rigaud, Que, Canada
FOLLETT, Thomas L. (*S* 1936), 10900 Euclid Ave, Cleveland, and (for mail), 110 Streetsboro St., Hudson, Ohio
FOOTE, Earle E (*M* 1936), Chief Engr, Consumers Central Heating Co, 108 East 11th St, and (for mail), 3412 North 28th St., Tacoma, Wash
FORD, Edward F. (*A* 1937), Sales Repr (for mail), American Radiator Co, 1344 Broadway, and 2075 W Chicago, Detroit, Mich
FORFAR, Donald M. (*M* 1917), Mech Engr, Grinnell Co, Inc, 240 Seventh Ave. S, and (for mail), 4817 Emerson Ave. S., Minneapolis, Minn
FORRESTER, Norman J. (*A* 1936), Chief Estimator, Garth Co, 750 Belair Ave, Montreal, and (for mail), 51 Northview Ave., Montreal W, Que, Canada

FORSBERG, William (*M* 1919), Hopson & Chapin Mfg Co, 231 State St, New London, Conn

FORSLUND, Oliver A (*M* 1936), Gen Mgr, Forslund Pump & Machinery Co, 1717-19 Main St, and (for mail), 1109 West 75th Terrace, Kansas City, Mo

FOSS, Edwin R (*A* 1936), Southern Mgr (for mail), Powers Regulator Co, 509 Bona Allen Bldg, and 257 Bolling Rd N E, Atlanta, Ga

FOSTER, Charles (*M* 1923), Consulting Engr (for mail), 508 Sellwood Bldg, and 2831 E First St, Duluth, Minn

FOSTER, James M (*M* 1930, *A* 1920), Owner (for mail), 1526 Olive St, St Louis, and 7021 Lindell Ave, University City, Mo

FOSTER, Tillman R (*A* 1936, *J* 1930), Carrier Corp, 180 N Michigan Ave, Chicago, Ill

FOULDS, P. A. L. (*M* 1916), Consulting Engr (for mail), Office of Hollis French, 210 South St, Boston, and 72 Whitin Ave, Revere, Mass

FOWLES, Harry H (*J* 1931), Htg-Vtg Engr (for mail), Carman-Thompson Co, 12-14 Lincoln St, and 97 Summer St, Lewiston, Maine

FOX, Edward L. (*J* 1936), 715 Sanford St., Peoria, Ill

FOX, Ernest (*M* 1935), Asst to Engr, C A Dunham Co, Ltd, 1523 Davenport Rd., and (for mail), 141 Westmount Ave., Toronto, Ont, Canada

FOX, John H. (*M* 1935), Sales Engr (for mail), Minneapolis-Honeywell Regulator Co, Ltd, 117 Peter St, and 37 Macdonell Ave, Toronto, Canada.

FRAMPTON, Alfred C. (*S* 1931), 924 N Columbia Rd, Tulsa, Okla

FRANCE, Clarence N. (*A* 1936), Service Mgr (for mail), Colonial Fuel Oil, Inc, 1709 De Sales St. N.W, and 204 Channing St N E, Washington, D C

FRANK, John M (*M* 1918, *A* 1912), Ilg Electric Ventilating Co, 2850 N Crawford Ave., Chicago, Ill

FRANK, Olive E.* (*M* 1919), Pres (for mail), Frank Engineering Co, 11 Park Place, and 610 West 110th St, New York, N Y

FRANKEL, Gilbert S (*M* 1926), Mgr (for mail), Buffalo Forge Co, 820-21 Woodward Bldg, and 2749 Macomb St N.W, Washington, D C

FRANKLIN, Ralph S (*M* 1919), Pres-Treas (for mail), Albert B. Franklin, Inc, 38 Chauncy St, Boston, and 320 Grove St, Melrose, Mass

FRASER, James J. (*A* 1936), Dist Mgr (for mail), Minneapolis-Honeywell Regulator Co, 637 Craig St W, and 4814 Melrose Ave, Montreal, P. Q, Canada

FRAZIER, J. Earl (*A* 1936), Secy.-Treas (for mail), Simplex Engineering Co, 1028 Washington Trust Bldg, and 417 E Beau St, Washington, Pa

FREAS, Royal B (*M* 1928), Vice-Pres, Freas Thermo Electric Co, 1750 N Springfield Ave, Chicago, Ill, and (for mail), Schodack Landing, N. Y.

FREEMAN, John C. (*J* 1936), Engr., 8103 Blackburn Ave, Los Angeles, Calif

FREITAG, Frederic G. (*M* 1932), Chief Engr, Sylvestre Oil Co, Inc, 703 S Columbus Ave, and (for mail), 9 Harrison St., Mt Vernon, N Y.

FRENCH, Donald (*M* 1926), Vice-Pres (for mail), Carrier Corp., 850 Frelinghuysen Ave., Newark, and 45 Waldon Ave, Summit, N. J

FRENTZEL, Herman C. (*M* 1936), Chief Engr., Htg and Water Systems Divs., The Heil Co, 3000 W. Montana St., and (for mail), 4363 N. Wildwood Ave, Milwaukee, Wis

FREY, George O. (*J* 1934), Engr, Const Dept (for mail), Warner Bros. Theatres, Inc, 321 West 44th St., New York, and 221 Linden Blvd., Brooklyn, N. Y

FRIED, Harold V. (*A* 1935), Air Cond Specialist (for mail), Birmingham Electric Co, 2100 N First Ave., and 1585 Druid Hill Drive, Birmingham, Ala

FRIEDMAN, Arthur (*A* 1936), Pres (for mail), Air Controls, Inc, 1933 West 114th St, Cleveland, and 15700 S Moreland Blvd, Shaker Heights, Ohio

FRIEDMAN, D Harry, Jr. (*M* 1936), Engr (for mail), Peoples Water & Gas Co, 637 Washington Ave, and 330 West 58th St, Miami Beach, Fla

FRIEDMAN, Ferdinand J.' (*M* 1921), Consulting Engr, McDougall & Friedman, 1221 Osborne St, Montreal, Que, Canada

FRIEDMAN, Milton (*J* 1935, *S* 1933), 470 West End Ave, New York, N Y

FRIMET, Maurice (*J* 1936), Pres-Owner, Ace Refrigerating Co, 62 Sherman Ave, and (for mail), 28 Oxford Place, Tompkinsville, S I, N Y

FRISSE, John L. (*S* 1935), 5538 Forbes St, Pittsburgh, Pa

FRITZ, Charles V (*J* 1936; *S* 1933), Designer and Estimator, Chas F Fritz (for mail), 67 W Merrick Rd, and 26 Cottage Court, Freeport, N Y

FRITZBERG, L H (*A* 1936, *J* 1931), Asst Mgr, Extended Surface Dept (for mail), B F Sturtevant Co, Hyde Park, and 1338 River St, Hyde Park, Boston, Mass

FUKUI, Kunitaro (*M* 1926), Auditor (for mail), Oriental Carrier Engrg Co, Ltd, Osaka Mitsui Bldg, Nakanoshima, Osaka, Japan

FULLER, James H. (*S* 1936), 304 E Main St, Frankfort, N Y, and (for mail), 1310 Chase Ave, Lakewood, Ohio

G

GALE, Hamilton A. (*J* 1936), Sales Engr, Hudson Air Conditioning Corp, 1517 Connecticut Ave, Washington, D C, and (for mail), Murray Hill, Annapolis, Md

GALLAGHER, Paul (*S* 1935), 1209 Sixth St, Peru, and (for mail), 902 W California, Urbana, Ill

GALLIGAN, Andrew B. (*M* 1921), 716 South 51st St, Philadelphia, Pa

GALLOWAY, James F. (*S* 1934), 176 Clarkson Ave, Brooklyn, N Y

GAMBLE, Cary B. (*A* 1935), Consulting Engr. (for mail), Leo S Weil & Walter B Moses, 427 S Peters St, and 732 St Peter St., New Orleans, La

GAMMILL, Oscar E., Jr (*J* 1930), Sales Engr. (for mail), Carrier Corp, 1416 Hibernia Bank Bldg, and 2133 Calhoun St, New Orleans, La

GANT, H. P.* (*M* 1915), (*Presidential Member*), (Pres, 1923; 1st Vice-Pres, 1922, 2nd Vice-Pres, 1921, Council, 1918-1924), Vice-Pres (for mail), Carrier Corp, 12 South 12th St, Philadelphia, and R D 1, Glen Moore, Pa

GARDNER, S. Franklin (*M* 1911), Pres (for mail), Standard Engrg Co, 2129 Eye St N W, and 4901 Hillbrook Lane, Washington, D. C

GARDNER, William (*A* 1921), Vice-Pres. (for mail), Garden City Fan Co., 1842 McCormick Bldg, and 7836 Loomis Blvd., Chicago, Ill

GARNEAU, Leo (*J* 1930), Repr, C A Dunham Co, Ltd, Room 743 Dominion Square Bldg, and (for mail), 2541 Maplewood Ave., Apt 2, Montreal, P Q, Canada

GARNETT, Ralph E. (*A* 1936), Sales Engr (for mail), Standard Asbestos Mfg. & Insulating Co, 10 N Olive St., and 3114 Benton Blvd., Kansas City, Mo

GATES, Robert A. (*M* 1936), Sole Owner, Gates Engineering Co, 510-77 Sheet St., and (for mail), 248 Bay 38th St, Brooklyn, N Y

GAULEY, Ernest R. (*A* 1935), Salesman (for mail), W H Cunningham & Hill, Ltd, 269-71 Richmond St W., and 110 Lee Ave, Toronto, Ont, Canada

GAULT, George W. (*S* 1934), Firestone Tire & Rubber Co, and (for mail), 200 Cole Ave, Akron, Ohio.

GAUSMAN, Carl E. (*M* 1923), Consulting Engr, Gausman & Moore, 1526 First National Bank Bldg E, and (for mail), 2360 Chilcombe Ave, St Paul, Minn

GAWTHROP, Fred H. (*M* 1919), Pres, Gawthrop & Bro Co, 705 Orange St, and (for mail), 2211 Shallcross Ave, Wilmington, Del

GAY, Lewis M. (*A* 1934), Power Engr (for mail), Texas Power & Light Co, Box 962, Dallas, and 806 First Ave, Terrell, Texas

GAYLOR, William S. (*M* 1919), Vice-Pres, Flameking Co, Inc, 450 North Ave, New Rochelle, and (for mail), 6 West Ave, Larchmont, N Y

GAYLORD, Frank H (*M* 1921), Western Sales Mgr (for mail), Hoffman Specialty Co, Inc, 130 N Wells St, Chicago, and 362 N York, St, Elmhurst, Ill

GEIGER, Irvin H (*M* 1919), Registered Prof Engr and Mfrs Repr (for mail), 319 Telegraph Bldg (P O B 83), and 240 Maclay St, Harrisburg, Pa

GEISSBUHLER, John O. (*J* 1936, *S* 1934), Student Engr, General Electric Co, Glass Machine, 1133 East 152nd St, and (for mail), 9820 Zimmer Ave, Cleveland, Ohio

GELB, Amiel (*S* 1935), 1042 Irving Ave N, Minneapolis, Minn

GELTZ, Ralph W. (*J* 1936), Air Cond Engr, York Ice Machinery Corp, 2700 Washington Ave N W, and (for mail), 2541 Abington Rd, Cleveland, Ohio

GEMMILL, Robert A. (*A* 1935), Mgr, Norge Oil Burner Div, Trilling & Montague, 2409 Walnut St, and (for mail), 4710 Locust St, Philadelphia, Pa

GERMAIN, Oscar (*M* 1935), Foreman, Germain & Frere, Ltd, 237 St Antoine St, and (for mail), 1343 Blvd St Louis, Three Rivers, P Q, Canada

GERRISH, Grenville B. (*A* 1936, *J* 1930), Mgr (for mail), Fitzgibbons Boiler Co, Inc, 31 Main St, Cambridge, and 89 Warwick Rd, Melrose Highlands, Mass

GERRISH, Harry E. (*M* 1910), (Council, 1919), Partner (for mail), Morgan-Gerrish Co, 84 S Tenth St, 307 Essex Bldg, and 4534 Fremont S, Minneapolis, Minn

GETSCHOW, Roy M. (*M* 1919), Secy (for mail), Phillips-Getschow Co, 32 W Austin Ave, and 1336 Arthur Ave, Chicago, Ill

GIANNINI, Mario C (*M* 1935), Asst Prof of Mech Engrg, New York University, University Heights, New York, and (for mail), 31 French Ridge, New Rochelle, N Y

GIBBONS, Michael J. (*M* 1914), Owner, M J. Gibbons Supply Co, 601-31 E Monument Ave, and (for mail), 22 Oxford Ave, Dayton, Ohio

GIBBS, Edward W. (*M* 1919), Owner, The Smith-Gibbs Co (for mail), 201 S Main St, and 39 President Ave, Providence, R I

GIBBS, Frank C. (*M* 1921), Gen Supt (for mail), National Regulator Co, 2301 N Knox Ave, Chicago, and 150 N Cuyler Ave, Oak Park, Ill

GIESECKE, Frederick E.* (*M* 1913), (Council, 1932-1936), Director, Texas Engrg Experiment Station, Agricultural and Mechanical College, College Station, Texas

GIFFORD, Clarence A. (*A* 1934), Salesman, American Radiator Co, 374 Delaware Ave, and (for mail), 758 Parkside Ave, Buffalo, N Y

GIFFORD, Robert L. (*M* 1908), Pres, Illinois Engineering Co, 21st St and Racine Ave, Chicago, Ill, and (for mail), 1231 S El Molino Ave, Pasadena, Calif

GIFFORD, William R (*J* 1936), Sales Engr, American Radiator Co, Fourth and Channing Sts N E, Washington, D C, and (for mail), Box 295, College Park, Md

GIGUERE, George H. (*M* 1920), Consulting Engr, 17205 Fairport Ave, Detroit, Mich

GILES, Alfred F. (*M* 1936, *J* 1934), Owens-Illinois Glass Co, Alton, Ill

GILES, J. C. (*S* 1935), 546 South Blvd, Norman, Okla

GILFRIN, George F. (*M* 1932), Climas Artificiales, S A (for mail), Edificio "La Nacional" 608, and Esplanada No 715 Lomas de Chapultepec, Mexico, D F.

GILL, Eric F (*M* 1936), Chief Draftsman, Drayton Regulator & Instrument Co, Ltd, and (for mail), 30 Warwick Rd, West Drayton, Middlesex, England

GILL, John W (*A* 1936, *S* 1935), Box 255, Waterloo, Iowa

GILLE, Hadar B. (*M* 1930), Consulting Engr, Hugo Theorells Ingeniorsbyra, Skoldungagatan 4, Stockholm, Sweden

GILLETT, M. C. (*M* 1916), Engr, Hoffman Specialty Co, 500 Fifth Ave, New York, N Y, and (for mail), 6600 Rising Sun Ave, Philadelphia, Pa

GILLHAM, Walter E (*M* 1917), (Treas, 1926-1929, Council, 1926-1929), Consulting Engr (for mail), 314 Inter-State Bldg, and 3127 Bellefontain Ave, Kansas City, Mo

GILLING, William F. (*Life Member* 1934, *M* 1933, *A* 1919), Asst Mgr, American Radiator Co, 127 Federal St, Boston, and (for mail), 29 Abbott Rd, Wellesley Hills, Mass

GILMAN, Franklin W. (*M* 1935), Plant Engr (for mail), Atwater Kent Mfg Co, 1700 Wissahickon Ave, and 514 W Coulter St, Philadelphia, Pa

GILMORE, Louis A (*J* 1935, *S* 1930), Vice-Pres (for mail), John Gilmore & Co, 13 N. Tenth St, and 6186 Westminster Place, St Louis, Mo

GILMOUR, Alan B. (*A* 1932), Salesman, B F Gilmour Co, Inc, 152-41st St, and (for mail), 543 East 18th St, Brooklyn, N Y

GINI, Aldo (*M* 1933), via Correggio 18, Milano, Italy

GINN, Tony M (*M* 1935), Gen Mgr (for mail), Tony M Ginn Co, 211-21 Fifth St S, and 608 Sixth Ave S, Great Falls, Mont

GITTLESON, Harold (*A* 1936), Sales Mgr, Lanviere, Inc, 3715 St. Lawrence Blvd, and (for mail), 1125 Lajoie Ave, Montreal, Que, Canada

GIVIN, Albert W. (*A* 1925), Vice-Pres (Stove Sales), Gurney Foundry Co, Ltd (for mail), Box 1119, and Apt 4-S, 4870 Cote-des-Neiges Rd, Montreal, P Q, Canada

GLASS, William (*M* 1934), Mgr (for mail), Partridge-Halliday, Ltd, 141 Lombard St, Winnipeg, and 190 Braemar Ave, Norwood, Manitoba, Canada

GLASSEY, J. Wilbur (*M* 1922), Partner (for mail), Vapor Engineering Co, 10 South 18th St, Philadelphia, and 7818 Ardleigh St, Chestnut Hill, Philadelphia, Pa

GLEASON, Gilbert H. (*M* 1923), Partner (for mail), Gilbert Howe Gleason & Co, 25 Huntington Ave, Boston, and 10 Edgehill Rd, Winchester, Mass

GLORE, Evins F. (*A* 1916), 644 Riverside Drive, New York, N Y

GODDARD, William F (*A* 1936), Gov't Repr, American Radiator Co, Fourth and Channing Sts N E, and (for mail), Apt 34, 3150-16th St N W, Washington, D C.

GOELZ, Arnold H. (*M* 1931), Pres.-Treas (for mail), Kroeschell Engineering Co., 215 W Ontario St, Chicago, and 827 Greenwood Ave, Wilmette, Ill

GOENAGA, Roger C. (*M* 1931), Tech Director, Ateliers Ventil (for mail), 109 Cours Gambetta, Lyon, and 33 Avenue Valioud-Ste-Foy-les-Lyon, Rhone, France.

GOERG, Bernhard (*M* 1928), Director (for mail), American Radiator Co, 675 Bronx River Rd, Yonkers, and 294 Bronxville Rd, Bronxville, N Y

GOLDBERG, Moses (*A* 1934), Pres, Electric Motors Corp, 168 Centre St, New York, and (for mail), 1311 E Seventh St, Brooklyn, N Y

GOLDSCHMIDT, Otto E. (*M* 1915), Consulting Engr (for mail), 22 East 40th St, New York, N Y, and Greens Farms, Conn

GOLDSMITH, F. Willius (*M* 1930), Pres (for mail), W Clasmann Co, 321 E Wisconsin Ave, and 629 E Day Ave., Milwaukee, Wis

GOMBERS, Henry B (*Life Member*, *A* 1901), Secy Emeritus, Heating, Piping and Air Conditioning Contractors National Association, 1250 Sixth Ave , New York, N Y , and (for mail), 160 Halsted St , East Orange, N J

GONZALEZ, Rafael A (*M* 1936), Mgr , Application Engrg (for mail), Airtemp, Inc , Leo St , and 927 Salem Ave , Dayton, Ohio

GOODMAN, Daniel J (*S* 1935), 2704 Filbury St , Pittsburgh, Pa

GOODRAM, William E (*A* 1930), Partner, Goodram Bros , 88 King St W , Hamilton, and (for mail), R R 2, Freeman, Ont , Canada

GOODRICH, Charles F (*M* 1919), Andrews & Goodrich, Inc , Boston, and (for mail), 336 Adams St , Dorchester, Mass

GOODWIN, Eugene W. (*M* 1936), Sr Mech Engr , U S Treasury Dept , Procurement Bldg , Washington, D C , and (for mail), 7024 Hampden Lane, Bethesda, Md

GOODWIN, Samuel L. (*M* 1921), Consulting Engr , 217 Madison Ave , Hasbrouck Heights, N J

GOODWIN, Walter C (*M* 1933), Div Engrg , Air Cond Equip (for mail), Supply Engrg Dept , Westinghouse Electric & Mfg Co , East Pittsburgh, and 6032 Marie St , Pittsburgh, Pa.

GORDON, Edward B., Jr (*M* 1908), Pres , Pillsbury Engineering Co , 1200 Second Ave S , and (for mail), 2150 West 24th St , Minneapolis, Minn

GORDON, Peter B. (*J* 1935), Engr (for mail), Wolff & Munier, Inc , 222 East 41st St , New York, N Y , and 35 Park Ave , Bloomfield, N J

GORDON, William D. (*A* 1935), Engr. (for mail), Tuttle & Bailey Mfg Co of Canada Ltd , and 25 Bowen Rd , Fort Erie N , Ont , Canada

GORDON, William J , Jr. (*J* 1936; *S* 1935), 2208 Oliver Ave S , Minneapolis, Minn

GORNSTON, Michael H. (*A* 1923), Stationery Engr (for mail), Thomas Jefferson High School, 102 Pennsylvania Ave , Brooklyn, and 90-11–119th St , Jamaica, N Y

GOSSETT, Earl J (*M* 1923), Pres (for mail), Bell & Gossett Co , 3000 Wallace St , Chicago, and 311 Woodland Ave , Winnetka, Ill

GOTHARD, William W. (*A* 1936), Managing Editor (for mail), Domestic Engineering, 1900 Prairie Ave , Chicago, and 1027 Arlington Ave , LaGrange, Ill

GOTSCHALL, Harry C. (*M* 1935), Air Cond Instructor, Lane Technical High School, Western Ave and Addison St., and (for mail), 4706 N. Artesian Ave , Chicago, Ill

GOTTWALD, C. (*A* 1916), Pres. (for mail), Ric-Wil Co., Union Trust Bldg , Cleveland, and 2225 Stillman Rd , Cleveland Heights, Ohio

GOUEDY, Kenneth E. (*A* 1935), Salesman and Engr (for mail), Modern Building Insulating Co , 510 Bona Allen Bldg., Atlanta, and 218 Columbia Drive, Decatur, Ga

GOULD, Henry E. (*J* 1936), Sales Engr (for mail), Natkin & Co , 2020 Wyandotte St , and 6021 Locust St , Kansas City, Mo.

GOULDING, William (*A* 1933), Air Cond Engr , World Broadcasting Co , 711 Fifth Ave , New York, and (for mail), 360 East 19th St., Brooklyn, N Y

GRABENSTEDER, Louis (*J* 1935), Power Sales Engr , Union Gas & Electric Co , and (for mail), 3620 Stettinius Ave , Cincinnati, Ohio

GRABER, Ernst (*J* 1936), Engr , Minneapolis-Honeywell Regulator Co , 801 Second Ave , New York, and (for mail), 42-12 Ditmars Blvd , Astoria, L. I , N. Y.

GRAHAM, Charles H (*M* 1934), Sales Repr , Lennox Furnace Co , Inc., 400 N. Midler Ave., Syracuse, and (for mail), 377 Highland Ave , Hamburg, N. Y

GRAHAM, Earl W. (*J* 1935), Student Engr. (for mail), Carrier Corp., 180 N Michigan, Chicago, Ill , and Bristow, Ky

GRAHAM, John M. (*J* 1936), Sales Engr (for mail), B F. Sturtevant Co., 913 Provident Bank Bldg , and Kemper Lane Hotel, Cincinnati, Ohio

GRAHAM, William D (*M* 1929; *A* 1925; *J* 1923), Branch Mgr (for mail), Carrier Corp , 795 Union Trust Bldg , Cleveland, and 3179 Chadbourne Rd , Shaker Heights, Ohio

GRAHN, Victor F. (*M* 1927), Htg -Vtg Engr , Tenney & Ohmes, Inc , 101 Park Ave , New York, N Y , and (for mail), 120 Greenwood Ave , East Orange, N J

GRANSTON, Ray O. (*J* 1935, *S* 1930), Engr , University Plumbing & Heating Co , 3939 University Way, and (for mail), 4558 Fourth Ave N E , Seattle, Wash

GRANT, Walter A. (*A* 1933, *J* 1929), Dist Chief Engr , Carrier Corp , Newark, and (for mail), 1142 Anna St , Elizabeth, N J

GRAVES, Willard B (*Life Member*, *M* 1906), Pres , W B Graves Heating Co , 162 N. Desplaines St , Chicago, Ill

GRAY, Earle W (*A* 1934), In charge of Air Cond , Commercial Dept (for mail), Oklahoma Gas & Electric Co , Third and Harvey Sts , and 2125 Northwest 18th St , Oklahoma City, Okla

GRAY, Everett W. (*M* 1936), Mgr (for mail), The Trane Co , 1900 Euclid Ave , Cleveland, and 17545 Madison Ave , Lakewood, Ohio

GRAY, George A. (*M* 1924), Branch Mgr (for mail), C A Dunham Co , Ltd , 404 Plaza Bldg , and 114 Belmont Ave , Ottawa, Ont , Canada.

GRAY, William E. (*M* 1922), Sales Engr , Powers Regulator Co , 2720 Greenview Ave , Chicago, Ill , and (for mail), Box 264, High Point, N C

GREBEN, David (*M* 1936), Mech Engr , 339 West 58th St , New York, N Y

GREEN, Arthur W. (*A* 1935), 37-60–88th St , Jackson Heights, L I , N Y

GREEN, William C (*Life Member*, *M* 1906), Dist Mgr (for mail), Warren Webster & Co , 704 Race St , Room 602-5, and 244 Erkenbrecher Ave , Cincinnati, Ohio

GREENBURG, Dr. Leonard* (*M* 1932), Exec Director, Div of Industrial Hygiene (for mail), New York State Department of Labor, 80 Centre St , and 241 West 97th St , New York, N Y

GREENLAND, Sidney F. (*M* 1934), Engr , Gee, Walker & Slater, Ltd , Fitzmaurice Place, Berkeley Square, London W 1, and (for mail), 71 Arodene Rd , Brixton, London, S W 2, England.

GREER, Willis R (*J* 1934), Air Cond Engr , Arkansas Power & Light Co , and (for mail), 1401 Linden St , Pine Bluff, Ark

GREGG, Scranton H. (*A* 1936), Pres , Shellenberger-Gregg Co , 2203 N Prospect Ave , and (for mail), 5134 N Woodburn St , Milwaukee, Wis

GREGG, Stephen L. (*J* 1936), Sales Engr (for mail), Potomac Electric Power Co , Tenth and E Sts. N W , and 3614 Connecticut Ave N W , Washington, D C

GREINER, George E , Jr (*S* 1935), 2149 Webster Ave , Pittsburgh, Pa

GRIESSER, Charles E. (*A* 1936), Owner, Electric Contractor Dealer, Bryan, Texas

GRIEST, Kermit (*J* 1936), Sheet Metal Worker and Sales, Frank-Limbach & Co , 1722 E Ohio St , and (for mail), 134 Groveland St., Pittsburgh, Pa

GRIMES, Fenner M. (*J* 1935), Junior Engr., T H Urdahl, Consulting Engr , 726 Jackson Place N W , and (for mail), 7705 Alaska Ave. N W , Washington, D C.

GROSECLOSE, John B. (*A* 1929), Engr -Estimator, Dixie Heating & Ventilating Co., 109 Fannin St , and (for mail), 3424 University Blvd , Houston, Texas

GROSS, Lyman C. (*M* 1931), Sales Engr , Minneapolis-Honeywell Regulator Co., 2753 Fourth Ave. S , and (for mail), 4653–13th Ave S , Minneapolis, Minn

GROSSMAN, Harry E. (*A* 1933; *J* 1927), Sales Repr , Haynes Selling Co., Inc., 1518 Fairmount Ave , Philadelphia, and (for mail), 218 Parkam Rd., Springfield, Pa.

GROSSMANN, Harry A. (*M* 1931), Owner, H A. Grossman Co, 3221 Olive St, and (for mail), 3122 Geyer Ave, St Louis, Mo

GROVES, Samuel A. (*J* 1935), Salesman, American Radiator Co, 40 West 40th St, New York, and (for mail), 70-10 Continental Ave, Forest Hills, N Y

GUNTHER, Felix A.† (*M* 1925), Sales Engr (for mail), 429-B Oliver Bldg, and Box 226, R D 9, S Hills Branch, Pittsburgh, Pa

GURNEY, Edward H (*M* 1929), (2nd Vice-Pres, 1936, Council, 1931-1936), Pres (for mail), Gurney Foundry Co, Ltd, 4 Junction Rd, and 347 Walmer Rd, Toronto, Ont, Canada.

GURNEY, Edward R. (*J* 1937), Asst Engr, Gurney Foundry Co, Ltd, and (for mail), 33 Oriole Parkway, Toronto, Canada

H

HAAS, Emil, Jr (*A* 1936, *J* 1929), Secy-Treas (for mail), Natkin & Co, 2020 Wyandotte and Newbern Hotel, Kansas City, Mo

HAAS, Richard B. (*S* 1935), Htg Engr, c/o L C Pemberton, 406 S Washington Ave, Lansing, Mich.

HAAS, Samuel L. (*M* 1923), Pres (for mail), Advance Heating Co, 117-19 N Desplaines St, and 1513 Fargo Ave, Chicago, Ill

HAATVEDT, Sheldon R. (*J* 1936, *S* 1935), 408 Fourth St S E., Little Falls, Minn

HACKETT, H. Berkeley (*M* 1921), Consulting Engr, 901 Architects Bldg, 17th and Sansom Sts, Philadelphia, Pa

HADDOCK, Isaac T. (*A* 1926), New England Gas & Electric Association, 719 Massachusetts Ave, Cambridge, Mass.

HADEN, G. Nelson (*M* 1934, *A* 1928, *J* 1922), Managing Director (for mail), G N Haden & Sons, Ltd, 60 Kingsway, London, W C 2, and 36 Wildwood Rd, Hampstead Heath, London, N W 11, England

HADEN, William N (*Life Member*, *M* 1902), Late Chairman, G N Haden & Sons, Ltd, St Georges Works, and (for mail), Arnolds Hill, Trowbridge, Wilt, England.

HADJISKY, Joseph N. (*M* 1930), Consulting Engr, 744 Bates St, Birmingham, Mich

HAGAN, William V. (*A* 1933; *J* 1926), Secy, V. J. Hagan Plumbing & Heating Co, 506 Pearl St, and (for mail), 1811 Jones St, Sioux City, Iowa.

HAGEDON, Charles H. (*M* 1919), Secy-Treas (for mail), S. E Fenstermaker & Co, 937 Architects and Builders Bldg, and 4156 Broadway, Indianapolis, Ind

HAHN, Roy F. (*J* 1936), Engrg Dept (for mail), Delco-Frigidaire Conditioning Corp, 675 Greenwood Ave., and 20–11th St N E., Atlanta, Ga

HAINES, John J. (*M* 1915), Pres (for mail), The Haines Co., 1933 W Lake St., Chicago, and 623–17th Ave, Maywood, Ill

HAJEK, William J. (*M* 1932), Resident Vice-Pres (for mail), Minneapolis-Honeywell Regulator Co, 285 Columbus Ave, and 56 Commonwealth Ave, Boston, Mass

HAKES, Leon M (*M* 1932, *J* 1929), Sales Engr (for mail), R T Coe Co, 410 Reynolds Arcade Bldg, and 71 Strathmore Drive, Greece, Rochester, N Y

HALE, Fred J. (*M* 1936), Mgr (for mail), Empire Sheet Metal Works, Ltd, 1606 W. First Ave, and 1080 Gilford St, Vancouver, B C, Canada

HALE, John F. (*M* 1902), (*Presidential Member*), (Pres, 1913; 1st Vice-Pres, 1912, Board of Governors, 1908-1910, 1912-1913), Dist Mgr (for mail), Aerofin Corp, 1058, 111 W Washington St, Chicago, and 416 S Brainard Ave, LaGrange, Ill

HALEY, Harry S.* (*M* 1914), Consulting Engr, Partner (for mail), Leland & Haley, 58 Sutter St., and 735–21st Ave, San Francisco, Calif

HALL, John R. (*J* 1932), Mech Engr, U S Air Conditioning Corp, 2101 N E Kennedy St, and (for mail), 1416 Lakeview Ave, Minneapolis, Minn

HALL, Mora S (*M* 1931), Development Engr (for mail), May Oil Burner Corp, Maryland and Oliver St, Baltimore, and R F D 3, Westminster, Md

HALLER, Arthur L (*M* 1920), Pres-Treas (for mail), Haller Appliance Sales Co, Inc, 3321 Washington Blvd, St Louis, and 121 W Cedar Ave, Webster Groves, Mo

HAMAKER, Ambrose C (*J* 1937), Mgr Home Htg Dept (for mail), J L Hudson Co, and 18624 Santa Rosa Drive, Detroit, Mich

HAMENT, Louis (*A* 1933), Mgr (for mail), Aquatic Chemical & Metallurgical Engrs, 114 East 32nd St, and 568 East 166th St, New York, N Y

HAMERSKI, Francis D. (*J* 1934), Mgr (for mail), Winona Coal Co, Front and High Forest Sts, and 626 E Fifth St, Winona, Minn

HAMIG, Louis L (*J* 1935), Engr, Controlled Air Corp, 3319 Olive St, and (for mail), 3514 Utah St, St Louis, Mo

HAMILTON, James E (*A* 1933), Branch Mgr (for mail), U S Radiator Corp, 4004 Duncan Ave, St Louis, and 7701 Shirley Drive, Clayton, Mo

HAMJE, Milton C. (*J* 1930), Engr, Syska & Hennessy, Consulting Engrs, 420 Lexington Ave, New York, and (for mail), 198 Hancock St, Brooklyn, N Y

HAMLET, Francis A. (*A* 1936), Branch Mgr (for mail), C A Dunham Co, Ltd, Room 931, Dominion Square Bldg, 1010 St Catherine St W, and 3550 Shuter St, Montreal, Que, Canada

HAMLIN, Chauncey J., Jr. (*A* 1931), Hamlin Air Conditioning Co, and (for mail), 1014 Delaware Ave, Buffalo, N Y

HAMLIN, Harry A. (*A* 1916), Sales Engr. (for mail), Johnson Service Co, 427 Brainard St, Detroit, and 120 Winona Ave, Highland Park, Mich

HANCE, W. Wayne (*A* 1935), Draftsman, E I. DuPont de Nemours Co, Wilmington, Del, and (for mail), 711 Anderson Ave, Drexel Hill, Pa

HANKINS, Richard P. (*J* 1935), Engr and Draftsman (for mail), G A. Peple, Jr, Consulting Engr, 925 American Bldg, and 1826 Hanover Ave, Richmond, Va

HANLEY, Edward V. (*A* 1933), Pres. (for mail), S V Hanley Co, 1653 N. Farwell Ave., Milwaukee, and 844 E Birch Ave, Whitefish Bay, Wis

HANLEY, Thomas F., Jr. (*M* 1933), Pres. (for mail), Hanley & Co, 1503 S Michigan Ave, and 4940 East End Ave, Chicago, Ill.

HANSEN, Charles C. (*M* 1928), Engr, 428 Prospect St, South Orange, N J

HANSON, Leslie P. (*A* 1936; *J* 1935; *S* 1933), Engr, U S Air Conditioning Corp, 2101 Kennedy N E, and (for mail), 4330-46th Ave S, Minneapolis, Minn

HARBORDT, Otto E. (*A* 1930), Sales Mgr (for mail), U. S Supply Co., 1315 West 12th St, and 303 Brush Creek Blvd., Kansas City, Mo.

HARDING, Edward R. (*M* 1936), Branch Sales Mgr and Engr (for mail), Kewanee Boiler Corp, P O Box 536, 704 Jefferson Bldg, and 2603 Sherwood St., Greensboro, N C.

HARDING, Louis A.* (*M* 1911), (*Presidential Member*), (Pres, 1930, 1st Vice-Pres, 1929; 2nd Vice-Pres, 1928, Council, 1922-1931), (for mail), L A Harding Construction Corp, Prudential Bldg, and 85 Cleveland Ave, Buffalo, N. Y.

HARE, W. Almon (*M* 1930), 307 Victoria Rd, Windsor, Ont, Canada

HARMONAY, William L. (*A* 1935), Mgr (for mail), M. J. Harmonay, Inc, 124 Elm St., and 34 Alida St., Yonkers, N Y

HARRIGAN, Edward M. (*M* 1915), Gen Mgr (for mail), Harrigan & Reid Co, 1365 Bagley Ave, and 7450 LaSalle Blvd, Detroit, Mich.

HARRINGTON, Charles (*M* 1923), 43 Indian Grove, Toronto, Ont, Canada.

HARRINGTON, Elliott (*M* 1932, *A* 1930), Mgr, Commercial Engrg Div, Air Cond Dept (for mail), General Electric Co, 5 Lawrence St, Bloomfield, and 5 Wilson Terrace, Caldwell, N J

HARRIS, Jesse B (*M* 1918), Pres (for mail), Rose & Harris Engineers, Inc, 416 Essex Bldg, and 3620 Colfax Ave S, Minneapolis, Minn

HARRIS, John G (*M* 1936), Dist Chief Engr (for mail), Delco-Frigidaire Conditioning Corp, 211 Prospect Ave N W, Terminal Tower Bldg, Cleveland, and 11132 Delaware Ave, Lakewood, Ohio.

HART-BAKER, Henry W (*M* 1918), Director (for mail), Merritt, Ltd, 8 French Bund, and 37 Rte Rene, Delastre, Shanghai, China

HARSCH, Richard J (*M* 1936), Assoc Naval Archt, U S Government, and 112 Avenue O, Brooklyn, N Y

HART, Harry M† (*M* 1912), (*Presidential Member*), (Pres, 1916, 1st Vice-Pres, 1915; Council, 1911-1917), Pres (for mail), L H Prentice Co, 1018 Van Buren St, and 3730 Lakeshore Drive, Chicago, Ill

HARTIN, William R, Jr (*J* 1935), Htg Engr, W R Hartin & Son, 830 Harden St, and (for mail), 212 S Saluda Ave., Columbia, S C.

HARTLINE, William R (*A* 1930), Vice-Pres. and Treas (for mail), Combustioneer Stoker Corp, 409 Tenth St S.W, and 3112 Mt. Pleasant St N W, Washington, D C.

HARTMAN, Fred S (*A* 1933), Dist. Mgr, Industrial Dept (for mail), General Electric Co, 570 Lexington Ave, New York, N. Y, and 168 Montclair Ave, Montclair, N J

HARTMAN, John M (*M* 1927), Engr (for mail), Kewanee Boiler Corp and (for mail), 719 Henry St, Kewanee, Ill.

HARTON, A J. (*A* 1935), Sales Engr., St Joseph Railway, Light, Heat & Power Co, Sixth and Francis, and (for mail), 701 E Hyde Park, St Joseph, Mo

HARTWEIN, Charles E. (*M* 1933), Supervisor, House Htg Dept., St Louis County Gas Co, 231 W Lockwood, Webster Groves, and (for mail), 6271 Magnolia Ave., St Louis, Mo.

HARTWELL, Joseph C. (*M* 1922), Pres. (for mail), Hartwell Co, Inc, 87 Weybosset St, and 16 Freeman Parkway, Providence, R I

HARVEY, Alexander D. (*A* 1928; *J* 1925), Sales Mgr. (for mail), Nash Engineering Co., Wilson Rd., South Norwalk, and West St, New Canaan, Conn.

HARVEY, Lyle C. (*M* 1928), Vice-Pres (for mail), Bryant Heater Co., 17825 St Clair Ave, and 3388 Glencairn Rd., Cleveland, Ohio

HARVEY, Robert A. (*S* 1936), 3200 Braemar Rd, Shaker Heights, Ohio.

HASHAGEN, John B. (*M* 1930), 121 Manhattan Ave, Jersey City, N J

HATEAU, William M. (*J* 1934), Draftsman and Designer, J O. Ross Engineering Corp., 350 Madison Ave., and (for mail), 1530 Sheridan Ave., New York, N Y

HATTIS, Robert E. (*M* 1926), Consulting Engr. (for mail), 820 N. Michigan Ave., and 4251 N. Mozart St., Chicago, Ill

HAUAN, Merlin J. (*M* 1933), Consulting Engr, 3112 10th S., Seattle, Wash

HAUCK, Elden L. (*J* 1936), Sales Mgr and Engr (for mail), Hauck Brothers, 232 S Center St., Springfield, and 54 S. June St., Dayton, Ohio

HAUS, Irvin J. (*J* 1935), Commercial Engr, Morley Murphy Co., 342 Water St, Milwaukee, and (for mail), 925 Division St, Green Bay, Wis.

HAUSMAN, Louis M. (*M* 1935), Pres, L M. Hausman & Co, 410 Dasmarinas, and (for mail), P. O Box 1729, Manila, P. I.

HAUSS, Charles F.* (*Charter Member; Life Member*), Via Giobert No 2, Milano, Italy

HAWK, Joseph K. (*J* 1936), Engr, Hudson Air Conditioning Corp, 1517 Connecticut Ave, and (for mail), 4726 Third St, Washington, D C

HAWKINSON, C. F. (*J* 1936), Mech Engr, U S Air Conditioning Corp, 2101 N E Kennedy St, and (for mail), 1805 Columbus Ave, Minneapolis, Minn

HAYDEN, Carl F (*A* 1930), Sales Engr and Dist Repr (for mail), Askania Regulator Co, 1603 S Michigan Ave, Chicago, and 2227 Ewing Ave, Evanston, Ill

HAYES, James J. (*M* 1920), Sales Engr (for mail), Stannard Power Equipment Co, Room 925–53 W Jackson Blvd, and 7443 Jeffery Ave, Chicago, Ill

HAYES, Joseph G (*M* 1908), Pres and Engr (for mail), Hayes Bros, Inc, 236 W Vermont St, and 2849 N Capitol Ave, Indianapolis, Ind

HAYMAN, A. Eugene, Jr (*J* 1935, *S* 1930), Engr, 2500 Washington St, Wilmington, Del

HAYNES, Charles V (*M* 1917), (*Presidential Member*), (Pres, 1934, 1st Vice-Pres, 1933, 2nd Vice-Pres, 1932; Council, 1926-1929; 1932-1935), Vice-Pres, Hoffman Specialty Co, 500 Fifth Ave, Room 3324, New York, N Y, and (for mail), 115 Llanfair Rd, Ardmore, Mont Co, Pa

HAYTER, Bruce (*M* 1934), Chief Engr, Institute of Thermal Research (for mail), American Radiator Co, 675 Bronx River Rd, Yonkers, and 114 Birchall Drive, Scarsdale, N Y

HAYWARD, Ralph B. (*M* 1909), Pres (for mail), R B Hayward Co, 1714 Sheffield Ave, Chicago, and 201 S Stone Ave, LaGrange, Ill

HEARD, John A E (*J* 1930), Asst Mgr (for mail), Carrier Corp, Ltd, Connaught Place, New Delhi, India, and 28, Leighcliff Rd, Leigh on Sea, Essex, England

HEARD, Roderick G. (*A* 1933), Oil Burner Sales Dept (for mail), Imperial Oil, Ltd, 56 Church St., and 158 Highbourne Rd, Toronto, Ont, Canada

HEATH, William R (*M* 1931), Asst Chief Engr, Buffalo Forge Co, 490 Broadway, and (for mail), 119 Wingate Ave, Buffalo, N Y

HEBERLING, C W (*A* 1934), Box 115, Wayzata, Minn

HEBLEY, Henry F J. (*M* 1934), Advisory Engr, Commercial Testing & Engineering Co, 360 N Michigan Ave, and (for mail), 636 Wrightwood Ave, Chicago, Ill.

HECHT, Frank H. (*M* 1930), Sales Engr (for mail), B F Sturtevant Co, 2635 Koppers Bldg, and 1467 Barnesdale St, Pittsburgh, Pa

HECKEL, E P. (*M* 1918), Vice-Pres, Carrier Corp, 180 N Michigan Ave, Chicago, and (for mail), 314 Cuttriss Place, Park Ridge, Ill

HEDGES, H. Berkley (*M* 1919), Mgr of Industrial Sales, J J Nesbitt, Inc, Holmesburg, Philadelphia, Pa, and (for mail), 1021 Park Lane, Plainfield, N J

HEDLEY, Park S (*M* 1923), Park S Hedley Co, 361 Delaware Ave, Buffalo, N Y

HEEBNER, Walter M. (*M* 1922), Sales Engr, Warren Webster & Co, 95 Madison Ave, New York, N. Y, and (for mail), 282 Highwood Ave, Teaneck, N J

HEIBEL, Walter E. (*M* 1917), Dist Mgr (for mail), Aerofin Corp, 11 West 42nd St, New York, N Y., and Old Greenwich, Conn

HEILMAN, Russell H.* (*M* 1923), Senior Industrial Fellow (for mail), Mellon Institute of Industrial Research, Thackary and Ohara Sts, and 5637 Wilkins Ave, Pittsburgh, Pa

HELBURN, I. B. (*M* 1929; *J* 1927), Junior Assoc (for mail), Wyman Engineering, 610 Chamber of Commerce Bldg, and 700 Chalfonte Place, Apt. 17, Cincinnati, Ohio

HELLSTROM, John (*A* 1929), Vice-Pres (for mail), American Air Filter Co, 215 Central Ave, Louisville and Anchorage, Ky

HELMRICH, G. Bernard (*M* 1936), Detroit Edison Co, 2000 Second Ave, Room 750, Detroit, and (for mail), 26590 Dundee Rd, Huntington Woods, Royal Oak, Mich

HELSTROM, Herman G. (*M* 1928), Salesman (for mail), Wm Bros Boiler & Mfg. Co, and 4608 Arden Ave, Minneapolis, Minn

HENDRICKSON, Harold M. (*M* 1933), Engr (for mail), York Ice Machinery Corp, 3051 Santa Fe Ave, Los Angeles, and 7022 Middleton St, Huntington Park, Calif

HENION, Hudson D. (*A* 1923), Sales Mgr (for mail), C A Dunham Co, Ltd, 1523 Davenport Rd, and 45 Ridge Drive, Toronto, Ont, Canada

HENRY, Alexander S., Jr. (*M* 1930), 300 Central Park W, New York, N Y

HENSZEY, William P. (*J* 1935), Pres, W P Henszey Co, Lemont, Pa

HERENDEEN, Frederick W. (*M* 1920), Secy (for mail), The Institute of Boiler & Radiator Mfrs, 29 Seneca St, and 815 S Main St, Geneva, N Y

HERING, Alfred (*M* 1935), Pres, Hering Heating Co, Inc, 1672 Second Ave, and (for mail), 1830 Tenbroeck Ave, New York, N Y

HERKIMER, Herbert (*M* 1934), Director (for mail), Herkimer Institute, 1819 Broadway, and 25 Central Park W, New York, N Y

HERLIHY, Jeremiah J. (*Life Member, M* 1914), 3634 N Keeler Ave, Chicago, Ill

HERMAN, Neil B (*S* 1936), Installation and Service Dept, Minneapolis-Honeywell Regulator Co, 2753 Fourth Ave S, and (for mail), 4217 Garfield Ave S, Minneapolis, Minn

HERRICK, Leo (*M* 1935), Mgr (for mail), Crane Co, 521 S Ninth, and 323 Greenwood Ave, Fort Smith, Ark

HERRING, Edgar (*M* 1919), Chairman and Governing Director (for mail), J Jeffreys & Co, Ltd, Barrons Place, Waterloo Rd, London S E, and "Kenia," Keswick Rd, Putney, London S W, England

HERSKE, Arthur R. (*M* 1926), Vice-Pres-Gen Mgr Sales (for mail), American Radiator Co, 40 West 40th St, New York, and 25 Vernon Parkway, Mt Vernon, N Y

HERTY, Frank B. (*M* 1933), Retail Sales Supervisor (for mail), Brooklyn Union Gas Co, 176 Remsen St, and 50 East 18th St, Brooklyn, N Y

HERTZLER, John R * (*M* 1936, *J* 1928), Mgr Air Cond Div (for mail), York Ice Machinery Corp, Roosevelt Ave, and 863 S George St, York, Pa

HESS, Arthur J (*M* 1937), Engr, English & Lauer, Inc, 309 West 12th St, and (for mail), 2616 West 70th St, Los Angeles, Calif

HESS, David K. (*J* 1936, *S* 1932), Hess Warming & Ventilating Co, 1211-27 S Western Ave, and (for mail), 5824 Harper Ave, Chicago, Ill

HESSLER, Lester W (*M* 1936), Branch Mgr, Trane Co, 125 E Wells St., and (for mail), 6034 N Bayridge, Milwaukee, Wis.

HESTER, Thomas J. (*M* 1919), Vice-Pres-Treas (for mail), Hester Bradley Co, 2835 Washington Blvd, St. Louis, and 67 Aberdeen Place, Clayton, Mo.

HEWETT, John B. (*A* 1935), Sales Engrg Mgr, Quinby Air Conditioning Corp., 618 E Main St, Rochester, N Y

HEXAMER, Harry D. (*M* 1931), Sales Engr (for mail), Excelso Products Corp, 65 Clyde Ave, and 163 E Delavan Ave, Buffalo, N Y

HEYDON, Charles G. (*A* 1923), Mgr Sales of Western Div, Wright-Austin Co, 315 W Woodbridge St, and (for mail), 2681 Nebraska, Detroit, Mich

HIBBS, Frank C. (*M* 1917), Salesman, H B Smith Co, 2209 Chestnut St, and (for mail), 846 North 65th St, Philadelphia, Pa

HICKEY, Daniel W. (*A* 1931), Pres, D W Hickey & Co, Inc (for mail), 1631 University Ave, and 1874 Highland Parkway, St Paul, Minn

HICKS, H. Kimble (*J* 1936, *S* 1935), 215 East 15th St, New York, N Y.

HIERS, Charles R. (*M* 1929; *J* 1927), Sales Engr, Minneapolis-Honeywell Regulator Co, 801 Second Ave, New York, and (for mail), 4518-258th St, Great Neck, N Y.

HILDEBRANDT, Henry A. (*M* 1918), Elliott Equipment Co, 708 Sixth Ave S, Minneapolis, Minn

HILDRETH, Egbert S (*A* 1936), Air Cond Engr (for mail), Indianapolis Power & Light Co, 17 N Meridian St, and 5734 E New York St, Indianapolis, Ind

HILDRETH, Lane W (*M* 1935), Dist Mgr (for mail), Anthracite Institute, 19 Rector St, New York, N Y, and 221 Wheatsheaf Lane, Abington, Pa

HILL, Charles F. (*J* 1936), Engr (for mail), Carrier-Brunswick International, 850 Frelinghuysen Ave, Newark, and 803 Colonia Rd, Elizabeth, N J

HILL, Dr. E Vernon[1] (*M* 1911, *A* 1912), (*Presidential Member*), (Pres, 1920, 1st Vice-Pres, 1919, 2nd Vice-Pres, 1918, Council, 1915-1921), Owner (for mail), 179 W Washington St, and 6826 Newell, Chicago, Ill

HILL, Fred M. (*M* 1930), 225 East Ave 39, Los Angeles, Calif

HILL, Harold H. (*M* 1935), Sales Engr, Delco-Frigidaire Conditioning Corp, Room 3-212, General Motors Bldg, and (for mail), 16537 Indiana, Detroit, Mich

HILLIARD, Charles E (*M* 1932, *J* 1927), Htg-Vtg Engr (for mail), E C Hilliard Co, 27 B St, South Boston, and 1301 Washington St, South Braintree, Mass

HILLS, Arthur H. (*M* 1924), Mgr Sarco Canada, Ltd, 725-6 Federal Bldg, 85 Richmond St W, Toronto, Ont, Canada

HINCKLEY, Harlan B (*A* 1934), Engr-Custodian, Chicago Board of Education, 8510 S Green St, and (for mail), 6933 Princeton Ave, Chicago, Ill

HINKLE, Edwin C. (*Life Member, M* 1911), 170 N Franklin St, Hempstead, N Y

HINRICHSEN, Arthur F (*M* 1928), Pres (for mail), A F Hinrichsen, Inc, 50 Church St, New York, N Y, and Mountain Lakes, N J

HINTZ, Harvey P (*J* 1936, *S* 1935), 211 E Armory Ave, Champaign, Ill

HIRSCHMAN, William F. (*M* 1929), Pres and Chief Engr, W F Hirschman Co, Inc, 220 Delaware Ave, and (for mail), 165 Le Brun Circle, Buffalo, N Y

HITCHCOCK, Paul C. (*M* 1931), Vice-Pres-Treas, Burlingame, Hitchcock & Estabrook, Inc, 521 Sexton Bldg, and 4939 Girard Ave S, Minneapolis, Minn

HITT, John C. (*A* 1936), Branch Mgr (for mail), Holland Furnace Co, 34-17th St, and 301 Valley View Ave, Wheeling, W Va.

HOBBS, J. Clarence (*M* 1920), Gen Mgr, and Mfg Operations (for mail), Diamond Alkali Co, and 60 Wood St, Painesville, Ohio

HOBBS, William S. (*A* 1936), Owner and Mgr, 511 Yale Ave, Swarthmore, Pa.

HOCHSTEIN, George E. (*A* 1935), Mgr Oil Burner Div, Heil Co, 810 North 26th St, Philadelphia, Pa

HOCKENSMITH, Francis E. (*M* 1936), Chief Engr (for mail), Lennox Furnace Co, Inc, 400 N Midler Ave, and 124 Ludington St., Syracuse, N Y.

HODEAUX, Walter L. (*M* 1931), Owner (for mail), W L Hodeaux Plumbing & Heating Co., 215-17 N Flagler Drive, and 310 Tenth St, West Palm Beach, Fla.

HODGDON, Harry A. (*M* 1919), 153 Norfolk St, Wollaston, Mass

HODGE, William B. (*M* 1934), Vice-Pres, Parks-Cramer Co., and (for mail), P. O Box 1234, Charlotte, N C

HOEHL, Edward R. (*J* 1935), Engr (for mail), Carrier Corp, 850 Frelinghuysen Ave, Newark, and 645 Jefferson St, West New York, N J

HOFFMAN, Charles S. (*M* 1924), Vice-Pres (for mail), Baker, Smith & Co, Inc, 576 Greenwich St, and 108 East 38th St, New York, N Y

HOFFMAN, James D † (*M* 1903), (*Presidential Member*), (Pres, 1910, 1st Vice-Pres, 1908, Board of Governors, 1911-1912), Prof of Practical Mechanics, Head of Dept, Director of Practical Mech Lab (for mail), Purdue University, and 323 University St, West Lafayette, Ind

HOGAN, Edward L † (*M* 1911), Consulting Engr (for mail), American Blower Co, 6000 Russell St, and 700 Seward Ave, Detroit, Mich

HOGUE, William M (*A* 1935), Sales Engr (for mail), U S Electrical Motors, Inc, 200 E Slauson Ave, and 602 N Manhattan Place, Los Angeles, Calif

HOLBROOK, Frank M † (*M* 1923), Engr, Apt J-1-5, 10 Lexington St, Newark, N J

HOLLISTER, E. Wallace (*M* 1936, *J* 1931), Owner, Hollister's, 31 Ridge St, Glens Falls, and (for mail), 88 Oak St, Hudson Falls, N Y

HOLLISTER, Norman A (*M* 1933), 7101 Colonial Rd, Brooklyn, N Y

HOLMES, Arthur D. (*M* 1935), Vice-Pres (for mail), Plumbers Supply Co, 323 W First, and 1848 East 18th St, Tulsa, Okla

HOLMES, Paul B (*A* 1936), Branch Mgr (for mail), National Radiator Corp, 600 W St, N E, and 1525 Fessenden St N.W, Washington, D C

HOLMES, Richard E (*J* 1931), Air Cond Design Engr, Westinghouse Electric & Mfg Co, 653 Page Blvd, and (for mail), 11 Bushwick St, Springfield, Mass

HOLT, James (*M* 1933), Assoc Prof of Mech Engrg (for mail), Massachusetts Institute of Technology, Charles River Rd, Cambridge, and 1062 Massachusetts Ave, Lexington, Mass

HOLTON, John H (*M* 1927), Mgr, Construction and Service Dept, Eastern Contract Dept (for mail), Carrier Corp, 850 Frelinghuysen Ave, Newark, and 5 Mountain Ave, Maplewood, N J

HOOD, O. P. (*Honorary Member* 1929), 1831 Irving St N W, Washington, D C

HOPKINS, Frank L (*S* 1936), Inspector, Minneapolis-Honeywell Regulator Co, 2711 Fourth Ave S, Minneapolis, and (for mail), 2819 Colorado Ave, St Louis Park, Minn

HOPP, Herbert K (*S* 1935), 530 McLean Ave, Yonkers, N Y

HOPPE, Albert A. (*M* 1935), Design and Application Engr, Carrier Corp, and (for mail), 524 Northwest 17th St, Oklahoma City, Okla

HOPPER, Garnet H. (*M* 1923), Engr, Taylor Forbes, Ltd, 1088 King St W, and (for mail), 19 Brummell Ave, Toronto, Ont., Canada

HOPSON, William T. (*Life Member, M* 1915), The Hopson & Chapin Mfg Co, New London, Conn

HORNUNG, John C. (*M* 1914), Engr (for mail), Central Heat Appliances, 343 S Dearborn St, Chicago, and 851 Bluff St, Glencoe, Ill

HORTON, Homer F. (*M* 1925), Sales Repr (for mail), National Regulator Co, 2301 Knox Ave, Chicago, and 1301 Judson Ave, Evanston, Ill

HOSHALL, Robert H (*M* 1930), Associate (for mail), Thos. H Allen, Consulting Engr, 65 McCall St., and 789 N Evergreen St, Memphis, Tenn

HOSKING, Homer L. (*M* 1930), Sales Mgr (for mail), Pacific Steel Boiler Corp, 101 Park Ave, New York, and 5 Church Lane, Scarsdale, N. Y.

HOSTERMAN, Charles O. (*M* 1924), Supt, McMurrer Co., 303 Congress St, Boston, and (for mail), 25 Bateswell Rd, Dorchester, Mass

HOTCHKISS, Charles H. B (*M* 1927), Editor, Heating and Ventilating, 148 Lafayette St, New York, N. Y.

HOUGHTEN, Ferry C.* (*M* 1921), Director (for mail), Research Lab, A S H.V E, U S Bureau of Mines, 4800 Forbes St., and 1136 Murray Hill Ave., Pittsburgh, Pa

HOULIS, Louis D. (*M* 1935), Chief Engr, Master Baker Ovens, and (for mail), 655 Pedretti Rd, West Price Hill, Cincinnati, Ohio.

HOULISTON, G. Baillie (*A* 1928), Secy (for mail), W. C. Green Co, 704 Race St, Cincinnati, Ohio, and 33 Tremont Ave, Ft Thomas, Ky

HOWATT, John‡ (*M* 1915), (*Presidential Member*), (Pres, 1935, 1st Vice-Pres, 1934, 2nd Vice-Pres, 1933, Council, 1927-1936), Chief Engr (for mail), Board of Education, 228 N LaSalle St, and 1940 East End Ave, Chicago, Ill

HOWE, Willis W. (*M* 1936, *A* 1917), Div Sales Engr, Pacific Gas & Electric Co, and (for mail), 108 Central Ave, Sausalito, Calif

HOWLETT, Ira G (*M* 1935, *S* 1934), Consulting Engr (for mail), I G Howlett Co, 120 E Main St, and 2132 N Fonshill Ave, Oklahoma City, Okla

HOWELL, Frank B (*M* 1920), Tech Advisor (for mail), American Radiator Co, 40 West 40th St, and 15 Central Park W, New York, N Y

HOWELL, Lloyd (*M* 1915), Engr, Industrial Dept, Peoples Gas Light & Coke Co, 122 S. Michigan Ave, and (for mail), 7927 Langley Ave, Chicago, Ill

HOYT, Charles W. (*A* 1931), Pres-Treas (for mail), Wolverine Heating & Ventilating Equipment Co, 31 Main St, Cambridge, and 45 Thaxter Rd, Newtonville, Mass

HOYT, Leroy W (*M* 1930), N Stamford Ave, Stamford, Conn

HUBBARD, George W * (*M* 1911), Chief Mech Engr (for mail), Graham, Anderson, Probst & White, 1417 Railway Exchange, Chicago, and 710 Bonne Brae, River Forest, Ill.

HUCH, A J. (*M* 1919), Secy-Treas (for mail), Central Supply Co, 312 S Third St, and 4037 Harriet Ave, Minneapolis, Minn

HUCKER, Joseph H. (*M* 1921), Partner, Hucker-Prybil Co, 1700 Walnut St, Philadelphia, and (for mail), 715 Stanbridge St, Norristown, Pa

HUDEPOHL, Louis F (*M* 1936), Pres (for mail), T J Conner, Inc, 3290 Spring Grove Ave, and 4305 Haight Ave, Cincinnati, Ohio

HUDSON, Robert A. (*M* 1934), Consulting Engr (for mail), Hunter & Hudson, Room 710, 41 Sutter St, and 2850 Union St, San Francisco, Calif

HUETTNER, Henry F. (*S* 1934), 124 Jerusalem Ave, Hicksville, L I, N Y, and (for mail), Box 388, Carnegie Tech, Pittsburgh, Pa

HUFF, James M (*M* 1936), Mgr Air Cond Dept (for mail), Silkensen & Co, Inc, 401-23rd St, and 1720 Ave E, Galveston, Texas

HUGHES, Charles E. (*J* 1936, *S* 1934), 140 Beacon Ave, New Haven, Conn

HUGHES, Lewis K (*J* 1936), Gen Mgr, Howard Furnace Co, 881 Yonge St, and (for mail), 43 Rivercourt Blvd, Toronto, Canada

HUGHES, William U. (*M* 1936), Managing Dir (for mail), Lewis-Brown Co., Ltd, 1409 Crescent St, and 4330 Wilson Ave, Montreal, P Q, Canada

HUGHEY, Thomas M. (*A* 1935), Sales Engr (for mail), Westerlin & Campbell Co, 906 N Fourth St, and 2350 North 58th St, Milwaukee, Wis

HUGONIOT, Victor E (*M* 1935), Zone Engrg Mgr (for mail), Airtemp Sales Corp, 3717 Washington Ave, and 1347 Kingsland Ave, St Louis, Mo

HULL, Harry B. (*M* 1931), Mgr Research Engr (for mail), Frigidaire, Div of General Motors Corp, and 1430 Glendale Ave, Dayton, Ohio

HUMPHREY, Dwight E.* (*M* 1921), Htg-Vtg Engr, Goodyear Tire & Rubber Co, 1144 E Market St, Akron, and (for mail), 2499 Sixth St, Cuyahoga Falls, Ohio.

HUMPHREYS, Clark M. (*M* 1931), Asst Prof of Mech Engrg (for mail), Carnegie Institute of Technology, Schenley Park, and 1934 Remington Drive, Pittsburgh, Pa

HUNGER, Robert F. (*M* 1927), Assoc Dist Mgr (for mail), Buffalo Forge Co, 220 South 16th St, and 4618 Chester Ave, Philadelphia, Pa

HUNGERFORD, Leo (*M* 1930), Mgr Air Cond Dept, (for mail), Delco-Frigidaire Corp, 1057 N Le Brea Ave, and 2360 Laurel Canyon Blvd, Hollywood, Calif

HUNT, MacDonald (*A* 1936), Mfrs Agent (for mail), McDonnell Miller Co, 12 W Madison St, and Roland Park Apts, Baltimore, Md

HUNT, Noel P. (*M* 1934), Managing Dir (for mail), Carrier Australasia, Ltd, 41-49 Forbes St, and 52 Lang Rd, Centennial Park, Sydney, N S W., Australia

HUNTER, Louis N (*M* 1936), Mgr of Research (for mail), National Radiator Corp, 221 Central Ave, and 403 Wayne St, Johnstown, Pa

HUNZIKER, Chester E (*A* 1934), Branch Mgr (for mail), American Blower Corp, 331 State St, Schenectady, and 422 Reynolds St, Scotia, N Y

HUSKY, S T (*J* 1936, *S* 1934), Engr, Zenith Gas System, Box 397, Alva, Okla

HUST, Carl E (*M* 1932), Supv Htg Engr (for mail), Cincinnati Gas & Electric Co, Fourth and Main Sts, and Hillcrest Apts, 15 Mason St, Cincinnati, Ohio

HUSTOEL, Arnold M (*A* 1930), 2414 N Kedzie Blvd, and (for mail), 2623 N Ballou St, Chicago, Ill

HUTCHINS, William H (*M* 1934), Chief Engr, Delco Appliance Div, General Motors Corp, and (for mail), 88 Magee St, Rochester, N Y

HUTZEL, Hugo F. (*M* 1918), Air Cond Applications Dept, Kelvinator Corp, and (for mail), 2635 Woodstock Drive, Detroit, Mich

HVOSLEF, Frederick W (*M* 1931, *A* 1921), Htg Research Engr (for mail), Kohler Co, and 523 Audubon Rd, Kohler, Wis

HYDE, Elmer H (*A* 1937), Tech Repr, Koppers Co, Forbes St. and Meyran Ave, Flannery Bldg, and (for mail), 2119 Los Angeles Ave, Pittsburgh, Pa

HYDE, Lawrence L (*J* 1934), Gen Mgr, M J O'Neil, and (for mail), 54 S Cretin St, St Paul, Minn

HYMAN, Wallace M (*M* 1920), Pres (for mail), Reis & O'Donovan, Inc, 12 West 21st St, and 23 West 73rd St, New York, N Y

HYNES, Lee P* (*M* 1919), Pres and Chief Engr (for mail), Hynes Electric Heating Co, 240 Cherry St, Philadelphia, Pa, and 127 West End Ave, Haddonfield, N J

I

ICKERINGILL, John C. (*M* 1923), Engr, Spencer Heater Co, 2504 N Broad St, and (for mail), 235 Rector St, Philadelphia, Pa

ILLIG, Walter R. (*M* 1935, *A* 1927), Owner, 1 Cushing St, and (for mail), 210 South St, Fitchburg, Mass

INGALLS, Frederick D. B. (*M* 1906), Consulting Engr, 1 Hopkins St, Reading, Mass

INGELS, Margaret* (*M* 1923, *J* 1918), Mech Engr (for mail), Carrier Corp, 850 Frelinghuysen Ave, Newark, and Hotel East Orange, East Orange, N J

ISETT, William M. (*A* 1936), Pres (for mail), C. B Isett & Son, Inc, 3035-37-39 N Rockwell St, and 4236 N Drake Ave, Chicago, Ill

d'ISSERTELLE, Henry G.* (*Life Member* 1937; *M* 1913, *A* 1912), Consulting Engr, 31 Park Terrace W, Apt A-8, New York, N Y.

IVERSON, Henry R (*M* 1936, *A* 1936), Sales Engr (for mail), Trane Co, 726 Jackson Place N W, and 1601 Argonne Place N W, Washington, D C

J

JACKSON, Charles H. (*M* 1923), Vice-Pres (for mail), Blower Application Co., 918 N Fourth St, and 2706 N Farwell Ave, Milwaukee, Wis

JACKSON, Marshall S. (*M* 1919), Repr (for mail), Powers Regulator Co, 250 Delaware Ave., and 108 Larchmont Rd, Buffalo, N. Y.

JACOBI, Bruce A (*S* 1936), 1731 East 17th St, Brooklyn, N Y, and (for mail), Box 276, Carnegie Institute of Technology, Pittsburgh, Pa

JACOBUS, Dr. David S (*Life Member*, *M* 1916), Advisory Engr (for mail), The Babcock & Wilcox Co, 85 Liberty St, New York, N Y., and 93 Harrison Ave, Montclair, N J

JALONACK, Irwin G (*A* 1933, *S* 1930), Engrg Mgr (for mail), c/o A L Hart, 82 Railroad Ave, Patchogue, and Beaver Dam Rd, Brookhaven, N Y

JAMES, Hamilton R (*M* 1931), Service Equip Engr, United Engineers & Constructors, Inc, 1401 Arch St, Philadelphia, and (for mail), 55 W Drexel Ave, Lansdowne, Pa

JAMES, John W* (*J* 1933), Tech Asst, American Society of Heating & Ventilating Engineers, 51 Madison Ave, New York, N Y

JAMES, Richard E (*M* 1936), Mgr Htg Dept, Harry Cooper Supply Co, and (for mail), 597 E Elm St, Springfield, Mo

JANET Harry L (*M* 1920), Mech Engr, Buensod-Stacey Air Conditioning, Inc, 60 East 42nd St, New York, and (for mail), 688 Decatur St, Brooklyn, N Y

JARCHO, Martin D (*J* 1936), Vice-Pres (for mail), Jarcho Bros, 215 East 37th St, New York, and 941 Washington Ave, Brooklyn, N Y

JARDINE, Douglas C (*M* 1929, *A* 1920), Pres (for mail), Jardine & Knight Plumbing & Heating Co, 312 N Custer Ave, and 1512 E Platte Ave, Colorado Springs, Colo.

JARRATT, Paul R. (*A* 1931), 117 Fifth Ave N, Nashville, Tenn

JEFFREY, Thomas G (*A* 1935), Mgr (for mail), Bastian-Morley, Ltd, 2302 Dundas St W, and 178 Windemere Ave, Toronto, Canada

JELINEK, Frank R (*J* 1937), Sales Engr, Automatic Temp Control (for mail), Johnson Service Co, 127 Brainard St, and 2051 W Grand Blvd, Detroit, Mich

JENNEY, Hugh B (*A* 1933), Gen Sales Mgr (for mail), Dominion Radiator & Boiler Co, Ltd, Royce and Lansdowne Aves, and 15 Quebec Ave, Toronto, Ont, Canada

JENNINGS, Hal K (*M* 1937), Sales Engr., Avery Engineering Co, 2311 Carnegie, Cleveland, and (for mail), 814 Chestnut Blvd, Cuyahoga Falls, Ohio

JENNINGS, Irving C. (*M* 1921), Pres (for mail), Nash Engineering Co, and 138 Flax Hill Rd, South Norwalk, Conn

JENNINGS, Stanley A. (*M* 1935), Chief Draftsman, Trane Co of Canada, Ltd, 430 King St W., and (for mail), 80 Glen Manor Drive, Toronto, Ont, Canada

JENNINGS, Warren G. (*A* 1930), Resident Vice-Pres, Minneapolis-Honeywell Regulator Co, 285 Columbus Ave., Boston, Mass

JENNINS, Henry H (*Life Member*, *M* 1901), 15 Grange View, Chapeltown Rd, Leeds, England

JENSON, Jean S (*M* 1912), Consulting Engr (for mail), 431 S Dearborn St, and 1634 West 106th St, Chicago, Ill

JEPERTINGER, Richard C. (*A* 1931), Vice-Pres, Gen Mgr (for mail), Syncromatic Air Conditioning Corp, 1317 N Third St, and 1628 W Vienna St, Milwaukee, Wis

JEX, John, Jr. (*A* 1936), Sales Engr (for mail), Mercoid Corp, 3137 N Broad St, Philadelphia, Pa, and Earleville, Cecil County, Md

JIMENEZ, Joaquin G. (*M* 1935), Engr. Director (for mail), Ave. Edwardo Dato, 34, Madrid, Spain

JOHN, Victor P. (*M* 1931), Mgr, Buffalo Branch, American Blower Corp, 822 White Bldg, Buffalo, and (for mail), 136 Berryman Drive, Snyder, N Y.

JOHNS, Harold Byron* (*M* 1928, *J* 1927), (for mail), Peoples Gas Light & Coke Co, 122 S Michigan Ave, Chicago, and 543 N Elmwood Ave, Oak Park, Ill

JOHNSON, Allen J.* (*M* 1935), Director (for mail), Anthracite Institute Laboratory, Primos, Del Co, and 344 Congress Ave, Lansdowne, Pa

JOHNSON, Carl W. (*M* 1912), Pres (for mail), C W Johnson, Inc, 211 N Desplaines St, and 1809 Morse Ave, Chicago, Ill.

JOHNSON, Clarence W (*M* 1933; *A* 1933, *J* 1931), Dist Mgr (for mail), Canadian Sirocco Co, Ltd, 630 Dorchester St W, and 333 Dresden Ave, Mt Royal, Montreal, Que, Canada
JOHNSON, Edward B. (*M* 1919), Sales Engr, Staten Island Supply Co, and (for mail), 154 Wardwell Ave, West New Brighton, S I, N Y
JOHNSON, Helge S (*A* 1933, *J* 1927), Dist Mgr (for mail), Buffalo Forge Co (112 State St), 611 Standard Bldg, and 20 Fleetwood Ave, Albany, N Y
JOHNSON, Leslie O (*J* 1930), Engr, C A. Dunham Co, 30 Rockefeller Plaza, New York, N Y, and (for mail), 42 W. Holly St, Cranford, N J
JOHNSON, Louis H (*M* 1931), 918 LaSalle Ave, Minneapolis, Minn
JOHNSON, Walter A. (*J* 1936; *S* 1935), Asst Instructor in Mech Engrg (for mail), Columbia University, Mech Engrg. Dept, and 1138 John Jay Hall, New York, N Y
JOHNSON, Wayne G. (*S* 1936), Asst Research Engr, Herman Nelson Corp, 1824 Third Ave, and (for mail), 810-20th Ave, Moline, Ill
JOHNSTON, Hugo D. (*A* 1934), Engr and Contr, Box 282, Wellington, Ont, Canada
JOHNSTON, J. Ambler (*M* 1912), Partner, Carneal, Johnston & Wright, 806 Electric Bldg, and 2016 Hanover Ave, Richmond, Va
JOHNSTON, Robert E (*M* 1929; *A* 1926), Managing Dir (for mail), R E Johnston Co., Ltd, 1070 Homer St, and 3342 West 33rd Ave, Vancouver, B C, Canada
JOHNSTON, William H. (*M* 1924), 306 East 26th St, New York, N Y
JONES, Alfred (*M* 1928), Chief Consulting Engr. (for mail), Armstrong Cork Co, P O Box 510, and 402 President Ave, Lancaster, Pa
JONES, Alfred L (*M* 1926), Plbg and Htg Contr (for mail), 431 Greenwich Ave, Greenwich, and Breezemont Ave, Riverside, Conn
JONES, Allan T. (*J* 1935), Mech Engr (for mail), S A. Armstrong, Ltd, 720 Bathurst St, and 325 Kingswood Rd, Toronto, Ont, Canada
JONES, Andrew P. (*J* 1936, *S* 1935), Elec and Mech Engr, 504 E Fifth, Hereford, Texas
JONES, Bernard G. (*M* 1928), Mgr (for mail), Acme Fan & Blower Co, Ltd, 868 Arlington St, and 512 Raglan Rd., Winnipeg, Man, Canada
JONES, Charles R. (*A* 1928), Jones Supply Co, Inc, Siloam Springs, Ark
JONES, David J. (*M* 1936), Control Engr, Vapor Car Heating Co, Inc, Railway Exchange Bldg, Chicago, and (for mail), 391 Poplar Ave, Elmhurst, Ill
JONES, Edwin (*M* 1933, *J* 1924), Engr and Estimator (for mail), Watt Plumbing, Heating & Supply Co, 608 S Cincinnati, and 1436 East 17th Place, Tulsa, Okla.
JONES, Edwin A. (*M* 1919), Chief Engr (for mail), L J Mueller Furance Co, 2001 W Oklahoma, and 4065 Prospect, Milwaukee, Wis
JONES, Edwin F. (*M* 1923), Consulting Engr, 922 New York Bldg, and (for mail), 220 Montrose Place, St Paul, Minn.
JONES, Harold L. (*M* 1920), Supt (for mail), W W Farrier Co, 44 Montgomery St., Jersey City, and 11 Cambridge Rd, Glen Ridge, N J
JONES, Norman R. (*A* 1935), Sales Mgr, Orr & Sembower, Inc, Reading, and (for mail), 109 Rutledge Ave, Rutledge, Pa
JONES, Sprague (*M* 1936), Pres.-Mgr. (for mail), Sprague Jones, Inc, 1116 Madison Ave, and 3709 S Lockwood Ave., Toledo, Ohio
JONES, William T. (*M* 1915), (*Presidential Member*), (Pres., 1933; 1st Vice-Pres, 1932, 2nd Vice-Pres., 1931; Council, 1925-1933), Treas., Barnes & Jones, 128 Brookside Ave, Jamaica Plain, and (for mail), 16 Harvard St, Newtonville, Mass
JOPSON, John M. (*M* 1936), Engr, W M Anderson Co, 600 Schuylkill Ave, Philadelphia, and (for mail), 3455 W Penn St, East Falls, Philadelphia, Pa.
JORDAN, Lebert E (*A* 1934), Installation Supv, 122 Arthur Ave S E, Minneapolis, Minn
JORDAN, Richard C (*J* 1935, *S* 1933), Instructor, University of Tulsa, and (for mail), 1004 S College, Tulsa, Okla
JORDAN, W. D. (*M* 1935), Mgr, Air Cond Div (for mail), Savage Arms Corp, 100 East 42nd St, New York, and 132 Overlook Rd, New Rochelle, N Y
JOSEPHSON, Simon (*J* 1936), Engr (for mail), Astor Plumbing & Heating Corp, 1134 Bedford Ave, and 525 Linden Blvd, Brooklyn, N Y.
JOYCE, Harry B. (*M* 1922), Consulting Engr (for mail), 810 Commerce Bldg, and 501 Liberty St, Erie, Pa
JUNG, John S (*M* 1930, *A* 1923), Htg, Piping and Air Cond Contr (for mail), 2409 W Greenfield Ave, and 1516 S Layton Blvd, Milwaukee, Wis
JUNKER, William H. (*M* 1936), Chief Mech Engr, Thos Emery's Sons, Inc, 2109 Carew Tower, and (for mail), 6068 Dryden Ave, Cincinnati, Ohio
JUTTNER, Otto J. (*M* 1915), Pres (for mail), Juttner Heating Co, 814 N Milwaukee St, and 910 E Wis. Ave, Milwaukee, Wis.

K

KACZENSKI, Chester (*J* 1933), 135 Madison Ave, Elizabeth, N J
KAGEY, I. B.* (*J* 1929), Sales Engr (for mail), Carrier Corp, 404 Bona Allen Bldg, and Atlanta Athletic Club, Atlanta, Ga
KAIN, Edward M. (*S* 1936), Draftsman, Babcock & Wilcox Co, Stirling Ave, Barberton, and (for mail), 3656 E 146th St, Cleveland, Ohio
KAISER, Fred (*M* 1935), Branch Mgr (for mail), Minneapolis-Honeywell Regulator Co, 45 Allen St, and 481 Stann Ave, Buffalo, N Y
KAJUK, Andrew E. (*M* 1936), Engr, Resettlement Administration, U. S Govt, 2022 Massachusetts Ave. N W, and (for mail), 5304 Fourth St N W, Washington, D C
KALINSKY, Alex G. (*J* 1936, *S* 1934), Htg Engr, Fox Furnace Co, Elyria, and (for mail), 13706 Durkee Ave, Cleveland, Ohio
KAMMAN, Arnold R. (*A* 1925, *J* 1921), (for mail), 493 Franklin St, Buffalo, and R F D No 3, Hamburg, N Y
KAMPISH, Nick S. (*J* 1935, *S* 1934), Air Cond Engr, Air Temp New York Sales Corp, Chrysler Bldg, New York, N Y, and (for mail), 214 E Lincoln Ave, Roselle Park, N J.
KAPPEL, George W. A. (*M* 1921), Pres and Treas (for mail), Camden Heating Co, Wilson Blvd and Waldorf Ave, Camden, and 347 W Kings Highway, Haddonfield, N J
KARAKASH, Todori (*J* 1936), Engr, Carrier of Turkey (for mail), Posta Kutusu 468, and Engin Apt, Ferus-Aga, Galatasaray, Istanbul, Turkey
KARCHMER, Jacob H. (*A* 1936), Mgr (for mail), Karchmer Co, 600-14 N Jefferson Ave, and 1316 Roanoke Ave., Springfield, Mo.
KARGES, Albert (*A* 1935), Vice-Pres. and Managing Dir, James Stewart Mfg Co., Ltd, Tecumseh St, and (for mail), 37 Perry St, Woodstock, Ont., Canada
KARLSON, Alfred F. (*M* 1918), Chief Engr. (for mail), Parks-Cramer Co, 970 Main St, Fitchburg, and 186 Prospect St, North Leominster, Mass
KARLSTEEN, Gustav H. (*M* 1935), Plant Engr, Dunlop Tire & Rubber Corp, Buffalo, and (for mail), Box 55, Route 1, Tonawanda, N Y.
KARTORIE, V. T. (*J* 1935, *S* 1933), Engr, York Ice Machinery Corp., 2700 Washington Ave, and (for mail), 2982 East 102nd St, Cleveland, Ohio
KASTNER, George C. (*J* 1935, *S* 1933), 654 East 226th St, New York, N Y
KAUFMAN, Charles W. (*J* 1935), Engr (for mail), Carrier Corp, 180 N Michigan Ave, and 4932 Blackstone Ave, Chicago, Ill

KAWASE, Sumio (*M* 1936), Chief Htg Engr, Eizen Juhin Kyoku—Manchoukuo, and (for mail), 614 Suchikodo, Hsinking, Manchoukuo

KEEFE, Edmund T. (*M* 1931), Pres (for mail), Underground Steam Construction Co, 75 Pitts St, Boston, and 185 Commonwealth Ave, Chestnut Hill, Mass

KEENEY, Frank P. (*A* 1915), Pres, Keeney Publishing Co, 6 N Michigan Ave, and South Shore Country Club, Chicago, Ill

KEHM, Horace S. (*M* 1928), Pres, Kehm Bros Co & Stevens-Root Co (for mail), 51 E Grand Ave, and Belmont Hotel, Chicago, Ill

KELBLE, Frank R. (*M* 1928), Vice-Pres and Mgr (for mail), Hoffman-Wolfe Co of Philadelphia, 11 W Rittenhouse St, Philadelphia, and 115 Rosyln Ave, Glenside, Pa

KELL, Waldo R (*A* 1934), Sales Engr (for mail), The Marley Co, 1915 Walnut St, and 1107 West 49th St, Kansas City, Mo

KELLEY, James J. (*A* 1924), Vice-Pres and Gen Mgr (for mail), Arthur H Ballard, Inc, 535 Commonwealth Ave, Boston, and 142 Governors Ave, Medford, Mass

KELLNER, Day C (*S* 1933), Cuba City, Wis

KELLOGG, Alfred S (*Life Member, M* 1916), (Council, 1920-1921, 1923-1924), Consulting Engr (for mail), 585 Boylston St, Boston, and 6 Hawthorne St, Belmont, Mass

KELLY, Charles J. (*M* 1931), Repr, James P Marsh Corp, 551 Fifth Ave, New York, N Y, and (for mail), 96 Duncan Ave, Jersey City, N J

KELLY, John G. (*A* 1919), 374 Park Ave, Yonkers, N Y

KELLY, Wilbur C (*M* 1935), Field Engr (for mail), Iron Fireman Mfg Co, 602 King St, W and 58 Elmsthorpe Ave, Toronto, Ont, Canada

KENNEDY, Maron (*A* 1936, *J* 1930), Sales Engr, York Ice Machinery Corp, P O Box 786, Sacremento, Calif

KENNEDY, Owen A (*S* 1933), 112 Dixie Highway, South Fort Mitchell, Ky

KENNEDY, Paul V (*J* 1936, *S* 1934), 105 Avon St, New Haven, Conn, and (for mail), 4915 Forbes St, Pittsburgh, Pa

KENT, J. King (*J* 1928), Pres (for mail), J King Kent Co, Inc, 6477 Manchester, and 530 N Union St, St Louis, Mo

KENT, Laurence F. (*A* 1927, *J* 1924), Pres (for mail), Moncrief Furnace Co, P O Box 1673, Atlanta, and R F D No 2, Smyrna, Ga

KENT, Richard L. (*M* 1936), Dist Mgr (for mail), Trane Co of Canada, Ltd, 138 Portage Ave, and 31 Kennedy St, Winnipeg, Man, Canada.

KEPLER, Donald A. (*J* 1936, *S* 1934), Draftsman, Gibbs & Cox, Inc, 21 West St, New York, N Y, and (for mail), 30 Maplewood Ave, Maplewood, N J

KEPLINGER, William L (*M* 1929), Combustioneer Corp, 409 Tenth St, S W, and 1437 Rhode Island Ave N W, Washington, D C

KEPPNER, Harry W. (*M* 1930), (for mail), 1310 South 56th Ave, and 1245 S Austin Blvd, Cicero, Ill

KERN, Raymond T. (*M* 1927), Chief Engr, Jennison Co, 17 Putnam St, Fitchburg, and (for mail), 51 Claflin St, Leominster, Mass

KERSHAW, Melville G. (*M* 1932, *A* 1926; *J* 1921), Vtg and Air Cond Engr (for mail), E I DuPont de Nemours & Co, Wilmington, Del, and 7313 North 21st St, Philadelphia, Pa

KESSLER, Jacob (*M* 1936), Pres (for mail), Jacler Heating Co, Inc, 3810 Third Ave, and 2115 Ryer Ave, New York, N Y

KEYES, Robert E. (*M* 1913), Asst to Chief Engr in charge of Air Cond, Surface Combustion Corp, Toledo, Ohio

KEYS, Lee F. (*A* 1936, *S* 1934), Route 1, Box 366, Phoenix, Ariz

KEYSER, Herman M. (*A* 1937), Sales Engr, Murray W Sales & Co, and (for mail), 3007 Whitney Ave, Detroit, Mich.

KICZALES, Maurice D (*M* 1935), Mech Engr, U S Army Motion Picture Service, 726 Jackson Place N W, and (for mail), 3000 Connecticut Ave, Washington, D C

KIEFER, Carl J (*M* 1922), Consulting Engr (for mail), 611 Schmidt Bldg, and 984 Lennox Place, Cincinnati, Ohio

KIEFER, E J., Jr (*A* 1932, *J* 1928), Treas and Gen Mgr (for mail), H C Archibald Co, 8 S Sixth St, and 108 N Sixth St, Stroudsburg, Pa

KIESLING, Justin A (*M* 1930), Pres (for mail), Robischung Kiesling, Inc, 4818 Main St, P O, Box 1295, and 1806 Holman Ave, Houston Texas

KILNER, John S (*M* 1929), Mfrs Agent (for mail), Kilner Co, 7310 Woodward Ave, and 1091 Seminole Ave, Detroit, Mich

KILPATRICK, W. S (*M* 1923), W S Kilpatrick & Co (for mail), 1682 W Washington Blvd, and 943 S Cochran St, Los Angeles, Calif

KIMBALL, Charles W (*M* 1915), Treas, (for mail), Richard D Kimball Co, 6 Beacon St, Boston, and 65 Prescott St, West Medford, Mass

KIMBALL, Dwight D.[1] (*M* 1908), (*Presidential Member*), (Pres, 1915, 2nd Vice-Pres, 1914; Board of Governors, 1912-1916), Consulting Engr (for mail), Room 1728 Grand Central Terminal Bldg, and 307 East 44th St, New York, N Y

KIMMELL, Phillip M (*J* 1936), Dist Engr, L R Krumm Co, (Delco-Frigidaire Cond Corp), 121 E Gay St, Columbus, Ohio

KINCAIDE, Merrill C (*J* 1936), Air Cond Engr (for mail), Timken Silent Automatic Division, 100-400 Clark Ave, and 1160 Seward Ave, Detroit, Mich

KING, Arthur C (*M* 1936), Consulting Engr (for mail), 35 S Dearborn St, and 2018 Lane Court, Chicago, Ill

KING, Harry K. (*A* 1937), Dist Mgr, Tube-Turns, Inc, Louisville, Ky, and (for mail), 5657 Horger Ave, Dearborn, Mich

KING, Roy L. (*J* 1936, *S* 1933), Engr, Lewis Air Conditioners, Inc, 1600 Broadway N E, and (for mail), 2538 Clinton Ave S, Minneapolis, Minn

KINGSLAND, George D. (*M* 1935), Vice-Pres (for mail), Minneapolis-Honeywell Regulator Co, 2753 Fourth Ave S, and 2036 Queen Ave S, Minneapolis, Minn

KINGSWELL, William E. (*M* 1935), Pres (for mail), W E Kingswell, Inc, 3707 Georgia Ave N W, and 2739 Macomb St, N W Washington, D C

KINNEY, Aldon M (*M* 1936), Pres-Treas, (for mail), A M Kinney, Inc, 1828 Carew Tower, Cincinnati, and 3812 Beech St, Mariemont, Ohio.

KIPE, J. Morgan (*M* 1919), Spencer Heater Co, 2020 N Broad St, Philadelphia, and (for mail), 801 Homestead Ave, Beechwood, Del Co, Pa

KIRKPATRICK, Arthur H (*M* 1935, *J* 1931), Salesman, Ilg Electric Ventilating Co, 415 Brainard, and (for mail), Hotel Webster Hall, Detroit, Mich

KISTLER, Milton L. (*A* 1936), Owner-Engr (for mail), Kistler's Air Conditioning & Sheet Metal Works, Box 67, and 609 S Lexington Ave, Burlington, N C

KITAURA, Shigeyuki (*M* 1918), 101 Gotanda 6 Chrome, Shinagawa-ku, Tokyo, Japan

KITCHELL, Herbert N. (*A* 1926), Retired, 4328 Circle Ave, Cincinnati, Ohio

KITCHEN, Francis A. (*A* 1927, *J* 1923), Pres (for mail), American Warming & Ventilating Co, 1514 Prospect Ave, and 2077 Campus Rd, South Euclid, Cleveland, Ohio

KITCHEN, John H. (*M* 1906), Owner and Mgr (for mail), John H Kitchen Co, 1016 Baltimore Ave, and 5015 Westwood Terrace, Kansas City, Mo

KLEIN, Albert R (*M* 1920), Managing Dir (for mail), Lufttechnische Gesellschaft m b H, Stuttgart W, Konigstrasse 84, and Stuttgart N, Panoramastr 23, Germany

KLEIN, Edward W (*M* 1917), S E Dist Mgr (for mail), Warren Webster & Co, 152 Nassau St N W, and 456 Peachtree Battle Ave, Atlanta, Ga

KLIE, Walter (*M* 1915), Pres (for mail), The Smith & Obey Co, 6107 Carnegie Ave, Cleveland, and 18111 S Woodland Ave, Shaker Heights, Ohio

KNAB, Edward A. (*M* 1930, *A* 1927), Prop, E A Knab Heating Contractor (for mail), 1707 E Lake Bluff Blvd, Milwaukee, Wis

KNAPP, Donald S (*A* 1936), Branch Mgr (for mail), Chamberlin Metal Weather Strip Co, Inc, 2400 Hennepin Ave, and 4607 Wooddale Ave, Minneapolis, Minn

KNAPP, Joseph H (*M* 1936), Designing, Htg and Vtg Equip, Utica Products Corp, and (for mail), 111 Lowell Ave, Utica, N Y

KNIBB, Alfred E. (*M* 1930), Htg Engr (for mail), L 1 McConachie Co, 1003 Maryland Ave, and 9333 E Jefferson Ave, Detroit, Mich

KNOPF, Charles (*A* 1936, *J* 1935, *S* 1933), 1201 Liberty Ave, Brooklyn, N Y

KNOWLES, Mahlon G (*M* 1935), Instructor in Applied Science, Wentworth Institute, 550 Huntington Ave, Boston, and (for mail), 255 Burrill St, Swampscott, Mass

KNOX, James R (*M* 1930), Consulting Engr, 26 Commercial St, Dundee, Angus, Scotland

KOCH, Richard G (*A* 1935), Htg and Air Cond Engr, Milwaukee Gas Light Co, 626 E Wisconsin Ave, and (for mail), 731 North 34th St, Milwaukee, Wis

KOEHLER, C Stewart (*A* 1936), Sales Engr (for mail), Minneapolis-Honeywell Regulator Co, 801 Second Ave, and 361 East 163rd St, New York, N Y

KOHLER, Walter J., Jr (*A* 1933), Htg Sales Supervisor (for mail), Kohler Co, and 605 W Park Lane, Kohler, Wis

KONZO, Seichi (*M* 1937, *A* 1936, *J* 1932), Special Research Associate, University of Illinois, 213 Mech Engrg Lab, and (for mail), 1108 W. Stoughton St, Urbana, Ill

KOOISTRA, John F. (*M* 1933), Engr (for mail), Carrier Corp, 748 E Washington St, and 6063 Roy St, Los Angeles, Calif

KORN, Charles B (*M* 1922), Member of Firm, Reber-Korn Co, 817 Cumberland St, and (for mail), 1022 S. Eighth St, Allentown, Pa

KOTTCAMP, Horace A. (*M* 1915), Mgr (for mail), Kottcamp Construction Co, 147 N Second St, and 527 Philadelphia Ave, Chambersburg, Pa.

KOZU, Tamiichiro (*M* 1930), Chief Engr (for mail), Japan Radiator Industrial Association, 506 Marunouchi Bldg, and 1701 Yonchome Shimoochiai, Yodobashiku, Tokyo, Japan

KRAMIG, Robert E., Jr. (*A* 1933), Vice-Pres, Treas (for mail), R E Kramig & Co, Inc, 222-4 East 14th St, Cincinnati, and 51 Central Terrace, Wyoming, Ohio.

KRAMINSKY, Victor (*M* 1936), Managing Dir (for mail), Air Conditioning & Engineering, Ltd, 4-12 Palmer St, Westminster S W. 1, and 36 Manor Court, Aylmer Rd, Highgate, London, England.

KRATZ, Alonzo P.* (*M* 1925), Research Prof (for mail), Dept of Mech Engrg, University of Illinois, and 1003 Douglas Ave, Urbana, Ill

KREISSL, Hans G. (*M* 1925), Mgr, Vento Dept (for mail), American Radiator Co, 816 S Michigan Ave, Chicago, and 408 Lee St, Evanston, Ill

KRENZ, Alfred S (*A* 1935), Pres (for mail), Krenz & Co, 5114 W Center St, Milwaukee, and 1766 North 74th St, Wauwatosa, Wis

KREZ, Leonard (*A* 1935), Secy (for mail), Paul J Krez Co, 444 N. LaSalle St, and 4716 N Paulina St, Chicago, Ill.

KRIBS, Charles L, Jr (*M* 1935), Consulting Mech Engr, Kribs & Landauer, 807 Southwestern Life Bldg, and (for mail), 4209 Shenandoah Ave, Dallas, Texas

KRIEBEL, Arthur E. (*M* 1920), Sales Engr (for mail), Haynes Selling Co, Inc, Ridge Ave and Spring Garden St, Philadelphia, and Warren Ave, Berwyn, Pa

KRINTZMAN, Harry (*S* 1936), Carnegie Tech (for mail), 1929 Forbes St, Pittsburgh, Pa, and 44 Chamberlain Parkway, Worcester, Mass

KROEKER, J Donald (*M* 1936), Pres (for mail), Columbia Engineering Co, 610 Failing Bldg, and 6831 N E Siskiyou St, Portland, Ore

KRUEGER, James I. (*M* 1921), Repr (for mail), Illinois Engineering Co, 357 Ninth St, and 1920 Sacramento St, San Francisco, Calif

KUBASTA, Robert W. (*J* 1936), Sales Engr (for mail), Bryant Air Conditioning Corp, 230 Park Ave, New York, N Y, and 1088 Summit Ave, Lakewood, Ohio

KUEHN, Walter C (*A* 1933), Kuehn Heating & Ventilating Co, 915 Seventh Ave S, Minneapolis, Minn

KUEMPEL, Leon L (*M* 1936, *J* 1929), Delco-Frigidaire Sales Corp, Dayton, Ohio, and (for mail), 643 Park Drive N E, Atlanta, Ga

KUHLMANN, Rudolf (*M* 1928), Engr, 122 East 42nd St, New York, N Y

KUNS, Joseph W. (*J* 1936, *S* 1935), 2110 Hawthorne Ave, Minneapolis, Minn

KURTZ, Robert W. (*J* 1936), Htg and Air Cond Inspector, City of Minneapolis (for mail), 213 City Hall, and 1513 W Lake St, Minneapolis, Minn

KWAN, I. K (*M* 1933), Gen Mgr, China Engineering Co, 30 Brenan Rd, Shanghai, China

KYLBERG, V. C (*A* 1934), Owner, Mgr, Contr Engr, 607 Ridgewood Rd, Maplewood, N J

KYLE, W J. (*A* 1935), Power Sales Engr (for mail), Byllesby Engineering & Management Corp, 231 S LaSalle St, and 1239 Jarvis Ave, Chicago, Ill

L

LAFFOLEY, Laurence H (*A* 1936), Asst Engr (for mail), Canadian Pacific Railway, Room, 401, C P R Windsor Station, and 175 Woodlands, Montreal, Que, Canada

LAGODZINSKI, Harry J. (*A* 1927, *J* 1920), Sales Engr, Ilg Electric Ventilating Co, 182 N LaSalle St, and (for mail), 3628 N Tripp Ave, Chicago, Ill

LAMBERT, Robert D. (*M* 1936), Design Engr (Air Cond), American Radiator Co (for mail), P O Box 356, New Rochelle, and 89 Young Ave, Pelham, N Y

LaMONTAGNE, Arthur F (*A* 1936), Sales Mgr Htg Div (for mail), Gurney Foundry Co, Ltd, P. O Box 1149, Montreal, and 24 Prince Arthur St, St Lambert, Que, Canada

LANDAUER, Leo L. (*J* 1932), Consulting Engr (for mail), Kribs & Landauer, Houseman Bldg, and 5705 Velasco, Dallas, Texas

LANDERS, John J (*M* 1930, *J* 1924), Mfrs. Agent (for mail), 701 Crosby Bldg, Buffalo, and 120 Burroughs Drive, Snyder, N Y

LANE, D Duffy (*M* 1934), Mgr (for mail), Weber & Merritt, Inc, 75-12 Roosevelt Ave, and 94-06-34th Rd, Jackson Heights, N. Y

LANGE, Fred F. (*A* 1934), Asst. Vice-Pres (for mail), Mechanical Service Co, 602 Pence Bldg, Minneapolis, and Route 3, Box 80, Excelsior (Cottagewood L M), Minn

LANGE, Raymond T. (*M* 1936), Engr, Hartzell Propeller Fan Co, Box 909, and (for mail), 1700 N Broadway St., Piqua, Ohio

LANGENBERG, Everett B. (*M* 1914), (Council, 1926-1931), Pres (for mail), Langenberg Heating Co, 3800 W Pine Blvd, and 6031 Enright, St Louis, Mo

LANNING, E. K. (*A* 1927), Asst Secy and Sales Mgr. (for mail), Warren Webster & Co, Camden, and Box 311, Clayton, N J

LANOU, J. Ernest (*M* 1931), Mgr (for mail), F. S Lanou & Son, 90 St Paul St, and 48 Brookes Ave, Burlington, Vt

LaROI, George, II, (*J* 1936), Engrg Correspondent (for mail), McDonnell & Miller, Room 1316, Wrigley Bldg, 5536 N Campbell Ave, Chicago, Ill

LARSON, Carl W. (*M* 1936), Service Engr, Barnes & Jones, Inc, 128 Brookside Ave, Jamaica Plain, and (for mail), 641 Hyde Park Ave, Roslindale, Mass

LARSON, Clifford P. (*J* 1936), Sales Engr (for mail), Insulite Co, 205 W Wacker Drive, and 6317 Kenmore Ave, Chicago, Ill

LARSON, Gustus L+ (*M* 1923), (*Presidential Member*), (1st Vice-Pres, 1935, 2nd Vice-Pres, 1934 Council, 1929-1936), Prof, Steam and Gas Engrg, and Chairman of Dept of Mech Engrg (for mail), University of Wisconsin, Mech Engrg Bldg, and 1213 Sweetbriar Rd, Shorewood Hills, Madison, Wis

LaSALVIA, James J (*M* 1930), Mech Engr, Frigidaire Corp, and (for mail), 2250 Emerson Ave, Dayton, Ohio

LAUER, Harold B. (*M* 1930), Vice-Pres (for mail), English & Lauer, Inc, 309 West 12th St, and 1121 S Hayworth Ave, Los Angeles, Calif

LAUER, Rodney F. (*J* 1936), Sales Engr, York Ice Machinery Corp, 1238 North 44th St, Philadelphia, and (for mail), 236 Glentay Rd, Lansdowne, Pa

LAUFKETTER, Fred C. (*M* 1936), Supt and Chief Engr (for mail), New Hotel Jefferson, 12th and Locust St, and 7056 West Park Ave, St Louis, Mo

LAUTENSCHLAGER, Fred (*M* 1915), Vice-Pres, Treas, Kroeschell Boiler Co (Office and Factory), 100 Reichert Ct, Racine, Wis, and (for mail), 3846 Alta Vista Terrace, Chicago, Ill

LAUTERBACH, Henry, Jr. (*M* 1935), Engr (for mail), Carrier Corp, 180 N Michigan Ave, and 6959 Merrill Ave, Chicago, Ill

LAUTZ, Fritz A. (*M* 1936), Dist Engr, Kelvinator Corp, and (for mail), 319 Howell Ave, Cincinnati, Ohio.

LAWLOR, John J. (*M* 1935), Mgr Htg Div, James Robertson Co, Ltd, 215 Spadina Ave, and (for mail), 35 Tennis Cres, Toronto, Ont, Canada

LEDGETT, F. Donald (*S* 1936), 108 Clinton St, Toronto, Ont, Canada

LEE, Robert T. (*S* 1936), Eastman Kodak Co, 343 State St, and (for mail), 300 Meigs St, Rochester, N Y

LEEK, Walter (*M* 1903), Managing Dir (for mail), Leek & Co, Ltd, 1111 Homer St, and 4769 W. Second Ave, Vancouver, B C, Canada

LEES, Herbert K. (*M* 1924; *J* 1912), Secy-Treas. (for mail), William Lees, Inc, 542 W Washington St, and 5855 N Kenneth Ave, Chicago, Ill

LEGLER, Frederick W (*M* 1935; *A* 1933), Sales Mgr, Waterman-Waterbury Co, 1121 Jackson St N E, and (for mail), 2919 Johnson St N.E, Minneapolis, Minn

LEICHNITZ, Robert W. (*J* 1936), Estimator, Leichnitz Engineering Co, 14 E. A St, and (for mail), 2506 W Chestnut St, Yakima, Wash

LEILICH, Roger L. (*M* 1922), Pres. (for mail), Baltimore Heat Corp, 2000 W Pratt St, and 2810 Elsinor Ave, Baltimore, Md

LEINROTH, J. Paul (*M* 1929), Gen Industrial Fuel Repr (for mail), Public Service Electric & Gas Co, 80 Park Place, Newark, and 37 The Fairway, Montclair, N J

LEITCH, Arthur S. (*M* 1908), Pres and Managing Director (for mail), Arthur S Leitch Co, Ltd., 1123 Bay St, and 421 Russell Hill Rd, Toronto, Ont, Canada.

LELAND, Warren B. (*M* 1929), Sales Engr, H. B Smith Co, Westfield, and 34 Leyfred Terrace, and (for mail), P O Box 1522, Springfield, Mass

LELAND, William E. (*M* 1915), Consulting Engr (for mail), Leland & Haley, 58 Sutter St, San Francisco, and 704 The Alameda, Berkeley, Calif

LENIHAN, William O (*A* 1936), Pres (for mail), W O Lenihan, Inc, 122 Pearl St, and 703 W. Ferry St, Buffalo, N Y

LEONHARD, Lee W (*M* 1936), Supv, Eastman Kodak Co, Kodak Park, and (for mail), 1075 Winona Blvd, Rochester, N Y

LEOPOLD, Charles S. (*M* 1931), Consulting Engr (for mail), 213 S Broad St, Philadelphia, and 614 Elkins Ave, Elkins Park, Pa

LESLIE, Donald E. (*J* 1936, *S* 1933), Sales Engrg Dept (for mail), Bryant Heater Co, 17825 St. Clair Ave, Cleveland, Ohio, and 3541 Bloomington Ave, Minneapolis, Minn

LEUPOLD, Herbert W (*J* 1933), Time Study Engr, S S White Dental Mfg Co, Princes Bay, Staten Island, and (for mail), 147 Sprague Ave, Tottenville, S I, N Y

LEVENTHAL, Bernard (*S* 1935), Estimating and Designing Engr, Schwerin Air Conditioning Corp, 570 Lexington Ave, New York, and (for mail), 3913-13th Ave., Brooklyn, N Y

LEVERANCE, Herbert J (*A* 1935), Salesman (for mail), J M O'Connor Co, 434 N Rock Island, Wichita, Kans

LEVY, Marion I. (*A* 1936, *J* 1931), Sales Mgr and C Engr (for mail), Viking Air Conditioning Corp, 1935 Euclid Ave, and 1273 West 108th St, Cleveland, Ohio

LEWIS, Carroll E (*M* 1930), Chief Engr, Delco-Frigidaire Conditioning Division, General Motors Corp, 1420 Wisconsin Blvd, and (for mail), 1660 Catalpa Drive, Dayton, Ohio

LEWIS, L. Logan+ (*M* 1918), Chief Engr-Vice-Pres (for mail), Carrier Corp, 850 Frelinghuysen Ave, Newark, and 724 Carlton Ave., Plainfield, N J

LEWIS, Samuel R.* (*M* 1905), (*Presidential Member*), (Pres, 1914, 2nd Vice-Pres, 1910; Board of Governors, 1909-1910-1912; Council, 1914-1915), Consulting Mech Engr (for mail), 407 S Dearborn St, and 4737 Kimbark Ave, Chicago, Ill

LEWIS, Thornton* (*M* 1919), (*Presidential Member*), (1st Vice-Pres, 1928; 2nd Vice-Pres., 1927, Council, 1923-1930), Pulp Products Co, Inc, 60 East 42nd St, New York, N Y, and (for mail), 156 Irving Ave, South Orange, N J

LIBBY, Ralph S. (*J* 1933), Air Cond Engr, Arthur S Leitch Co, Ltd, 1123 Bay St, and (for mail), 275 Winona Drive, Toronto, Ont., Canada

LICHTY, Charles P. (*M* 1920), Pres (for mail), C P Lichty Engineering Co, Inc, 400*M* South 21st St, and 125 Windsor Drive, Birmingham, Ala

LIEBRECHT, Walter J. (*J* 1936), Sales Engr. (for mail), American Radiator Co, Fourth and Channing Sts. N E., and 3032 Rodman St N W., Washington, D C

LIGHTHART, Charles H (*M* 1935), Mfrs Sales Engr. (for mail), 254 Court St, and 19 E. Winspear Ave, Buffalo, N Y.

LILJA, Oscar L. (*J* 1936), Mech. Engr, Toltz, King & Day, Inc, 1509 Pioneer Bldg, St. Paul, and (for mail), 2543 Clinton Ave, Minneapolis, Minn

LINCOLN, Roland L. (*M* 1935), Engr, Hoffman Specialty Co, 193 Grand St, and (for mail), 487 Farmington Ave, Waterbury, Conn.

LINDBERG, Arthur F. (*J* 1935, *S* 1933), Administrative Inspector of Colorado Parks, National Park Service, 915 Midland Savings Bldg, and (for mail), 428 Humboldt, Denver, Colo

LINGO, Charles K (*A* 1936; *J* 1935), Air Cond Engr (for mail), Florida Power & Light Co, and 2224 S W Sixth St, Miami, Fla

LINN, Homer R. (*M* 1914), Consulting Engr, 321 S Ashland Ave., LaGrange, Ill

LINSENMEYER, Francis J. (*M* 1935), Head, Dept, Mech Engrg (for mail), University of Detroit, Livernois & McNichols, and 17375 Prairie Ave, Detroit, Mich

LINTON, John P. (*M* 1927), Vice-Pres and Managing Dir, Garth Co, St Henry Postal Station, Drawer 5, and (for mail), 247 Brock Ave N, Montreal W, P Q, Canada
LITTLE, Kenneth B (*A* 1935), Sales Engr (for mail), The Brownell Co, 736 Dixie Terminal Bldg, and 2132 Crane Ave, Cincinnati, Ohio
LITTLEFORD, Wallace H (*M* 1936), Estimating Engr (for mail), E J Febrey & Co, 616 New York Ave N W, Washington, D C., and Hyattsville R F D No 1, Md
LIVAR, Allen P (*M* 1935), Mgr., Mech Equip Div (for mail), Reynolds Corp, 19 Rector St, New York, and 830 Bronx River Rd, Bronxville, N Y
LLOYD, Edmund H (*J* 1936), Sales Engr, Washington Gas Light Co, 411 Tenth St N W, and (for mail), 2601 Calvert St N W, Washington, D C
LLOYD, Edward C (*M* 1927), Director of Tech Service (for mail), Armstrong Cork Products Co., and 129 W Walnut St, Lancaster, Pa
LOCHHEAD, Kenneth Y (*J* 1930), Asst. to Supt of Bldgs (for mail), Hudson's Bay Co., 79 Main St, and 19 Royal Oak Court, Winnipeg, Man, Canada
LOCKE, Robert A (*M* 1935), Mgr, Steel Heating Boiler Institute, and (for mail), 500 N Union St., Middletown, Pa
LOCKHART, Harold A (*A* 1936; *J* 1935), Chief Engr, Bell & Gossett Co, 3000 Wallace St, and (for mail), 11719 Hale Ave, Chicago, Ill
LOCKHART, William R. (*J* 1936), Air Cond Engr (for mail), York Ice Machinery Corp, 215 Investment Bldg, and 5310 Reno Rd N W, Washington, D C
LOEFFLER, Frank X. (*M* 1914), Pres (for mail), Loeffler-Greene Supply Co, 1604 N W Fifth St., and 320 Northwest 26th St, Oklahoma City, Okla
LOEFFLER, Louis, Jr. (*J* 1936, *S* 1934), 1815 W Ninth St, Oklahoma City, Okla
LOFTE, John A. (*J* 1936, *S* 1933), Engr, Pflugradt Co, 215 W Kilbourn Ave, and (for mail), 2305 W. Wisconsin Ave, Milwaukee, Wis
LOGAN, Thomas M. (*S* 1935), 330 North St, Murphysboro, and (for mail), 108 N Romine St, Champaign, Ill
LOH, Nan-Shee (*M* 1933, *A* 1931; *J* 1927), Mgr. (for mail), New Shanghai Heating & Plumbing Co, Room 330, National Commercial Bank Bldg, 400 Kiangse Rd, Shanghai, China
LONG, Dewey J. (*A* 1937), Sales Engr (for mail), Wm G Boales Co, 6439 Hamilton Ave, Detroit, and 466 Pearson Ave, Ferndale, Mich
LONG, Wayne E. (*M* 1935), Assoc. Prof. of Mech. Engrg, Texas Agricultural & Mechanical College, College Station, Texas
LONGCOY, Grant B. (*M* 1933), Maintenance Engr., Cleveland Board of Education, E Sixth St, and (for mail), 1215 Ramona Ave., Lakewood, Ohio.
LOO, Ping Yok (*M* 1933), Gen. Mgr. (for mail), China Engineering Co., 774 N Chung San Rd, Nanking, and 271-73 Dumbarton Rd, Tientsin, China.
LOUGHRAN, Patrick H., Jr. (*J* 1937), Cadet Engr, Washington Gas Light Co, 1100-29th St N.W., and (for mail), 4513-49th St., Washington, D. C.
LOVE, Clarence H. (*M* 1919), Mfrs Agent, Nash Engineering Co., 317 Chamber of Commerce, and (for mail), 289 Norwalk Ave., Buffalo, N Y
LOVING, William H. (*J* 1936), Htg Engr., Washington Gas Light Co, 411 Tenth St N W., and (for mail), 110 Maryland Ave N E, Washington, D C
LOWNSBERY, Benjamin F. (*M* 1920), Htg Engr, Benjamin F Shaw Co, Second and Lombard Sts, and (for mail), 21 S Sycamore St., Wilmington, Del.
LUCK, Alexander W.* (*Life Member, M* 1919), Pres and Gen Mgr (for mail), Reading Heater & Supply Co., Church and Woodward Sts, Reading, and Reiffton, Pa
LUCKE, Charles E. (*M* 1924), Stevens Prof of Mech Engrg, Columbia University, and Consulting Engr, Babcock & Wilcox Co (for mail), Pupin Laboratories Bldg, Columbia University, and 186 Riverside Drive, New York, N Y
LUDERS, Richard H. (*J* 1936), 5642 W Division St, Chicago, Ill
LUND, Clarence E (*M* 1936, *J* 1935, *S* 1933), Warm Air Htg and Air Cond Inspector, Bldg Inspector, City of Minneapolis, 213 City Hall, and (for mail), 4817 12th Ave S, Minneapolis, Minn
LUTY, Donald J. (*M* 1933), Chief Engr Air Cond Div (for mail), Gar Wood Industries, Inc, 7924 Riopelle St, Detroit, and 911 Forest Ave, Ann Arbor, Mich
LUTZ, James H., Jr. (*M* 1928), Owner (for mail), 140 Paxton St, and 1601 Foster St, Harrisburg, Pa
LYLE, Ernest T. (*M* 1919), 52 Grammercy Park N, New York, N Y
LYLE, J. I.[F] (*M* 1911), (*Presidential Member*), (Pres, 1917, Council, 1917-1918), Pres (for mail), Carrier Corp, 850 Frelinghuysen Ave, Newark, and 1200 W Seventh St, Plainfield, N J.
LYMAN, Samuel E. (*A* 1924), Buensod, Stacey Air Conditioning, Inc, 60 East 42nd St, New York, N Y, and (for mail), 820 Canton St., Elizabeth, N. J
LYNCH, William L. (*M* 1928), Treas-Gen Mgr. (for mail), Rome Turney Radiator Co, Canal St, and 1413 N George St, Rome, N. Y.
LYON, P. S. (*M* 1929), Sales Engr. (for mail), Turbine Equipment Co, 75 West St, New York, N Y, and 42 Hawthorne Place, Summit, N J
LYONS, Cornelius J. (*A* 1932), Sales Engr (for mail), Nash Engineering Co, Wilson Ave, and 22 Haviland St., South Norwalk, Conn
LYONS, Michael A. (*M* 1935), Htg Contr, 238 West 20th St, New York, N Y.

M

MACCUBBIN, Howard A. (*M* 1934), Buyer, Htg. Materials, Montgomery Ward & Co, Chicago, and (for mail), 2135 Ridge Ave, Evanston, Ill.
MacDADE, Ambrose H. (*M* 1923), Sales (for mail), Burnham Boiler Corp, S E Cor. 31st and Jefferson Sts, Philadelphia, Pa, and 225 Haddon Ave, Westmont, N J
MacDONALD, Donald B. (*M* 1930), Mgr, C A Dunham Co, 101 E Walnut St, Kingston, Pa.
MacDONALD, Douglas J. (*M* 1935), Vice-Pres (for mail), Dominion Radiator & Boiler Co, Ltd, 1322 Dufferin St, and 96 Hudson Drive, Toronto, Ont, Canada
MacDONALD, Everett A. (*A* 1933), Branch Mgr. (for mail), Spencer Heater Co, 115 Broadway, Cambridge, and 154 Standish Rd., Watertown, Mass
MACHEN, James T. (*J* 1934), Chicago Branch Mgr (for mail), The Ric-wiL Co, 111 W Monroe St, and 420 Diversey Parkway, Chicago, Ill
MACHIN, Donald W. (*J* 1935), Fuel Engr, Pittsburg & Midway Coal Mining Co., 810 Dwight Bldg, Kansas City, Mo, and (for mail), 2029 Vermont St, Lawrence, Kans.
MACK, Ludwig (*M* 1935), Dist Mgr, Cooling & Air Conditioning Corp, Cresmont and Haddon Aves, Camden, N J, and (for mail), 246 W. Upsal St, Germantown, Philadelphia, Pa
MacLEOD, Kenneth F. (*A* 1933), Mgr Htg Dept, Crane Co., 419 Second Ave S, and (for mail), 7703 First Ave N E, Seattle, Wash
MacMILLAN, Alexander R. (*M* 1936), Sales Engr, Educational and Prom. Dept, Delco-Frigidaire Conditioning Corp, and (for mail), 130 Beverly Place, Dayton, Ohio
MACRAE, Robert B. (*J* 1935), Air Cond. Engr., E. J Nell Co, Manila, P I
MACROW, Lawrence (*J* 1936), Engr (for mail), Carrier Corp, 850 Frelinghuysen Ave, Newark, and Kearny, N J.

MADDUX, Oliver L. (*M* 1935, *A* 1933), Chief Engr, United Gas & Fuel Co of Hamilton, Ltd, 88 King St E, and (for mail), 18 Whitton Rd, Hamilton, Ont, Canada

MADELY, Frederick J. (*A* 1936), Asst Contract Engr, Eastern Steel Products, Ltd, 1335 Delorimier Ave, and (for mail), 6408 Louis-Hemon St, Montreal, Que, Canada

MADISON, Richard D (*M* 1926), Research Engr (for mail), Buffalo Forge Co, 490 Broadway, Buffalo, and 218 Brantwood Rd, Snyder, N Y

MAEHLING, Leon S. (*M* 1932), Supervisor Sales, Equitable Gas Co, 427 Liberty Ave, and (for mail), 448 Sulgrave Rd, Pittsburgh, Pa

MAGINN, Peter F (*Life Member*, *M* 1908), Mfrs Agent, P F Maginn & Co, 207 Fulton Bldg, Pittsburgh, Pa

MaGIRL, Willis J (*M* 1934, *A* 1931, *J* 1927), Chief Engr (for mail), P H MaGirl Foundry & Furnace Works, 401-13 E Oakland Ave, and 108 Warner Ave, Bloomington, Ill

MAHON, B B (*M* 1935), Principal, School of Air Cond (for mail), International Correspondence Schools, Ash St and Wyoming Ave, and 433 Fig St, Scranton, Pa

MAHONEY, David J (*M* 1930, *A* 1926), Branch Mgr (for mail), Johnson Service Co, 503 Franklin St, and 130 Depew Ave, Buffalo, N Y

MAIER, Albert H (*A* 1936), Chief Engr (for mail), Board of Education, Baltimore and Girard Ave, and 1021 Kuntz Ave, Middletown, Ohio

MAIER, George M (*M* 1921), Asst to Vice-Pres and Gen Mgr of Mfg (for mail), American Radiator Co, 8007 Jos Campau, Detroit, Mich

MAILLARD, Albert L. (*M* 1934), Consulting Engr, Head of Air Cond Div (for mail), Kansas City Power & Light Co, 1330 Baltimore Ave, P O Box 679, and 3740 Washington St, Kansas City, Mo

MALLIS, William (*M* 1914), 330 Lyon Bldg, Seattle, Wash

MALONE, Dayle G (*M* 1929, *A* 1925), Branch Mgr, Petroleum Heat & Power Co, 1725 S Michigan Ave, and 7315 Merrill Ave, Chicago, Ill

MALONE, James S (*A* 1936), Dist Sales Repr (for mail), Hoffman Specialty Co, 411 N Tenth St, and 7124 Waterman Ave, St Louis, Mo

MALVIN, Ray C. (*M* 1929), Pres. (for mail), Malvin & May, Inc, 2427 S Michigan Ave, and 8220 Dante Ave, Chicago, Ill

MANAHAN, James E. (*M* 1934), Vice-Pres, Commercial Engineering & Sales Corp, 2833 Olive St, and (for mail), 3521 Juniatia, St Louis, Mo

MANDEVILLE, Edgar W. (*M* 1914), 1171 East 37th St, Brooklyn, N Y

MANN, Arthur R. (*M* 1930), Owner, Mann & Co, Archt-Engrs, 721 Wiley Bldg, Hutchinson, Kans.

MANN, Lee B. (*J* 1930), Air Cond Engr (for mail), Carrier Corp, 850 Frelinghuysen Ave, Newark, and 300 Montgomery St, Bloomfield, N J

MANNING, Walter M. (*M* 1930), Htg Engr, Crane Co, 115 E Front St, Grand Island, and (for mail), P O. Box 112, Clarks, Nebr

MANNY, J Harvey (*A* 1936), Vice-Pres, Secy (for mail), Robinson Furnace Co, 213 W Hubbard St, and 5950 Midway Park, Chicago, Ill

MARCONETT, Vernon G. (*A* 1936), Engr, Factory Supt, Farquhar Furnace Co, and (for mail), 208 Fulton St, Wilmington, Ohio

MARIN, Axel* (*M* 1935), Assoc Prof, Mech Engrg (for mail), University of Michigan, 241 West Engrg Bldg, and P O Box 175, Ann Arbor, Mich

MARKS, Alexander A. (*A* 1930), Asst Sales Mgr, Richmond Radiator Co, 2241 N American St, Philadelphia, and (for mail), 6635 McCallum St, Germantown, Philadelphia, Pa

MARKUSH, Emery U (*M* 1931), Consulting Engr (for mail), 225 East 21st St, New York, and 8442-85th Rd, Woodhaven, L I, N Y

MAROTTA, John A (*S* 1936), 11116 Tuscora Ave, Cleveland, Ohio

MARRINER, John M S (*M* 1934), Vice-Pres (for mail), Taylor & Engineering Construction Co, Ltd, 11 King St E, and 111½ Balsam Ave, Toronto, Ont, Canada

MARSCHALL, Peter J (*M* 1930, *J* 1927), Engr, Kroeschell Engineering Co, 215 W Ontario St, and (for mail), 2201 W Touhy Ave, Chicago, Ill

MARSHALL, Alexander G (*A* 1936), Sales Engr (for mail), Trane Co of Canada, Ltd, 660 St Catherine St W, and 2353 Wellington St, Montreal, Que, Canada

MARSHALL, William D (*M* 1935), Branch Mgr (for mail), Noland Co, Inc, 1823 N Arlington Ridge Rd, and 1307 N Wakefield St, Arlington, Va

MARTENIS, John V (*M* 1918), Assoc Prof, University of Minnesota, and (for mail), 1800 Bloomington Ave, Minneapolis, Minn

MARTIN, Albert B (*M* 1917), Kewanee Boiler Co, 1858 S Western Ave, Chicago, Ill

MARTIN, George W† (*M* 1911), Supervising Engr (for mail), U S Realty & Improvement Co, 111 Broadway, New York, N Y, and 310 Prospect St, Ridgewood, N J

MARTIN, Leonard (*J* 1936), Sales Engr (for mail), H L Peiler & Co, Ltd, 159 New Birks Bldg, and 4518 St Catherine St W, Montreal, Que, Canada

MARTINEZ, Juan J. (*J* 1929), Research and Rate Engr, Mexican Light & Power Co, Ltd, Gante 20, and (for mail), Paseo de la Reforma 183, Mexico City, Mexico

MARTINKA, Paul D (*S* 1934), 13703 Chautauqua Ave, Cleveland, Ohio

MARTOCELLO, Joseph A (*M* 1934), Pres, Jos A Martocello & Co, 229 North 13th St, Philadelphia, Pa

MARTY, Edgar O (*M* 1916), Pres and Gen Mgr, Indian Head Anthracite, Inc, Thompson Bldg, and (for mail), 1775 Howard Ave, Pottsville, Pa

MARUM, Otto (*M* 1931), Plant Engr, Agfa Ansco Corp, 29 Charles St, and (for mail), 12 Grand Blvd, Binghamton, N Y

MATCHETT, James C. (*M* 1923), Vice-Pres and Gen Mgr (for mail), Illinois Engineering Co, 21st St and Racine Ave, and 9936 S. Winchester Ave Chicago, Ill

MATHER, Harry H (*A* 1920), Promotional Analysis Secy (for mail), Philadelphia Electric Co, 1000 Chestnut St, Philadelphia, and 373 Lakeview Ave, Drexel Hill, Pa

MATHIS, Eugene† (*M* 1922), (for mail), New York Blower Co, 32nd St and Shields Ave, Armour P O Station, and 9151 S Hoyne Ave, Chicago, Ill

MATHIS, Henry (*M* 1921), New York Blower Co, 32nd and Shields Ave, and (for mail), 10317 Oakley Ave, Chicago, Ill

MATHIS, Julien W (*A* 1921), New York Blower Co, 32nd St and Shields Ave, Chicago, Ill

MATTHEWS, John E. (*M* 1934; *A* 1934), Dist Mgr, B F Sturtevant Co, 1106 Commerce Bldg, and (for mail), 5042 Lydia St, Kansas City, Mo

MATZEN, Harry B. (*M* 1919), York Ice Machinery Corp, 42nd St. and Second Ave, Brooklyn, and (for mail), 16 Addison Place, Rockville Centre, L I, N Y.

MAUTSCH, Robert (*A* 1928), Engr, Managing Dir (for mail), Compagnie Belge Des Freins Westinghouse, 97 Avenue Louise, Brussels, Belgium

MAWBY, Pensyl (*M* 1934), Service Engr., Lehigh Navigation Coal Co, 143 Liberty St, New York, N Y, and (for mail), 718 Center Ave, River Edge, N J

MAXWELL, George W. (*M* 1935, *S* 1932), Engr, Kenealy & Maxwell, Main St, and (for mail), P O Box 422, Harwich Port, Mass

MAY, Clarence W (*M* 1933), Consulting Engr (for mail), 126 Smith Tower, and 902 W Halliday, Seattle, Wash

MAY, Edward M (*M* 1931), Engr, Combustioneer, Inc, 1825 S Michigan Ave, Chicago, and (for mail), 1022 Hayes Ave, Oak Park, Ill

MAY, George E (*M* 1933), Utilization Engr (for mail), New Orleans Public Service, Inc, 317 Baronne St, and 2031 Short St, New Orleans, La

MAY, James W. (*J* 1935), Asst Prof, H V & A C (for mail), College of Engrg, University of Kentucky, and 261 Lyndhurst St, Lexington, Ky

MAY, Maxwell F (*M* 1929), Secy-Treas (for mail), Malvin & May, Inc, 332 S Michigan Ave, Chicago, and Palos Park, Ill

MAYES, Curtis (*S* 1936), 1002 Northwest 19th St, Oklahoma City, Okla

MAYETTE, Charles E (*M* 1926), P O Box 523, Washington, D C

MAYNARD, Herbert R (*J* 1936; *S* 1935), 1725 Woodland Ave, Duluth, Minn

MAYNARD, J Earle (*M* 1931), Chief Htg Engr, Fox Furnace Co, Woodford St, and (for mail), 324 Fifth St, Elyria, Ohio

McCAULEY, James H (*M* 1921), Pres (for mail), J H McCauley, Inc, 5558 West 65th St, Chicago, and 707 William St, River Forest, Ill

McCLELLAN, James E. (*M* 1922), Branch Mgr (for mail), American Blower Corp, Room 1401, 228 N LaSalle St, Chicago, and 8814 LaCrosse Ave, Niles Center, Ill

McCLINTOCK, Alexander, Jr (*M* 1928, *J* 1920), Member of Firm (for mail), A M McClintock's Sons, 1037 Ridge Ave, and 121 Rochelle Ave, Philadelphia, Pa

McCLINTOCK, William (*M* 1935), Supervising Engr, Department of Parks, City of New York, 37-13 West 65th St, and (for mail), 2 N Pinehurst Ave, New York, N Y

McCLOUGHAN, Charles (*A* 1936, *S* 1934), 11 Cottage St E, Norwalk, Conn, and (for mail), 279 Ryerson St, Brooklyn, N Y

McCONACHIE, Lorne L. (*A* 1928), Htg and Plbg, 1003 Maryland Ave, and (for mail), 1379 Maryland Ave, Detroit, Mich

McCONNER, Charles R (*A* 1925, *J* 1922), Gen Sales Mgr (for mail), Clarage Fan Co, and 1904 Waite Ave, Kalamazoo, Mich

McCORMACK, Denis (*M* 1933), Mgr, Air Cond Instruments and Controls Dept (for mail), Julien P Friez & Sons, Inc, 4 N Central Ave, Baltimore, and Lutherville Post Office, Baltimore County, Md

McCOY, C E (*M* 1930), Engr and Contractor, Brook Turner, Warren, Ark (for mail), P O. Box 222, Benton, and Little Rock, Ark

McCOY, T. F. (*M* 1924), Mgr (for mail), Powers Regulator Co, 125 St Botolph St, Boston, and Glen Rd, Wellesley Farms, Mass

McCRAE, George W. (*A* 1936), Mech Engr, John McCrae Machine & Foundry Co, 81 William St N, and (for mail), 51 Bond St, Lindsay, Ont, Canada

McCREA, Joseph B (*M* 1937), Owner, Heating & Ventilating, 3039 Coplin Ave, Detroit, Mich

McCREERY, Hugh J (*M* 1922), (for mail), Marine Bldg, and 1617–49th Ave W, Vancouver, B C

McCRIMMON, A Murray (*A* 1935), Asst Secy and Controller (for mail), Hydro-Electric Power Commission, 620 University Ave, Toronto 2, Ont, and 83 Glen Rd, Toronto, Ont, Canada

McCULLOUGH, Henry G. (*M* 1936), Mgr, Commercial Dept, S S Fretz, Jr, Inc, 2514 N Broad St, Philadelphia, and (for mail), 328 Glen Echo Rd, Germantown, Philadelphia, Pa

McCUNE, Byron V. (*M* 1928), 2310 W Yakima Ave, and (for mail), 101 W. Yakima Ave, Yakima, Wash

McDONALD, Anthony K (*A* 1936), Zone Mgr., Oil Htg Div, Standard Oil Co of New Jersey, 261 Constitution Ave, and (for mail), 3035 Rodman St N W, Washington, D C

McDONALD, Thomas (*A* 1931), Mgr (for mail), Minneapolis-Honeywell Regulator Co, Ltd, 117 Peter St, and 56 Kingsway, Toronto, Ont, Canada

McDONNELL, Everett N. (*M* 1923), Pres (for mail), McDonnell & Miller, Wrigley Bldg, and Drake Hotel, Chicago, Ill

McDONNELL, John E. (*A* 1936), Sales Engr (for mail), McDonnell & Miller, 400 N Michigan Ave, and 6101 Sheridan Rd, Chicago, Ill

McDOWELL, Bert W (*J* 1935), Scobell & Winston, 2027 State St, Erie, Pa

McELGIN, John W * (*J* 1931), 180 Rowland Park, Cheltenham, Pa

McELHANEY, Gerald W. (*J* 1936), Air Cond Engr (for mail), Pennsylvania Power Co, and 421 Norwood Ave, New Castle, Pa

McEWAN, Eugene E (*M* 1936), Sales Mgr, Air Cond Div (for mail), Frigidaire Corp, 224 West 57th St, New York, and Seawane Club, Hewlett Harbor, L I, N. Y

McGAUGHEY, John E, Jr. (*J* 1935), Air Cond Engr (for mail), Carrier Corp, 408 Chrysler Bldg, New York, N Y, and Hotel Winfield Scott, Elizabeth, N J

McGINNESS, J. E. (*M* 1903), Pres (for mail), McGinness, Smith & McGinness Co, 527 First Ave, and 142 Bellfield Ave, Pittsburgh, Pa

McGONAGLE, Arthur (*M* 1932), Consulting Engr (for mail), 1013 Fulton Bldg, Pittsburgh, and 6815 Prospect Ave, Ben Avon, Pa

McGRAIL, Thomas E (*M* 1926), Mgr, Htg Dept, Crane, Ltd, Beaver Hall, and (for mail), W A 9087, 3465 Belmore Ave, Montreal, P Q, Canada

McGUIGAN, L. A. (*A* 1919), Salesman, National Radiator Corp, and (for mail), 724 Hastings St, Pittsburgh, Pa

McILVAINE, John H * (*M* 1929), Pres and Treas (for mail), McIlvaine Burner Corp, 663 W Washington Blvd, and 1100 Lake Shore Drive, Chicago, Ill

McINTIRE, James F. (*M* 1915, *A* 1914), (Council, 1926-1928; 1932-1936), Vice-Pres (for mail), U S. Radiator Corp, 1056-44 Cadillac Square, P O Box 686, and 3261 Sherbourne Rd, Detroit, Mich

McINTOSH, Fabian C. (*M* 1921, *J* 1917), Branch Mgr (for mail), Johnson Service Co, 1238 Brighton Rd, and 302 Marshall Ave, Pittsburgh, Pa

McKEEMAN, Clyde A (*M* 1936), Asst Prof. of Mech Engrg (for mail), Case School of Applied Science, Cleveland, and 1359 Lynn Park Drive, Cleveland Heights, Ohio

McKIEVER, William H.† (*Life Member, M* 1897; *J* 1896), Pres (for mail), William H McKiever, Inc, 247 West 13th St, New York, and 479 Eighth St, Brooklyn, N Y

McKINLEY, Carroll B. (*J* 1936, *S* 1934), Service Mgr (for mail), New York Lipman Corp, 1716 Main St, and 90 N Pearl St, Buffalo, N Y.

McKINNEY, William J. (*A* 1934), Mgr, Atlanta Dist, American Blower Corp, 716-101 Marietta St Bldg, Atlanta, Ga

McKITRICK, Walter D. (*M* 1936), Htg-Vtg Engr (for mail), Mills, Rhines, Bellman & Nordhoff, Inc, 518 Jefferson Ave, and 3038 Gunckel Blvd, Toledo, Ohio.

McKITTRICK, Percy A. (*A* 1934), Treas-Gen Mgr. (for mail), Parks-Cramer Co, 970 Main St, and 219 Blossom St, Fitchburg, Mass

McLAREN, Fred S. (*J* 1935), Air Cond Engr, Frigidaire Corp, 4436 Toulouse St, and (for mail), 1032 Broadway, New Orleans, La.

McLARNEY, Harry W. (*M* 1933), Air Cond. Engr, Union Electric Light & Power Co, 315 North 12th Blvd, St Louis, Mo

McLAUGHLIN, Joseph D. (*A* 1930, *J* 1928), Owner (for mail), Braley & McLaughlin, 166 Aborn St, and 45 Roslyn Ave, Providence, R. I.

McLEAN, Dermid (*M* 1917), McColl-Snyder & McLean, 2304 Penobscot Bldg, Detroit, Mich

McLEAN, James E. (*M* 1936), Supt, Buildings and Grounds Dept, University of Pittsburgh, and (for mail), 520 Bigham St, Mt Washington Station, Pittsburgh, Pa

McLEISH, William S. (*A* 1932, *J* 1928), Dist Engr (for mail), Ric-wiL Co, 101 Park Ave, New York, and 4116 47th Ave, Long Island City, N Y

McLENEGAN, David W ' (*M* 1933), Asst Engr, Air Cond Dept (for mail), General Electric Co, 5 Lawrence St, Bloomfield, and 73 Arlington Ave, Caldwell, N J

McLOUTH, Bruce F (*M* 1936, *J* 1931) Chief Engr, Heater Div (for mail), Dail Steel Products Co, E Main St, Lansing, and 135 Gunson, East Lansing, Mich

McMAHON, John E (*S* 1936), Safety Engr, Liberty Mutual Insurance Co, 230 E Ohio St, and (for mail), 7712 N Paulina St, Chicago, Ill

McMAHON, Thomas W. (*M* 1928), Dist Sales Mgr (for mail), American Blower Corp, 1715 Railway Exchange Bldg, and 6151 Waterman Ave, St Louis, Mo

McMUNN, A H. (*J* 1936, *S* 1934), (for mail), 4915 Forbes St, Pittsburgh, Pa, and 311 St Clair St, Clarksburg, W Va

McMURRER, Louis J. (*M* 1928, *A* 1928, *J* 1924), Pres, McMurrer Co, 303 Congress St, Boston, and (for mail), 190 Harvard Circle, Newtonville, Mass

McNAMARA, William (*A* 1930), Mgr (for mail), Trane Co, 2694 University Ave, and 1355 Como Ave W, St Paul, Minn

McPHERSON, William A (*M* 1929), Chief, Htg and Vtg Div, Dept of School Bldgs, 11 Beacon St, Boston, and (for mail), 86 Dwinnell St, West Roxbury, Mass

McQUAID, Daniel J. (*M* 1934), Owner (for mail), D J McQuaid Engineering Service, 614 Cooper Bldg, and 1565 Milwaukee St, Denver, Colo

McTERNAN, Felix J. (*A* 1931), 1523 Main St, Buffalo, N Y

MEAD, Edward A. (*M* 1926), Asst Sales Mgr (for mail), Nash Engineering Co, and 5 Thames St, Norwalk, Conn

MEAKIN, John B. (*J* 1935), Sales Engr, Heating Equipment, 17 Farnsworth St, Boston, Mass, and (for mail), 673 Chestnut St, Manchester, N H.

MEARS, Leon A. (*J* 1935), 721 Alice St, and (for mail), 4035 Greenwood Ave, Oakland, Calif

MEFFERT, George H. (*J* 1930), Engr (for mail), Carrier-Bock Corp, 2022 Bryan St, and 4154½ Prescott Ave, Dallas, Texas

MEHL, Oscar H. (*J* 1935), Engr (for mail), Carrier Corp, 2022 Bryan St, and 5002 Columbia Ave, Dallas, Texas

MEHNE, Carl A. (*M* 1929), Htg and Vtg Expert (for mail), 101 Park Ave (Room 821), New York, and 35 Livingston St, Valhalla, N Y

MEINHOLTZ, Herbert W. (*M* 1930), Branch Mgr (for mail), York Ice Machinery Corp, 603½ W Main St, and 3012½ Classen, Oklahoma City, Okla

MEINKE, Howard G (*M* 1933), Asst Engr (for mail), New York Edison Co, Inc, 4 Irving Place, New York, and 41 Harte St, Baldwin, L I., N Y

MEISEL, Carl L. (*A* 1936, *J* 1931), 240 West 98th St, Apt 8-D, New York, N Y

MELLON, James T J. (*M* 1911), Owner (for mail), Mellon Co, 4415-21 Ludlow St, and 431 North 63rd St, Philadelphia, Pa

MENDEN, Peter J. (*M* 1935), Secy, Thomas Heating Co, Inc, 1046 Herrick Ave, and (for mail), 1509 Arthur Ave, Racine, Wis

MENSING, Frederick D. (*M* 1920), Consulting Engr, Mensing & Co, 2845 Frankford Ave, Philadelphia, Pa

MERLE, André (*M* 1934), Chief Engr., Control Corp of America, Inc (for mail), 3752-85th St, Jackson Heights, L I, N Y

MERRILL, Carl J (*M* 1919), Treas, (for mail), C J Merrill, Inc, 54 St John St, and 15 Longfellow St, Portland, Maine.

MERRILL, Frank A (*M* 1931), Consulting Engr (for mail), Office of Hollis French, 210 South St, Boston, and 19 Auburndale Rd, Marblehead, Mass

MERRITT, C J (*M* 1925), Director, Merritt, Ltd, 8 French Bund, Shanghai, China

MERTZ, W A (*M* 1919), Secy (for mail), Kehm Bros Co, 51 E Grand Ave, and 3753 N Keeler Ave, Chicago, Ill

MERWIN, Gile E (*M* 1924, *J* 1923), Secy-Treas, Rockford Plumbing Supply Co, 700 S Main St, and (for mail), 1530 Myott Ave, Rockford, Ill

MESSENGER, Theodore I. (*A* 1936), Power Engr (for mail), Buffalo Niagara & Eastern Power Corp, 504 Electric Bldg, and 110 Norwood Ave, Buffalo, N Y

MEYER, Frank L. (*M* 1932, *J* 1928), Vice-Pres, Meyer Furnace Co, and (for mail), 9 Cole Court, Peoria, Ill

MEYER, Henry C, Jr ! (*M* 1898), (Council, 1915-1916), Pres (for mail), Meyer, Strong & Jones, Inc, 101 Park Ave, New York, N Y, and 25 Highland Ave, Montclair, N J

MEYER, John W. (*A* 1929), 5000 Pine St, Philadelphia, Pa

MICHIE, D Fraser (*A* 1930), Engrg Sales Dept, Crane, Ltd, 93 Lombard St, and (for mail), 5 B, 553 Wardlaw Ave, Winnipeg, Man, Canada

MIDDLETON, David K (*J* 1930), Branch Mgr, Johnson Service Co, and (for mail), 1100 Northwest 38th St, Oklahoma City, Okla

MIDDLETON, Howard A (*A* 1935), Prop, Middleton Electric Co, 514 S Ohio St, Sedalia, Ohio

MILENER, Eugene D (*M* 1936), Secy, Industrial Gas Section (for mail), American Gas Association, 420 Lexington Ave, Suite 550, New York, and 3719-83rd St, Jackson Heights, N Y

MILES, James C (*M* 1914), Dept Mgr. (for mail), Henry Furnace & Foundry Co, 3471 East 49th St, and 1863 Crawford Rd, Cleveland, Ohio

MILLARD, Junius W. (*M* 1929), Dist Mgr, Carrier Corp, Statler Bldg, Boston, Mass

MILLER, Bruce R. (*M* 1935; *A* 1930), Mech Engr, Leo Sanders Construction Co, Tradesman National Bank Bldg, and (for mail), 1533 Northwest 25th St, Oklahoma City, Okla.

MILLER, Charles A. (*A* 1917), Salesman (for mail), H B Smith Co, 10 East 41st St., and 2870 Marion Ave, New York, N Y

MILLER, Charles W. (*M* 1919, *J* 1908), Pres (for mail), Rado Co, 338 S Second St, Milwaukee, and R-1, Box 42, Menomonee Falls, Wis

MILLER, Edgar R (*A* 1935), Chief Engr (for mail), Winnipeg Cold Storage, Cor Jarvis and Salter and Ste O, Bexley Court, Winnipeg, Man, Canada

MILLER, Floyd A. (*M* 1911), Asst Dist Engr, U S Treasury Dept, Public Bldgs Branch, and (for mail), 377 U. S Court House, Chicago, Ill

MILLER, George F. (*M* 1936), Mgr Washington D C, Sales Office (for mail), C A Dunham Co, 1625 K St N W, Washington, D C, and 209 Connecticut Ave, Kensington, Md

MILLER, Harold A (*J* 1936; *S* 1935), 1009-24th Ave S E, Minneapolis, Minn.

MILLER, H. M (*M* 1920), 3038 N Stowell Ave, Milwaukee, Wis

MILLER, Jacob (*M* 1936), Pres (for mail), Universal Heating Co, Inc, 121 St Marks Place, New York, and 435 East 92nd St, Brooklyn, N. Y

MILLER, James E. (*M* 1914, *J* 1912), Heating Contractor, 2210 Colfax St, Evanston, Ill

MILLER, John F. G. (*M* 1916), Vice-Pres (for mail), B F Sturtevant Co, Damon St, Hyde Park, Boston, and 20 Chapel St, Brookline, Mass.

MILLER, Leo B. (*M* 1926), Mgr, Refrigeration and Air Cond. Div (for mail), Minneapolis-Honeywell Regulator Co, 801 Second Ave, New York, and 66 Charlotte Place, Hartsdale, N. Y.

MILLER, Lester L (*S* 1935), 623-14th Ave, Minneapolis, and (for mail), 2127 Tenth Ave, Hibbing, Minn

MILLER, Prof Lorin G. (*M* 1933), Head, Dept of Mech Engrg (for mail), Michigan State College, R E Olds Hall, and 525 Albert St, East Lansing, Mich

MILLER, Merl W (*M* 1932, *A* 1932, *J* 1926), Mgr of Lab (for mail), Trane Co, and 333 North 23rd St, LaCrosse, Wis

MILLER, Robert A† (*M* 1931), Tech Sales Engr (for mail), Pittsburgh Plate Glass Co, 2200 Grant Bldg, Pittsburgh, and 1211 Carlisle St, Tarentum, Pa

MILLER, Robert E (*J* 1935), Sales Engr (for mail), American Radiator Co, 1344 Broadway, and 18261 Birchcrest Drive, Detroit, Mich

MILLER, Robert T (*A* 1927), Chief Engr (for mail), Masonite Corp, 111 W Washington St, Chicago, and 1228 Sunnyside Ave, Chicago Heights, Ill

MILLER, Tolbert G (*A* 1929, *J* 1921), Supt and Engr, Herre Bros, Seventh and Emeralds St, Harrisburg, and (for mail), 11 N Second St, Wormleysburg, Pa

MILLIKEN, J H.† (*M* 1923), Dist Repr (for mail), American Air Filter Co, Inc, 20 N Wacker Drive, Chicago, and 1021 Ridge Court, Evanston, Ill

MILLIKEN, Vincent D (*A* 1930), Mfrs Service (Appointing Agents), (for mail), 6918 Normal Blvd., Chicago, Ill

MILLIS, Linn W. (*Life Member* 1931, *M* 1918), Secy, Security Stove & Mfg Co, 1630 Oakland, and (for mail), 3531 Wabash Ave, Kansas City, Mo

MILLS, Clarence A (*M* 1936), Prof of Experimental Medicine, University of Cincinnati (for mail), Cincinnati General Hospital, and 5016 Oberlin Blvd, Cincinnati, Ohio

MILLS, Hartzell C (*A* 1935), Salesman, Minneapolis Gas Light Co, 800 Hennepin Ave, and (for mail), 1609 Blaisdell Ave, Minneapolis, Minn.

MILWARD, Robert K (*A* 1920), Mgr (for mail), U S Radiator Corp, 127 Campbell Ave, and 2411 Calvert Ave, Detroit, Mich.

MINER, Major H. (*J* 1936, *S* 1934), 1510½ Northwest 25th St., Oklahoma City, Okla.

MINOR, Fred K (*A* 1936), Dist Repr, American Air Filter Co, Bulkley Bldg, Cleveland, Ohio.

MITCHELL, Charles H. (*M* 1924), Engr, The Fels Co, 12 Union St, and (for mail), 314 Spring St., Portland, Maine

MITCHELL, John G. (*S* 1936), 704 Delaware St S E, Minneapolis, Minn

MITTENDORFF, Edward M. (*M* 1932), Asst Engr, Sarco Co., Inc, 222 N Bank Drive, Chicago, and (for mail), 950 Greenwood Ave, Winnetka, Ill

MJOLSNES, Leonard O. (*J* 1936, *S* 1935), 618-15th Ave S E, Minneapolis, Minn

MODIANO, Rene (*M* 1925), Managing Dir, Carrier Continentale, 4, Rue d'Aguesseau, Paris (8*e*) and (for mail), 55 Boulevard Beausejour, Paris, (16*e*), France.

MOELLER, Robert (*S* 1935), 3256 East 119th St, Cleveland, Ohio

MOHRFELD, Herbert H. (*J* 1935), Air Cond Engr (for mail), Mohrfeld Engineering Co, Lees and Atlantic Aves, Collingswood, and B-102, Haddon Manor, Haddonfield, N J

MOLER, William H (*M* 1927; *J* 1923), Sales Engr. (for mail), York Ice Machinery Corp, 412 Houston St, and 1036 High Point Drive, Atlanta, Ga

MOLLENBERG, Harold J. (*M* 1936), Vice-Pres, Mollenberg-Betz Machinery Co., 22 Henry St., Buffalo, and (for mail), 172 Westgate Rd, Kenmore, N Y

MONICK, Fred R. (*A* 1936), Mgr (for mail), Cochran Sargent Co, 605 E Eighth St, and 1114 S Sixth Ave, Sioux Falls, S D

MONIER, Kurt A J. (*S* 1935), A J Monier & Co (for mail), 1146 N Flores, and 135 Ligustrum Drive, San Antonio, Texas

MONTGOMERY, Ora C. (*M* 1933), Asst Supt of Power (for mail), New York Central Railroad, Grand Central Terminal, Room 1842, 70 East 45th St, and 255 West 84th St, New York, N Y

MOODY, Lawrence E. (*M* 1919), Member of Firm (for mail), Moody & Hutchison, 1701 Architects Bldg, 17th and Sansom Sts, Philadelphia, Pa, and 237 Jefferson Ave, Haddonfield, N J

MOON, Frank L (*A* 1935), Utilization Engr, Los Angeles Gas & Electric Corp, 810 S Flower St, Los Angeles, and (for mail), 1264 Ruberta Ave, Glendale, Calif

MOON, L Walter (*M* 1915), Pres (for mail), Bradley Heating Co, 3834 Olive St, and 5006 N Kingshighway, St Louis, Mo

MOORE, C Herbert (*A* 1936, *J* 1935), 17th floor, 6 N Michigan Ave, Chicago, Ill

MOORE, Don R. (*S* 1936), (for mail), 153 Pierce St, West Lafayette, Ind, and 402 W Penn St, Hoopeston, Ill

MOORE, H. Carlton (*M* 1935), Instructor in Mech Engrg (for mail), Massachusetts Institute of Technology, Mech Engrg Dept, Cambridge, and 145 Beaumont Ave, Newtonville, Mass

MOORE, H. Lee (*M* 1910), Repr (for mail), Buffalo Forge Co, 431 Fulton Bldg, Pittsburgh, and Flaccus Rd, Ben Avon, Pittsburgh, Pa

MOORE, Henry W. (*M* 1935), Mgr, Air Cond. Dept, Smith Distributing Co, Inc, 831 E Broadway, Louisville, Ky, and (for mail), 816 Greenland Drive, Murfreesboro, Tenn

MOORE, Herbert S (*A* 1923), Dist Repr, Iron Fireman Mfg Co of Canada, Ltd, 602 King St W, and (for mail), 107 Clendenan Ave, Toronto, Ont, Canada

MOORE, Robert E (*J* 1933), Junior Sales Engr. (Div of), Manning, Maxwell & Moore, 446 Communipaw Ave, Jersey City, N J, and (for mail), 1730 East 46th St, Brooklyn, N Y

MOORE, R. Edwin (*A* 1928), Vice-Pres in charge of Sales, Bell & Gossett Co, 3000 Wallace St, Chicago, and (for mail), 714 Brummel St, Evanston, Ill

MORGAN, Glenn C (*M* 1911), Partner (for mail), Morgan-Gerrish Co, 307 Essex Bldg, 84 S Tenth St, and 4308 Fremont Ave S, Minneapolis, Minn.

MORGAN, Robert C. (*M* 1915), 314 W Seymour St, Philadelphia, Pa

MOREHOUSE, H. Preston (*M* 1933), Gen Air Cond Repr (for mail), Public Service Electric & Gas Co, 80 Park Place, Newark, and 85 Halsted St, East Orange, N J.

MORRIS, Arnold M. (*J* 1934), Sheet Metal Worker, Philadelphia Navy Yard, Sheet Metal Shop Bldg, No. 17, and (for mail), 3022 Baltz St, Philadelphia, Pa

MORRIS, Fred H. (*A* 1929), 14704 Stratmore Ave, East Cleveland, Ohio

MORRIS, John A. (*J* 1936), Htg Dept., James Robertson Co, Ltd, 946 William St, and (for mail), 4134 Marlowe Ave, Montreal, Que, Canada

MORRISON, Chester B (*M* 1931), Mgr. (for mail), York Shipley, Inc, 81 Jinkee Rd., and 347 Route Cohen, Shanghai, China.

MORSE, Clark T. (*M* 1913), Pres (for mail), American Blower Corp, 6000 Russell St, and 16222 Shaftsbury Rd, Detroit, Mich

MORSE, Floyd W. (*A* 1934), Asst Gen Sales Mgr (for mail), Chamberlin Metal Weather Strip Co., 52 Vanderbilt Ave, New York, and 112 Sycamore Ave, Mt Vernon, N Y

MORSE, Louis S., Jr. (*J* 1936), Air Cond Sales Engr. (for mail), Westerlin & Campbell Co, 1113-23 Cornelia Ave, Chicago, Ill, and 19710 Roslyn Rd, Detroit, Mich

MORSE, Robert D. (*M* 1936), Branch Mgr (for mail), American Blower Corp, 1534 First Ave. S, and 4316 East 43rd St, Seattle, Wash.

MORTON, Charles H (A 1931), 1106 Sherman St S E, Grand Rapids, Mich
MORTON, Harold S (M 1931), Sales Engr, Sutherland Air Conditioning Corp, 385 Minnesota St, St Paul, and (for mail), 4330 Wooddale Ave, Minneapolis, Minn
MOSES, Walter B, Jr (S 1936), 1920 Broadway, New Orleans, La
MOSHER, Clarence H (A 1919), C H Mosher Co, 423 Ashland Ave, Buffalo, N Y
MOSS, Edward (M 1920), Htg and Vtg Engr (for mail), New York Rapid Transit Corp, 385 Flatbush Ave Extension, Brooklyn, and 9053–204th St, Hollis, L I, N Y
MOTZ, O Wayne (M 1932), Consulting Engr, 232 Paramount Bldg, Cincinnati, and (for mail), 2587 Irving Place, Norwood, Ohio
MOULD, Delmar E (M 1936), Mgr (for mail), J W Mould & Son, 10708 Jasper Ave, and 10548–128th St, Edmonton, Alb, Canada
MOULDER, Albert W * (M 1917), Vice-Pres (for mail), Grinnell Co, Inc, 260 W Exchange St, and 12 Blackstone Blvd, Providence, R I
MOZLEY, Raymond G (J 1935), 737 Johnson Ave, Lakeland, Fla
MUELLER, Harold C (M 1936, A 1930), Sales Engr (for mail), Powers Regulator Co, 2720 Greenview Ave, Chicago, and 2720 Lawndale Ave, Evanston, Ill
MUELLER, Harold P (M 1936), Pres (for mail), L J Mueller Furnace Co, 2005 W Oklahoma Ave, and 4721 N Larkin St, Milwaukee, Wis
MULLEN, Thomas J., Jr. (J 1935), Sales Engr, B F Sturtevant Co, 814 Mills Bldg, 17th and Pennsylvania Ave, Washington, D C
MUNIER, Leon L (M 1919, J 1915), Pres-Treas (for mail), Wolff & Munier, Inc, 222 East 41st St, New York, and 63 Columbia Ave, Hartsdale, N Y
MUNN, E Fitz (M 1935), Designing (for mail), Over & Munn, 903 McArthur Bldg, and 65 Berrydale Ave, Winnipeg, Man, Canada
MUNRO, Edward A (*Charter Member, Life Member*), Htg-Vtg Engr, 49 N W 34th Terrace, Miami, Fla
MUNSON, James L (A 1935), 15 Parkwold Drive W, Valley Stream, N Y
MURDOCH, John P, Jr. (M 1937), Pres, John P Murdoch Co (for mail), 30th and Oakford Sts, and 5423 Cedar Ave, Philadelphia, Pa
MURPHREE, Robert L (J 1936), Engr (for mail), C P Lichty Engineering Co, Inc, 400½ South 21st St, and 1907–14th Ave N, Birmingham, Ala
MURPHY, Charles G (A 1936, S 1934), 2766 Woodhull Ave, New York, N Y
MURPHY, Edward T.* (M 1915), Vice-Pres (for mail), Carrier Corp, 180 N Michigan Ave, and Seneca Hotel, 200 E Chestnut St, Chicago, Ill
MURPHY, Howard C * (M 1923), Vice-Pres (for mail), American Air Filter Co, Inc, 215 Central Ave, and 495 Lightfoot Rd, Louisville, Ky
MURPHY, Joseph R. (M 1934, A 1925), Vice-Pres (for mail), Taco Heaters, Inc, 342 Madison Ave, New York, N Y, and The Terrace, Riverside, Conn
MURPHY, William W. (M 1930), Treas (for mail), W W Murphy Co, 424 Worthington St, and 25 Mansfield St, Springfield, Mass
MURRAY, Hayward G. S (J 1936), Sales Engr (for mail), Humidaire Co, Ltd, 620 Cathcart St, and Apt 14-A, 3721 De L'Oratoire, Montreal, Que, Canada
MURRAY, John J. (A 1933), Salesman-Vice-Pres, Pierce Perry Co, 236 Congress St, Boston, and (for mail), 60 Commonwealth Park West, Newton Centre, Mass
MURRAY, Thomas F. (M 1923), State Arch, and (for mail), 14 S Lake Ave, Albany, N Y
MUSGRAVE, Merrill N. (A 1935), Pres (for mail), Harrison Sales Co, 314 Ninth Ave N, and 1005 E Roy St, Seattle, Wash.
MYERS, Charles R (J 1936, S 1935), White Bear Lake, Route No 2, Minneapolis, Minn
MYERS, Frank L. (M 1933), Sales Engr, Owens-Illinois Glass Co, Ohio Bldg, and (for mail), 22 Proctor Place, Toledo, Ohio
MYERS, George W F (M 1930, A 1928, J 1923), Pres, Myers Engineering Equipment Co, 3917 W Pine Blvd, St Louis, and (for mail), 476 Pasadena Ave, Webster Groves, Mo
MYTINGER, Kenneth L (M 1936), Mgr, Air Cond Div (for mail), Fitzgibbons Boiler Co, 101 Park Ave, New York, N Y, and 119 E Bergen Place, Red Bank, N J

N

NAROWETZ, Louis L, Jr (M 1929, A 1912), Secy (for mail), Narowetz Heating & Ventilating Co, 1711-17 Maypole Ave, Chicago, and 112 Park Ave, Park Ridge, Ill
NASS, Arthur F. (M 1927), Secy-Treas (for mail), McGinness, Smith & McGinness Co, 527 First Ave, Pittsburgh, and Elmhurst Rd, R D No 8, Crafton P P, Pa
NATKIN, Benjamin* (M 1909, J 1907), Pres (for mail), Natkin & Co, 2020 Wyandotte, and 5211 Rockhill Rd, Kansas City, Mo
NEALE, Laurance I (A 1927), 125 East 57th St, New York, N Y
NEE, Raymond M (M 1936), Head of Engrg and Utilization Div (for mail), Edison Electric Illuminating Co, 39 Boylston St, Boston, and 10 Orkney Rd, Brookline, Mass
NEILER, Samuel G (*Life Member*, M 1898), Senior Member (for mail), Neiler, Rich & Co, Consulting Engrs, 431 S Dearborn St, Chicago, and 737 N Oak Park Ave, Oak Park, Ill
NEIS, Willard A (S 1935), 5538 Forbes St, Pittsburgh, Pa
NELSON, Arnold W (J 1930), Salesman, American Radiator Co, 1 Manor St, and (for mail), 11 S Lake Ave, Albany, N Y
NELSON, Arthur W (A 1936), Mgr, Brockton Oil Heat, Inc, 27 Legion Parkway, Brockton, and (for mail), 12 Sylvan Rd, Sharon, Mass
NELSON, Chester L (J 1929), Air Cond Engr, Sears & Roebuck, 305 Arsenal St, and (for mail), 4425 South 37th St, St Louis, Mo
NELSON, D W* (M 1928), Asst Prof of Steam & Gas Engrg (for mail), University of Wisconsin, Mech Engrg Bldg, and 3906 Council Crest, Madison, Wis
NELSON, Edwin L (A 1936), Engrg Dept (for mail), Union Ice Co, 1315 E Seventh St, and 4313 Victoria Ave, Los Angeles, Calif
NELSON, George O (M 1923), Engr, Carstens Bros, Ackley, Iowa
NELSON, Herman W (M 1909), Pres, Herman Nelson Corp, 1824 Third Ave, and (for mail), 2500–11th St, Moline, Ill
NELSON, Richard H (A 1933, J 1928), Secy-Treas, Herman Nelson Corp, 1824 Third Ave., and (for mail), 1303–30th St, Moline, Ill
NESBITT, Albert J.* (M 1921, J 1921), Secy-Treas (for mail), John J Nesbitt, Inc, State Rd and Rhawn St, Philadelphia, and 304 Evergreen Rd, Jenkintown, Pa
NESBITT, John J (M 1923), John J Nesbitt, Inc, State Rd and Rhawn St, Philadelphia, Pa.
NESMITH, Oliver E (A 1928), Htg Engr, Williams Oil-O-Matic Heating Corp, and (for mail), 107 Warner Ave, Bloomington, Ill
NESS, William H. C. (M 1931), Gen Mgr (for mail), Master Fan Corp, 1323 Channing St, and 215 N Kingsley Drive, Los Angeles, Calif.
NESSI, André (M 1930), Ingr des Arts et Mfrs., Expert pres le Tribunal Civil de la Seine (for mail), 1 Ave du President Wilson, Paris, XVI, France
NEST, Richard E (M 1936), Oil Burner Div, Anchor Post Fence Co, Baltimore, Md, and (for mail), 725 Taylor St N W, Washington, D. C
NEU, Henri J. E. (M 1933), Pres, Etablissements Neu, 47-49, Rue Fourier, Lille (Nord), France
NEWCOMB, Lionel B. (A 1936, J 1933), Junior Engr, Philadelphia Electric Co, and (for mail), 6056 Walton Ave, Philadelphia, Pa.

NEWPORT, Charles F ¹ (*M* 1906), Sales Engr, Weil-McLain Co, Michigan City, Ind, and (for mail), 10001 Longwood Drive, Chicago, Ill

NICELY, John E (*A* 1925), Salesman, American Radiator Co, and (for mail), 1208 Marion St, Reading, Pa

NICHOLLS, Percy ¹ (*M* 1920), Supervising Engr, Fuel Section, U S Bureau of Mines, Pittsburgh, Pa

NICKLE, Arthur J (*A* 1936), Sales Engr (for mail), Darling Brothers, Ltd, 140 Prince St, and 1356 Marcil Ave, Montreal, Que, Canada

NIGHTINGALE, George F (*A* 1931), Western Sales Mgr (for mail), Tuttle & Bailey, Inc, 61 W Kinzie St, Chicago, and 621 S Maple Ave, Oak Park, Ill

NOBBS, Walter W (*M* 1919), Consulting Engr, 100 Victoria St, London, S W 1, and (for mail), 50 Fairhazel Gardens, London, N W 6, England

NOBIS, Harry M. (*M* 1911), 1827 Stanwood Rd, East Cleveland, Ohio

NOBLE, James P. (*A* 1937), Dist Sales Engr, Refrigeration Equipment Co, 32 E First St, and (for mail), 238 Fountain Ave, Dayton, Ohio

NOLL, William F. (*M* 1924), Htg and Vtg Contractor (for mail), 629 North 27th St, and 2850 North 17th St., Milwaukee, Wis

NOORD, Donald F (*S* 1935), 518 Warren St, Albany, N Y, and (for mail), 5519 Forbes St, Pittsburgh, Pa

NORDINE, Louis F (*M* 1911), Branch Mgr (for mail), Trane Co, 726 Jackson Place N W, Washington, D C, and 812 Silver Springs Ave, Silver Springs, Md

NORTHON, Louis (*M* 1929), Consulting Engr, 132 Park Ave, Mt Vernon, N Y

NOTTBERG, Gustav (*A* 1933), Secy-Treas (for mail), U S Engineering Co, 914 Campbell St, and 1835 East 68th St Terrace, Kansas City, Mo

NOTTBERG, Henry (*M* 1919), Vice-Pres (for mail), U S Engineering Co, 914 Campbell St, and 213 S Bales, Kansas City, Mo

NOVOTNEY, Thomas A (*M* 1928), Gen Mgr, Convector Sales, National Radiator Corp, and (for mail), 103 Wayne St, Johnstown, Pa

NOWITZKY, Herman S. (*A* 1931), Supt, Construction, Maintenance and Repairs, Wilmer & Vincent Corp (Theatrical Chain), New York, N Y, and (for mail), 151 Tenth St, Norfolk, Va

NUSBAUM, Lee ¹ (*M* 1915), Owner (for mail), Pennsylvania Engineering Co, 1119-21 N Howard St, and 315 Carpenter Lane, Germantown, Philadelphia, Pa

NYE, L. Bert, Jr (*J* 1936), Htg Engr, Washington Gas Light Co, 411 Tenth St N W, and (for mail), 3002 32nd St N.W, Washington, D C

O

OAKEY, William E. (*M* 1932), Consulting Engr, Oriskany, N. Y

OAKS, Orion O. (*M* 1917), Executive Engr, American Radiator Co, 40 West 40th St, New York, N Y, and (for mail), 119 Oakridge Ave, Summit, N J.

OATES, Walter A. (*M* 1931), Htg and Industrial Engr, Lynn Gas & Electric Co, 90 Exchange St, and (for mail), 285 Lynn Shore Drive, Lynn, Mass

O'BANNON, Lester S * (*M* 1928), Prof of Heat-Power Engrg, Head-Dept of Mech Engrg (for mail), University of Kentucky, and 123 State St, Lexington, Ky

OBERG, Harry C. (*A* 1933), Mgr Engrg Dept, Crane Co of Minnesota, Fifth and Broadway, and (for mail), 1362 W Minnehaha St., St Paul, Minn.

OBERT, Casin W. * (*M* 1916), Consulting Engr, Union Carbide & Carbon Research Lab, 30 East 42nd St, New York, and (for mail), 122 N Columbus Ave, Mt Vernon, N Y

O'BRIEN, Walter N. (*A* 1935), Secy-Treas (for mail), O'Brien Equipment Co, 2726 Locust St, and 2221 Thurman Ave, St Louis, Mo

O'CONNELL, Presly M (*M* 1916), Resident Engr Inspector, Public Works Administration, Textile Tower, Seattle, and (for mail), R F D-Route No 1, Box No 5, Washougal, Wash

OFFEN, Ben (*M* 1928), Owner (for mail), B Offen & Co, 608 S Dearborn St, and 1100 N Dearborn St, Chicago, Ill

OFFNER, Alfred J ¹ (*M* 1922), (National Treas, 1935-1936, Council, 1935-1936), Consulting Engr (for mail), 139 East 53rd St, New York, and 160-15-11th Ave, Beechhurst, L I, N Y

OGARD, Norris L (*S* 1936), 224 Walnut St S E, Minneapolis, Minn

O'GORMAN, John S, Jr (*A* 1934), Mgr (for mail), Johnson Service Co, 427 Brainard St, Detroit, and 775 Kennesaw Rd, Birmingham, Mich

OKE, William C (*J* 1934), Air Cond Engr, Sheldons, Ltd (for mail), 1221 Bay St, and 110 Oriole Parkway, Apt 101, Toronto, Ont, Canada

OLCHOFF, Maurice (*M* 1933), Mgr, Olchoff Engineering Co, 423 Dwight Bldg, and (for mail), 5341 Holmes, Kansas City, Mo

OLDES, Willard E. (*J* 1930), Draftsman and Designer, Department of Purchase, 326 Broadway, and (for mail), 610 West 204th St, Apt D-3, New York, N Y

OLSEN, Carlton F (*A* 1925, *J* 1920), Combustion Engr, Kewanee Boiler Corp, 1858 S Western Ave, and (for mail), 7314 Stewart Ave, Chicago, Ill

OLSEN, Gustav E. (*M* 1930), 6809 Amstel Blvd, Arverne, L I, N Y

OLSON, Barney (*A* 1929), Mfrs Repr (for mail), 122 S Michigan Ave, and 5724 N Natoma Ave, Chicago, Ill

OLSON, Gilbert E (*M* 1930), 5004 Nicholas St, Omaha, Nebr

OLSON, Robert G (*M* 1923), Eastern Mgr (for mail), Hydraulic Coupling Div, American Blower Corp, 40 West 40th St, and 22 East 38th St, New York, N Y

OLVANY, William J. (*M* 1912), Pres (for mail), Wm J Olvany, Inc, 100 Charles St, New York, and 109-40-71st Rd, Forest Hills, N Y

O'NEIL, Joseph M (*A* 1934), Htg Contractor, 332 Commonwealth Ave, Springfield, Mass

O'NEILL, James W. (*M* 1929, *A* 1927, *J* 1925), Chief Engr, Trane Co of Canada, Ltd, 439 King St W, and (for mail), 8 Springmount Ave, Toronto, Ont, Canada

OPPERMAN, Everett F. (*J* 1935, *S* 1933), Estimator, Frederick Opperman, Railroad Ave, and (for mail), 169 Milbank Ave, Greenwich, Conn

OREAR, Andrew G. (*M* 1930), Sales Engr and Mfrs Repr (for mail), San Fernando Bldg, Room 501, and 406 S Main St, Los Angeles, Calif

O'REAR, Lawrence R. (*M* 1934), Pres (for mail), Midwest Plumbing & Heating Co, 2450 Blake St, and 3033 West 37th Ave, Denver, Colo

O'ROURKE, Hugh D, Jr. (*S* 1936), Sales Engr, Trane Co, 24 Commerce St, Newark, and (for mail), 793 Communipaw Ave, Jersey City, N J

ORR, George M. (*M* 1936), Pres (for mail), G M Orr & Co, Consulting Engrs, 542 Baker Arcade Bldg, and 2223 Emerson Ave N, Minneapolis, Minn

OSBORN, Wallace J. (*A* 1927), Vice-Pres, Keeney Publishing Co, Grand Central Term Bldg, New York, N Y, and (for mail), 599 Old Post Rd, Fairfield, Conn

OSBORNE, Gurdon H. (*M* 1922), Gen Mgr, Ventilating & Blow Pipe Co, Ltd, 714 St Maurice St, Montreal, and (for mail), 836 Pratt Ave, Outremont, Montreal, P Q, Canada

OSBORNE, Maurice M. (*M* 1925), 367 Beacon St, Boston, Mass

OSTERLE, William H. (*M* 1934), West Penn Power Co, and (for mail), 1028 Woodberry Rd, New Kensington, Pa

OSTRIN, Albert (*J* 1936; *S* 1935), 1216 James N, Minneapolis, Minn

OTIS, Gerald E * (*M* 1922), Vice-Pres (for mail), Herman Nelson Corp, and 1921–23rd Ave, Moline, Ill

OTT, Oran W. (*M* 1925), (Council, 1934-1936), Consulting Mech Engr (for mail), 606 Washington Bldg, and 123 S Virgil Ave, Los Angeles, Calif

OTT, Rush C. (*M* 1931), Sales Engr, Refrigerating Equipment Corp, 927-31 N Meridian St, Indianapolis, Ind

OURUSOFF, L. S * (*M* 1931), Engr of Utilization (for mail), Washington Gas Light Co, 411 Tenth St N W, Washington, D C, and 25 W Irving St, Chevy Chase, Md

OVERTON, Sidney H (*M* 1929), Repr, N V Radiatoren, Amsterdam, Holland, and (for mail), 205 Mackay Mansions, Rissik St, Johannesburg, South Africa.

P

PABST, Charles S. (*M* 1934), Pres and Mgr, Adams Engineering Co, Inc, 55 West 42nd St, New York, and (for mail), 8727–98th St, Woodhaven, L I, N Y

PAETZ, Herbert E. (*M* 1922), Div Sales Mgr (for mail), American Blower Corp, 2539 Woodward Ave, and The Wardell, Detroit, Mich

PAGE, Arvin (*M* 1935), Asst Chief Engr. (for mail), Bahnson Co, 1001 S Marshall St, and 600 Arbor Rd, Winston-Salem, N C

PAGE, Harry W. (*M* 1923), Pres (for mail), Wisconsin Equipment Co, 918 N. Fourth St, Milwaukee, and 7927 Warren Ave, Wauwatosa, Wis

PAGE, Vernon C. (*A* 1936), Director of Sales, Automatic Heating & Cooling Systems, 2101 N. Charles St., and (for mail), 5610 Greenspring Ave, Baltimore, Md

PALMER, Robert T (*A* 1935), Patent Lawyer (for mail), 49 Federal St, Boston, and 15 N Pleasant St, Sharon, Mass.

PAPPENFUS, Wilfrid G (*J* 1936, *S* 1935), Production Engr, Collins Radio, 621 First Ave S E, and (for mail), 1615 Second Ave S E, Cedar Rapids, Iowa

PAQUET, Jean-Marie (*J* 1936), Engr, J A Y Bouchard, Ltd., 9 Buade St, and (for mail), 62, De Salaberry, Quebec, Canada

PARK, Clifton D. (*M* 1929), 67 Woodlawn Ave, Needham, Mass

PARK, Harold E (*J* 1936), Salesman, Shaw-Perkins Mfg Co, and (for mail), 31 Vilsack St, Etna, Pa

PARK, J. Frank (*A* 1936; *J* 1930), Sales Engr (for mail), Western Air & Refrigeration, Inc, 1231 S Grand, and 726 N. Occidental, Los Angeles, Calif

PARK, Nicholas W. (*M* 1936), Htg Engr, Philadelphia Saving Fund Society (Real Estate Dept), 12 South 12th St, Room 309, Philadelphia, and (for mail), 509 Jericho Rd., Abington, Pa

PARKER, Philip (*M* 1915), 8 Middle St, Woburn, Mass

PARROTT, Lyle G. (*M* 1922), Consulting Engr, McColl, Snyder & McColl, 2308 Penobscot Bldg, and (for mail), 4678 Seebaldt Ave, Detroit, Mich

PARSONS, Leonard D., Jr. (*S* 1936), Stoker Engr, Fairbanks-Morse & Co, 900 S Wabash Ave, Chicago, and (for mail), 795 Park Blvd, Glen Ellyn, Ill

PARSONS, Roger A. (*J* 1933), Htg. Engr, Board of Water and Electric Light Commissioners, 215 E Ottawa St, and (for mail), 525 W Grand River Ave, Lansing, Mich.

PARTLAN, James W. (*Life Member, M* 1916), 14290 Goddard Ave, Detroit, Mich

PATERSON, Frederick C., Jr. (*M* 1936, *J* 1928), Pres (for mail), F C Paterson & Co, Inc, 76 Mechanic St, and 117 W Corydon St, Bradford, Pa

PATERSON, James S † (*M* 1922), Mech Engr. (for mail), Board of Education, 155 College St, and 23 Norton Ave, Toronto, Ont, Canada

PATORNO, Sullivan A S (*M* 1923), Consulting Engr (for mail), 101 Park Ave, and 312 East 163rd St, New York, N Y

PATRICK, Horace M (*M* 1936; *J* 1929), Secy, Chubbuck & Patrick, Inc, 1210 Sansom St, Philadelphia, and (for mail), 411 Pembroke Rd, Bala-Cynwyd, Pa

PATTERSON, Frank H. (*M* 1936), Salesman, Hoffman Specialty Co, and (for mail), 9201 Boleyn, Detroit, Mich

PAUL, Donald I. (*M* 1936, *J* 1932), Chief Engr (for mail), Gurney Foundry Co, Ltd., 4 Junction Rd, and 222 Fern Ave, Toronto, Ont, Canada.

PAUL, Lawrence O. (*J* 1935), Engr (for mail), Carrier Corp, 180 N Michigan Blvd, and 2501 Wilcox St, Chicago, Ill

PAULING, Robert E (*A* 1936), Salesman and Repr, U S Radiator Corp, Detroit, Mich, and (for mail), 211 S. Gary Ave, Tulsa, Okla

PAYNE, Robert E. (*M* 1935), Draftsman, E I DuPont de Nemours Co, Wilmington, Del, and (for mail), 244 Sedgewood Rd, Springfield, Pa

PEACOCK, James K. (*M* 1921), Asst. Secy, Hoffman Specialty Co, Inc, 500 Fifth Ave, New York, and (for mail), 410 Fowler Ave, Pelham Manor, N Y

PEEBLES, John K., Jr. (*A* 1925; *J* 1924), 7 Brandon Apt, University, Va

PELLER, Leonard (*J* 1934), Engr, Perfection Gear Co, Harvey, and (for mail), 1359 N Wells St, Chicago, Ill

PELLMOUNTER, Thomas (*A* 1930), Dist Sales Mgr, Century Electric Co, 903 McGee St, Room 512, and (for mail), 3308 Euclid Ave, Kansas City, Mo

PENNOCK, William B. (*M* 1927), Sales Engr, Pennock Engineering, 63 Sparks St, and (for mail), 326 Waverly St, Ottawa, Ont, Canada

PERINA, Arthur E (*J* 1936, *S* 1933), 126 Courtland St, Port Richmond, S I, N Y

PERKINS, Robert C. (*A* 1935), Dist Repr. (for mail), Ilg Electric Ventilating Co, 230 Sterick Bldg, and 1888 South Parkway, Memphis, Tenn

PERRAS, George E (*M* 1936), Htg Expert, Thomas Robertson & Co, Ltd, 262 Craig St. W, and (for mail), 6286 Chambord St., Montreal, Que, Canada

PESTERFIELD, Charles H. (*J* 1930; *S* 1932), Acting Head of Mech Engr Dept, University of North Dakota, University Station, Grand Forks, N D

PETERS, Herbert H. (*M* 1930), Mgr, H H Peters Heating Co, 1812 North 40th St., Milwaukee, Wis

PETERSEN, Stanley E. (*J* 1935), Sales Engr (for mail), W A Ramsay, Ltd, P O Box 1721, and 1310 Center St, Honolulu, Hawaii.

PETERSON, Carl M. F. (*M* 1930), Instructor (Mech Engrg), (for mail), Massachusetts Institute of Technology, 69 Massachusetts Ave, Cambridge, and 40 Fletcher Rd., Woburn, Mass.

PETERSON, Sterling D (*A* 1930), Branch Mgr. (for mail), Johnson Service Co, 473 Colman Bldg, and 5051 Prince St, Seattle, Wash.

PEXTON, Frank S. (*A* 1936), Sales Engr (for mail), Kansas City Gas Co, 824 Grand, and 43 West 73rd Terrace, Kansas City, Mo.

PFEIFER, Otto J., Jr. (*A* 1935, *J* 1932), Engr, Ralph D. Thomas & Associates, 1200 Second Ave S, and (for mail), 1515 Monroe St. N E, Minneapolis, Minn.

PFUHLER, John L. (*A* 1925, *J* 1923), Plumbing and Heating, 600 Manor Rd, Staten Island, N Y

PHILLIPS, Frederic W. (*M* 1921), Engr, House Htg Dept, Queensborough Gas & Electric Co., 1610 Far Rockaway Blvd, Far Rockaway, and (for mail), 825 East 38th St., Brooklyn, N Y.

PHILLIPS, Ralph E. (*M* 1936), Consulting Engr. (for mail), 816 W Fifth St, and 5153 Angeles Vista Blvd., Los Angeles, Calif

PHIPPS, Frederick G. (*M* 1930), Vice-Pres, Preston Phipps, Inc, 955 St James St. W, and (for mail), 5131 Earnscliffe Ave, Montreal, P Q, Canada

PIERCE, Edgar D. (*J* 1933), Sales Engr, Carrier Corp, 718 E Washington, and (for mail), 5738 S Grammercy Place, Los Angeles, Calif

PIETSCH, James A. (*M* 1936), Consulting Engr (for mail), Jas A Pietsch, Inc, 155 Prospect Ave, and 101 Cebia Ave, New Brighton, S I, N Y

PIHLMAN, Arthur A. (*M* 1928), Service Engr (for mail), Consolidated Gas Co of New York, 4 Irving Place, New York, N Y, and 98 Sherman Place, Jersey City, N J

PIKE, Wallace H. (*M* 1935), Engr Designer (for mail), Newcomb David Co, 5779-81 Russell St, and 4708 Buckingham Rd, Detroit, Mich

PILLEN, Harry A. (*A* 1933), Owner (for mail), H A Pillen Co (Mfg Agent), 622 Broadway, and 2124 Crane Ave, Cincinnati, Ohio

PINES, Sidney (*M* 1920), Vice-Pres (for mail), Natkin & Co., 1800 Baltimore Ave, and 6625 Brookside Rd., Kansas City, Mo.

PISTLER, Willard C. (*M* 1934), Mech Engr in charge of design, Carl J Kiefer, Consulting Engr, 918 Schmidt Bldg, and (for mail), Orchard Lane and Crestview Ave, Pleasant Ridge, Cincinnati, Ohio.

PITCHER, Lester J. (*M* 1929; *A* 1928, *J* 1924), Electrimatic Corp, 2100 Indiana Ave, and (for mail), 7921 S LaSalle St, Chicago, Ill.

PIZIE, Stuart G. (*A* 1926), Vice-Pres and Gen Mgr, Engineering Sales Co. (for mail), 50 Southwest 19th Rd., Miami, Fla

PLACE, Clyde R. (*M* 1924), Consulting Engr (for mail), 420 Lexington Ave., and 333 East 57th St, New York, N. Y

PLAENERT, Alfred B. (*A* 1933, *J* 1927), Resident Engr, Inspector P W. A (for mail), 1102 S Park St, Madison, Wis

PLAYFAIR, George A. (*A* 1924), Mgr (for mail), Johnson Temperature Regulating Co of Canada, Ltd, 97 Jarvis St, Toronto, and West Hill, Ont, Canada

PLEWES, Stanley E. (*M* 1917), Branch Mgr (for mail), Johnson Service Co, 2853 North 12th St, Station 8, Philadelphia, and 309 Evergreen Rd, Jenkintown, Pa.

PLUM, Leroy H. (*M* 1935; *A* 1934), Engr. (for mail), Warren Webster & Co., 17th and Federal Sts., Camden, and 207 Guilford Ave, Collingswood, N J.

PLUNKETT, John H. (*M* 1925), Retired, 81 Woodrow Ave, Boston, Mass.

POEHNER, Robert E. (*M* 1928), Htg Contractor, 849 Massachusetts Ave., Indianapolis, Ind.

POGALIES, Louis H. (*M* 1931), Mech Engr., Wilbur Watson & Associates, 4614 Prospect Ave., and (for mail), 4102 Archwood Ave, Cleveland, Ohio.

POHLE, Kenneth F. (*A* 1930), Vice-Pres, W F. Hirschman Co., Inc., 202 East 44th St, New York, N Y

POLDERMAN, Lambert H. (*M* 1927), Western Dist. Mgr. (for mail), Carrier Corp, 748 E. Washington Blvd, and 3462 Lambeth St., Los Angeles, Calif.

POLING, Dudley B. (*M* 1936), Chief Engr, A C Div (for mail), Lamneck Products, Inc., 416 Dublin Ave, and 797 E. Fulton St, Columbus, Ohio.

POLLARD, Alfred L. (*A* 1932), Gen Supt (for mail), Puget Sound Power & Light Co, 860 Stuart Bldg., and 3009-28th A., Seattle, Wash.

PONSELL, Francis I. (*A* 1935), Partner and Sales Engr. (for mail), James P. Ponsell & Sons, 826 Orange St., and 2708 Madison St., Wilmington, Del.

POPE, S. Austin (*M* 1917), Pres (for mail), William A. Pope Co., 26 N. Jefferson St., Chicago, and 831 Ashland Ave., River Forest, Ill

PORTER, Carl W. (*J* 1936), Engr (for mail), Henry Bartlett, 6 E Lucerne Circle, and 915 Bradshaw Terrace, Orlando, Fla

POSEY, James (*M* 1919), Consulting Engr (for mail), 1755 Baltimore Trust Bldg, and 4005 Liberty Heights Ave, Baltimore, Md

POUCHER, Richard C. (*S* 1935), American Blower Corp, 1433 Oliver Bldg, Pittsburgh, Pa.

POWELL, Knox A. (*A* 1936, *J* 1935; *S* 1933), Research Engr, Minneapolis-Moline Power Implement Co, 29th and Minnehaha Aves, and (for mail), 1008-18th Ave. S.E, Minneapolis, Minn

POWERS, Edgar C (*A* 1934, *J* 1931), Sales Engr (for mail), Blue Coal Corp, 1122 N. Charles St, and 5509 Wayne Ave, Baltimore, Md

POWERS, Fred W. (*M* 1911), Pres (for mail), Powers Regulator Co, 2720 Greenview Ave, and 900 Castlewood Terrace, Chicago, Ill

POWERS, Lowell G. (*J* 1930), Sales Engr (for mail), Carrier Corp, 1501 Carew Tower, and 291 Southern Ave, Cincinnati, Ohio

PRATT, Joseph C. (*A* 1936), Air Cond-Htg. Specialist, Canadian General Electric Co, Ltd., and (for mail), 1 Rosemount Ave, Westmount, P Q, Canada

PRAWL, Frank E. (*J* 1936), Asst. Branch Mgr, Sidles Co, Airtemp Div, 849 Stuart Bldg, and (for mail), 1641 South 26th St, Lincoln, Nebr.

PREECE, Leo W. (*A* 1936), Htg. and Air Cond. Engr, Winter Co, 1015 State St, and (for mail), R F D No 7, Erie, Pa.

PRENTICE, Oliver J. (*A* 1927), Publicity Mgr. (for mail), C. A Dunham Co, 450 E. Ohio St., and 850 Lake Shore Drive, Chicago, Ill.

PRESDEE, Cliff W. (*A* 1926), Mgr. S65 Div, S R. Dresser Mfg. Co., Bradford, Pa.

PRICE, Charles E. (*A* 1933), Treas (for mail), Keeney Publishing Co, 6 N. Michigan Ave., Chicago, and 1151 Chatfield Rd., Winnetka, Ill.

PRICE, Douglas O. (*M* 1934), Htg and Air Cond Engr, General Steel Wares, Ltd., 199 River St, and (for mail), 145 Eastbourne Ave., Toronto, Ont, Canada

PRICE, Ernest H. (*J* 1934; *S* 1932), Htg Engr. (for mail), Haff Supply Co, Inc., Burch Ave. and L. I R. R, Amityville, and 11 E. Second St., Riverhead, N. Y.

PRIESTER, Gayle B. (*J* 1935; *S* 1934), Air Cond. Engr. (for mail), Carrier Corp, 180 N Michigan Ave, and 7004 Oglesby Ave., Chicago, Ill

PRINCE, Raymond F. (*J* 1936), Engr., Order Dept, American Radiator Co, Boston, and (for mail), 18 Mellon St, Cambridge, Mass.

PROIE, John (*M* 1936), Pres (for mail), Proie Brothers Furnace Co., 856 W North Ave., and 101 Dilworth St, Pittsburgh, Pa.

PRUDDEN, Orrin D. (*S* 1936), 178 Locust St., Lockport, N Y, and (for mail), Purdue Union Club, West Lafayette, Ind.

PRUDEN, Bradlee (*M* 1936), Engr, Barber-Colman Co, 150 Loomis St., and (for mail), 1509 Grant Ave, Rockford, Ill

PRYIBIL, Paul L. (*A* 1932), Partner, Hucker-Pryibil Co, 1700 Walnut St, and (for mail), 328 E Philellena St, Philadelphia, Pa.

PRYOR, Frederick L. (*M* 1913), 5 Colt St., Paterson, N J.

PULLEN, Royal R. (*M* 1935; *A* 1935), Mech. Engr., 109 East Hill St, Lead, S. D.

PUNG, Donald W. (*J* 1936; *S* 1935), Sales Engr., Insulite Co, 1100 Builders Exchange, and (for mail), 315-19th Ave. S E., Minneapolis, Minn.

PURCELL, Frederick C. (*M* 1926), Dist. Mgr. (for mail), National Regulator Co, 2847 Grand River Ave, and 4711 Second Blvd., Detroit, Mich.

PURCELL, Robert E. (*M* 1916), Heating & Plumbing Contractor, 4061 Seebaldt Ave., Detroit, Mich.

PURDY, A. K. (*M* 1922), Pres. (for mail), Purdy Mansell, Ltd., 63 Albert St., and 30 Glenrose Ave, Toronto, Ont., Canada

PURDY, Randall B (*A* 1927), Associate Editor, Power (for mail), McGraw-Hill Publishing Co, 330 West 42nd St, New York, and 224-05–139th Ave, Laurelton, L I., N Y

PURINTON, Dexter J. (*A* 1923), Associate (for mail), Voorhees, Gmelin & Walker, Archts, 101 Park Ave, New York, N Y, and 23 Sachem Rd, Greenwich, Conn

PURSELL, H E (*M* 1919), Special Repr, Kewanee Boiler Corp, Kewanee, Ill

Q

QUAY, D M* (*Charter Member Life Member, Presidential Member*), (Pres, 1900, 1st Vice-Pres, 1896-1899, 2nd Vice-Pres, 1895), 725 Eastern Ave, Bellefontaine, Ohio

QUEER, Elmer R. (*M* 1933), Instructor in Engrg Research (for mail), Pennsylvania State College, Engrg Experiment Station, and 338 Arbor Way, State College, Pa

QUIGLEY, William J. (*M* 1920), 27 Knowlton Ave, Kenmore, N Y

QUIRK, Clinton H. (*M* 1916, *J* 1915), Eastern Repr (for mail), Trane Co, 250 East 43rd St, New York, and 465 Front St, Hempstead, N Y.

R

RACHAL, John M. (*A* 1936, *J* 1930), Air Cond Section, Andersen, Meyer & Co, Ltd, P O Box 265, Shanghai, China

RACK, Edgar C. (*M* 1931), Consulting Engr, Johns-Manville Corp, 22 East 40th St, New York, N Y, and (for mail), 288 Park Ave, East Orange, N J

RAINE, John J. (*M* 1912), Vice-Pres (for mail), G S Blodgett Co, 190 Bank St, Burlington, and Essex Juntion, Vt

RAINGER, Wallace F. (*A* 1930, *J* 1924), 441 Hawthorne Ave, Yonkers, N Y

RAISLER, Robert K. (*A* 1933, *J* 1930), Treas (for mail), Raisler Heating Co, 129 Amsterdam Ave, and 25 East 77th St, New York, N Y

RAMSAY, James W. (*A* 1936), Salesman (for mail), King & Shepherd, 50 Church St, New York, and 8415 Fourth Ave, Brooklyn, N Y

RANDALL, W. Clifton* (*M* 1928), Chief Engr (for mail), Detroit Steel Products Co, 2250 E Grand Blvd, and 5540 Ridgewood Ave, Detroit, Mich

RANDOLPH, Charles H. (*M* 1930, *A* 1928; *J* 1926), Air Cond Engr, Milwaukee Electric Railway & Light Co, 231 W Michigan St, and (for mail), 1925 N Prospect Ave, Milwaukee, Wis

RANK, Arthur I. (*A* 1936), Pres (for mail), Universal Insulation Co, 2013 Sansom St, and 6308 Ross St, Philadelphia, Pa

RANSOM, Clifford F. (*J* 1936; *S* 1935), 313 E Springfield, Champaign, Ill.

RASMUSSEN, Robert P. (*M* 1931), Pres (for mail), Economy Equipment Co, 538 W Pershing Rd, and 1243 East 46th St, Chicago, Ill

RATHBUN, Perry W. (*M* 1933), 1809 Northwest 37th St, Oklahoma City Okla

RATHER, Max F. (*M* 1919), Branch Mgr, Johnson Service Co, 2142 East 19th St, Cleveland, Ohio.

RAUH, Edward M. (*J* 1936; *S* 1934), (for mail), 205 E. Boyd, Norman and Alva, Okla

RAY, Lewis B. (*M* 1932), Mech Engr (for mail), Ray Engineering Co, Inc, 800 Broad St, Newark, and 151 Augusta St, Irvington, N J

RAYMER, William F., Jr. (*A* 1936, *J* 1934), Sales Engr (for mail), American Blower Corp, 249 High St, Newark, and 266 Fourth Ave, East Orange, N J

RAYMOND, Fred I* (*A* 1929), Owner (for mail), F I. Raymond Co, 629 W Washington Blvd, Chicago, and 547 N. Keystone Ave, River Forest, Ill

RAYNIS, Theodore (*J* 1934), Draftsman, Central Drafting Office (C & R), Navy Yard, Brooklyn, and (for mail), 8528–118th St, Richmond Hill, N Y

READ, Robert R (*J* 1936, *S* 1934), 1611 East 117th St, Cleveland, and (for mail), 2722 Owaisa Rd, Cuyahoga Falls, Ohio

RECK, William E. (*M* 1927), Civil Engr (for mail), Reck Heating Co, Ltd, Copenhagen N, Esromgade 15, and Sundvej 16, Hellerup, Denmark

REDRUP, Will D. (*M* 1936), Pres (for mail), Majestic Co, and 310 Randolph St, Huntington, Ind

REDSTONE, Arthur L (*M* 1931), Research Engr (for mail), Proctor & Schwartz, Seventh and Tabor Rd, and Park Towers, Kemble and Ogontz Ave, Philadelphia, Pa

REED, Irving G (*J* 1934), Asst Supt and Chief Engr, Grant Building, Inc, 420 Grant Bldg, and (for mail), 3227 Middletown Rd, Sheradan, Pittsburgh, Pa

REED, Paul L (*A* 1932), 1034 Art Hill Place, St Louis, Mo

REED, Van A., Jr. (*M* 1930), Mech Engr (for mail), Federal Engineering Co, 239 Fourth Ave, Pittsburgh, and 114 Water St, Elizabeth, Pa

REGER, Henry P. (*M* 1934), Pres-Treas (for mail), H P Reger & Co, 1501 East 72nd Place, and 6939 Bennett Ave, Chicago, Ill

REID, Henry P. (*M* 1931, *A* 1927), Special Engr (for mail), Universal Atlas Cement Co, 208 S LaSalle St, Chicago, and 3507 Oak Park Ave, Berwyn, Ill

REID, Herbert F (*A* 1932), Reid-Graff Plumbing Co, 1417 Peck St, Muskegon Heights, Mich

REIK, Robert C (*J* 1935), Engr, L E Stevens Co, 622 Broadway, Cincinnati, Ohio, and (for mail), 37 W Southgate, Ft Thomas, Ky

REILLY, Charles E. (*A* 1936, *J* 1928), 4920 City Line Ave, Philadelphia, Pa

REILLY, J. Harry (*M* 1931; *A* 1931, *J* 1929), Sales Engr, American Radiator Co, 528 Ferry St, Newark, and (for mail), 11 Watson Ave, East Orange, N J

REINKE, Alfred G. (*J* 1933), Group Leader, Westinghouse Electric & Mfg Co, 95 Orange St, Newark, and (for mail), 319 Park Place, Irvington, N J

RENOUF, E. Prince (*M* 1933), Air Cond Supv, Westinghouse Electric & Mfg Co, 1007 Insurance Bldg, and 3431 Rankin, Dallas, Texas

RENTE, Harry W (*M* 1931), Owner, Oil Burner Engr and Contractor, 114 Morris Ave, Buffalo, N Y

REPKO, Joseph J (*J* 1936; *S* 1934), 4021 Hamm Ave, Cleveland, Ohio

RETTEW, Harvey F. (*M* 1920), Chief Engr, Board of Education, 21st and Parkway, and (for mail), 6821 Martins Mill Rd, Philadelphia, Pa

REYNOLDS, Thurlow W. (*M* 1922), Consulting Engr, 100 Pinecrest Drive, Hastings-on-Hudson, N Y.

REYNOLDS, Walter V (*A* 1928), Pres, Walter Reynolds, Inc, 861 Third Ave, New York, N Y

RHEA, Chester A. (*A* 1931), Steel Boiler Repr, National Radiator Corp, 2124 Arch St, and (for mail), 722 Carpenter Lane, Philadelphia, Pa

RHOTON, Walter R. (*M* 1936), Pres (for mail), W R Rhoton Co, 1305 East 107th St, Cleveland, and 1728 Lee Rd, Cleveland Heights, Ohio

RICE, Clarence J. (*A* 1923), Pres (for mail), Sterling Engineering Co., 3738 N Holton St, and 2965 N Maryland Ave, Milwaukee, Wis

RICE, Robert B. (*M* 1934), Assoc Prof of Mech. Engrg., North Carolina State College of Agriculture and Engineering, State College Station, Raleigh, N C

RICHARD, Edwin J. (*M* 1933), Sole Owner (for mail), Edwin J. Richard Equipment Co, 528-29 Chamber of Commerce Bldg, and 3504 Paxton Rd, Cincinnati, Ohio

RICHARDSON, Henry G. (*M* 1934), Vice-Pres. and Secy, Hawley-Richardson-Williams Co, 204 Dooly Bldg, and (for mail), 1433 Harvard Ave, Salt Lake City, Utah

RICHMOND, John (*S* 1933), 5285 Forbes St, Pittsburgh, Pa, and (for mail), 951 Helmsdale Rd, Cleveland Heights, Ohio

RICHTMANN, William M * (*A* 1932, *J* 1926), Asst Prof of Engrg (for mail), Texas College of Arts and Industries, and 709 W Santa Gertrudes St, Kingsville, Texas

RICKNER, Charles A. (*M* 1935), Air Cond Engr, Kelvinator Corp, 14250 Plymouth Rd., and (for mail), 14327 Forrer, Detroit, Mich

RIES, Lester S. (*M* 1929), Asst Supt of Bldgs and Grounds (for mail), University of Chicago, 960 East 58th St, and 5611 Blackstone Ave, Chicago, Ill

RIESMEYER, Edward H., Jr. (*A* 1936, *J* 1930), Engr, Htg and Air Cond, Schaffer Heating Co, 231-33 Water St, and (for mail), 4702 Stanton Ave, Pittsburgh, Pa.

RIETZ, Elmer W. * (*M* 1923), Gen Sales Mgr (for mail), Powers Regulator Co, 2720 Greenview Ave, Chicago, and 2250 S Sheridan Rd, Highland Park, Ill

RILEY, Champlain L (*M* 1906), (*Presidential Member*), (Pres, 1921, 1st Vice-Pres, 1920, Council, 1918-1922), Vice-Pres and Treas (for mail), Clark, MacMullen & Riley, Inc, 101 Park Ave, New York, N Y, and Washington Rock Rd, Plainfield, N J

RILEY, Edward C * (*J* 1935, *S* 1933), Asst Public Health Engr (for mail), Industrial Hygiene Laboratory, U S Public Health Service, 25th and E Sts, and 1737 Q St, Washington, D C

RILEY, Robert C. (*J* 1936, *S* 1934), Testing Engr, Leviton Mfg Co, 236 Green Point Ave., Brooklyn, and (for mail), 8837–179th St., Jamaica, N Y

RINEHART, Wilson R. (*A* 1936; *J* 1932), Choudrant, La.

RITCHIE, A. Gordon (*M* 1933), Pres and Mgr. (for mail), John Ritchie, Ltd, 102 Adelaide St E, and 41 Garfield Ave, Toronto, Canada

RITCHIE, Edmund J. (*M* 1923), Vice-Pres, Sales, Sarco Co, Inc, 183 Madison Ave, New York, and (for mail), 2 Grace Court, Brooklyn, N Y.

RITCHIE, William (*M* 1909), 17 Van Reipen Ave, Jersey City, N J

RITT, Chester F. (*A* 1936), Sales Promotion Mgr (for mail), Trane Co, 726 Jackson Place N.W, and 2700 Q St N W., Washington, D C

RITTER, Arthur (*M* 1911), Dist Mgr (for mail), American Blower Corp., 40 West 40th St, New York, and 20 Edgemont Rd., Scarsdale, N Y.

RIVARD, Melvin M. (*M* 1935), Mgr, Rivard Sales Co, 225 Plaza Theatre Bldg, and (for mail), 1805 West 49th Terrace, Kansas City, Mo.

ROBB, Joseph E. (*A* 1936), Dist Mgr (for mail), Minneapolis-Honeywell Regulator Co, 215 Pershing Rd, and 4316 Rockhill Rd., Kansas City, Mo.

ROBERTS, Henry L. (*M* 1916), Engr and Contractor (for mail), 228 North 16th St, Philadelphia, and 1014 Allston Rd, Brookline, Del. Co, Pa.

ROBERTS, Henry P (*A* 1936), Asst. Engr (for mail), Roberts-Hamilton Co, 713 S Third St., and 1901 James Ave S., Minneapolis, Minn

ROBERTS, James R. (*J* 1934), Engrg Mgr. (for mail), Sutherland Air Conditioning Corp, 627 Marquette Ave, and 2423 Portland Ave., Minneapolis, Minn.

ROBERTSON, James A. M. (*A* 1936), Vice-Pres. (for mail), James Robertson Co, Ltd, 946 William St., Montreal, and 109 Sunnyside Ave, Westmount, Que, Canada

ROBINSON, Arthur S (*M* 1936), E I duPont de Nemours Co, Wilmington, Del, and (for mail), 730 Ogden Ave, Swarthmore, Pa

ROBINSON, Donald M. (*A* 1936), Sales Engr (for mail), Buffalo Forge Co, 820 Woodward Bldg, Washington, D C, and 16 Cedar St., Hyattsville, Md

ROBINSON, George L. (*A* 1935), Draftsman and Designer, E I duPont de Nemours (for mail), 210 West 28th St, Apt 1, Wilmington, Del

ROBINSON, Harry C. (*M* 1930), Htg Engr, 676 Pleasant St, Worcester, Mass

ROBINSON, Jack A. (*J* 1936), Air Cond. Engr (for mail), Australian Gas Light Co, Parker St., Sydney, and 10 Manson Rd, Strathfield, N S W., Australia.

ROCHE, Ivor F (*A* 1936), Mgr (for mail), Fess Oil Burners of Canada, Ltd, 1405 Drummond St, Montreal, Que, Canada

ROCKWELL, Theodore F. (*M* 1933; *J* 1932), Instructor in Htg and Vtg (for mail), Carnegie Institute of Technology, Pittsburgh, and 313 Sixth St, Aspinwall, Pa

RODEE, E John (*M* 1936), Engr (for mail), John B Pierce Foundation, 290 Congress Ave, New Haven, and 130 Bellevue Ave, West Haven, Conn

RODENHEISER, George B. (*M* 1933), Head Htg and Air Cond Dept (for mail), David Ranken, Jr, School of Mechanical Trades, 4431 Finney Ave, and 3639a Dover Place, St Louis, Mo

RODGERS, Frederick A. (*A* 1934), Branch Mgr, Minneapolis-Honeywell Regulator Co, 4501 Prospect Ave, Cleveland, and (for mail), 2637 Dartmoor Rd, Cleveland Heights, Ohio.

RODGERS, Joseph S. (*J* 1934), Engrg Draftsman, U S Government, Edgewood Arsenal, Edgewood, and (for mail), 1 Third Ave, Brooklyn Park, Md

RODGERS, William C (*J* 1936, *S* 1935), 400 Morewood Ave, Pittsburgh, Pa.

RODMAN, Robert W. (*M* 1922), Supt of Plant Operation (for mail), Board of Education, City of New York, 500 Park Ave, and 175 West 73rd St, New York, N. Y.

ROEBUCK, William, Jr. (*M* 1917), Mfrs Agent (for mail), 220 Delaware Ave, and 154 Sanders Rd, Buffalo, N Y.

ROHLIN, Karl W. (*M* 1930), Engr (for mail), Warren Webster & Co, 17th and Federal Sts, Camden, and 4453 Terrace Ave, Merchantville, N. J

ROLLAND, S. L. (*A* 1934), Design Engr (for mail), Oklahoma Gas & Electric Co, 321 N. Harvey Ave, and 2131 Northwest 20th St, Oklahoma City, Okla.

ROOS, Erik B. J. (*J* 1935), c/o B N Petigrow, P O Box 872, Tel-Aviv, Palestine

ROOT, Edwin B. (*M* 1936), Htg Engr, Nelson Co, 2604 Fourth Ave, Detroit, and (for mail), 964 Pierce St, Birmingham, Mich

ROSE, Arnold A. (*A* 1935), Sales Engr, Fitzgibbons Boiler Co, 185 Main St, and (for mail), Briar View Manor Apt, White Plains, N. Y

ROSE, Howard J. (*M* 1934), Sales Engr, Fitzgibbons Boiler Co, Inc, 185 Main St., White Plains, and (for mail), 100 Siebrecht Place, New Rochelle, N Y.

ROSEBROUGH, Robert M. (*M* 1920), Branch Mgr (for mail), L J Mueller Furnace Co., 4246 Forest Park Blvd, St Louis, and 204 S Maple Ave, Webster Groves, Mo.

ROSELL, Axel F. (*M* 1935), Mech. Engr, A. B Svenska Flaktfabriken, Kingsgatan 8, Stockholm, and (for mail), Kv Atlas 3, Lidingo I, Sweden.

ROSENBERG, Philip (*A* 1928), Secy.-Treas, Universal Fixture Corp, 137 West 23rd St, and (for mail), 250 West 104th St, New York, N. Y.

ROSENBURG, William E. (*J* 1935), Plbg.-Htg., John C. Rosenburg, Birch Hill Rd., Locust Valley, L. I., N. Y.

ROSS, John D. (*A* 1937), Sales (for mail), Railway and Engineering Specialties, Ltd., 637 Craig St. W, and 4376 Earnscliff Ave., Montreal, P Q., Canada.

ROSS, John O.* (*M* 1920), Pres , Ross Industries Corp , 350 Madison Ave , New York, N Y

ROSSITER, Paul A. (*J* 1936; *S* 1935), 136 S. Eighth St , Forest City, Iowa

ROTH, Charles F. (*A* 1930), Mgr , International Heating & Ventilating Exposition, Vice-Pres , International Exposition Co , Grand Central Palace, and (for mail), 141 East 36th St , New York, N Y.

ROTH, Harold R. (*M* 1935), Mgr Toronto Office (for mail), Canadian Sirocco Co , Ltd , 57 Bloor St W, and 18 Tichester Rd , Toronto, Ont , Canada

ROTHMANN, S. C. (*M* 1936), Industrial Hygiene Engr. (for mail), West Virginia State Health Dept , State Capitol Bldg , and 207 Bradford St , Charleston, W Va

ROTTMAYER, Samuel I. (*A* 1933, *J* 1928), Mech Engr , Libby, McNeill & Libby, U S Yards, and (for mail), 7861-B South Shore Drive, Chicago, Ill

ROWE, Irving E. (*A* 1936), Shop Foreman and Estimator, Etie Sheet Metal Works, 1416 Summer St , and (for mail), 3911 Searle Drive, Houston, Texas

ROWE, William A.* (*M* 1921), (Council, 1929-1931), Mech Engr , 718 Longfellow Ave , Detroit, Mich.

ROWE, William M (*J* 1936), Sales Engr (for mail), American Blower Corp , 1302 Swetland Bldg , and 11433 Mayfield Rd , Cleveland, Ohio

ROWLEY, Frank B.* (*M* 1918), (*Presidential Member*), (Pres , 1932, 1st Vice-Pres , 1931; 2nd Vice-Pres , 1930, Council, 1927-1933), Prof of Mech. Engrg and Director of Experimental Engrg Lab , University of Minnesota, and (for mail), 4801 E Lake Harriet Blvd , Minneapolis, Minn

ROYER, Earl B. (*M* 1928), Designing Engr , Fosdick & Hilmer, Consulting Engrs , 1703 Union Trust Bldg , and (for mail), 6635 Iris Ave , Cincinnati, Ohio

RUDIO, H. M. (*M* 1921), 126 Stolp Ave , Syracuse, N. Y

RUEGER, Charles A., Jr. (*A* 1935), Salesman (for mail), American Radiator Co , Box 505, and 926 W Fourth St , Winston-Salem, N C

RUFF, Adolph G. (*M* 1935), Supt of Power, U S Playing Card Co , Park Ave , Norwood, and (for mail), 3824 Woodford Rd , Cincinnati, Ohio.

RUFF, DeWitt C. (*M* 1922), Healy-Ruff Co., 765 Hampden Ave , St Paul, Minn

RUGART, Karl (*A* 1924), Branch Mgr , Warren Webster & Co , 26 South 20th St , and 5830 Willows Ave , Philadelphia, Pa

RUGGLES, Robert F. (*M* 1936; *A* 1927, *J* 1926), Dist. Mgr , Autovent Fan & Blower Co., 2 Rector St., New York, and (for mail), 15 Gregg Place, Randall Manor, S I., N Y

RUNKEL, Charles (*M* 1935), Pres. (for mail), Acme Heating & Ventilating Co , Inc , 4224 S Lowe Ave., and 7921 S Hermitage Ave , Chicago, Ill

RUPLE, Paul E. (*A* 1936), Mgr , Bradford Oil Burner Co , 61 Main St , Bradford, Pa.

RUSSELL, Edward A. (*M* 1936), Chief Engr , Vapor Car Heating Co , Inc , 1600 S. Kilbourn Ave , and (for mail), 8103 Dorchester Ave , Chicago, Ill

RUSSELL, J. Nelson (*M* 1899), Managing Dir (for mail), Rosser & Russell, Ltd , Romney House, Marsham St , Westminster, and Fernacres Fulmer near Slough, Buckinghamshire, England

RUSSELL, Wayne B. (*A* 1936), Htg Engr , Russell Furnace Co , 608 N. Monroe, and (for mail), 1203 S Cedar, Spokane, Wash

RUSSELL, William A. (*M* 1921), (Council, 1934-1936), Branch Mgr. (for mail), U S. Radiator Corp., 1221 West 11th St , and 628 West 57th Terrace, Kansas City, Mo.

RYAN, Harry J. (*M* 1922), Branch Mgr. (for mail), Trane Co , 47 Harris Ave , Albany, N. Y.

RYAN, James D (*M* 1935), Supt and Engr , Whitney National Bank, St Charles and Gravier St , and (for mail), 215 N Rendon St , New Orleans, La

RYAN, William F. (*J* 1933), Sales Engr , Lee Hardware Co , 250-52 N Santa Fe, and (for mail), 310 W Republic, Salina, Kan

RYDELL, Carl A. (*M* 1931; *J* 1928), Owner, C A Rydell Associates (for mail), 168 Dartmouth St , Boston, and 286 Quinobequin Rd , Waban, Mass

S

SABIN, Edward R. (*M* 1919), Pres , Edward R Sabin Co , 4710-12 Market St , Philadelphia, Pa

SADLER, C. Boone (*M* 1928), Associate Civil Engr (for mail), Public Works Office, 11th Naval District, and 4671 Newport Ave , San Diego, Calif.

SAITO, Shozo (*M* 1923), Saito Shozo Shoten, Ltd (for mail), Marunouchi Bldg , Tokyo, Japan

SAKOUTA, Mathieu L. (*M* 1924), Consulting Engr and Expert, Gavan, Simanskaia 4-A, Leningrad, U S S R

SALTER, Ernest H. (*M* 1936), Engr (for mail), Electrical Testing Laboratories, 80th St and East End Ave , New York, and 182 Cleveland Ave , Great Kills, S I , N Y

SALZER, Alfred R., Jr. (*S* 1936), Box 7255, Oakland Station, Pittsburgh, Pa , and (for mail), 2322 N Villere St , New Orleans, La

SANBERN, E. Nute* (*M* 1923), Engr , Hoffman Specialty Co , 500 Fifth Ave , New York, and (for mail), 317 Windsor Ave , Rockville Centre, L. I , N Y

SANDS, Clive C. (*M* 1929), G P O Box 601 F F , Sydney, N S.W , Australia

SANFORD, Arthur L (*M* 1915), Mech Engr , C H Johnston, Archt , 360 Robert St , St. Paul, and (for mail), 4240 Aldrich Ave S , Minneapolis, Minn

SANFORD, Sterling S* (*M* 1930), Sales Engr (for mail), Detroit Edison Co , 2000 Second Ave., and 1503 Seyburn Ave , Detroit, Mich.

SAPP, Charles L. (*A* 1936), Sales Mgr , Farquhar Furnace Co , and (for mail), 620 N. Walnut St , Wilmington, Ohio.

SAUER, Robert L. (*A* 1930), Sales Mgr (for mail), Riley Stoker Corp , Ft of Walker St , and 3315 W Philadelphia, Detroit, Mich.

SAUNDERS, Laurence P. (*M* 1933), Chief Engr , Harrison Radiator Corp , Lockport, N Y

SAWDON, Will M.* (*M* 1920), Prof , Experimental Engrg (for mail), Cornell University, College of Engrg., and 1018 E State St , Ithaca, N. Y

SAWHILL, R. V. (*A* 1929), Exec. Vice-Pres (for mail), Domestic Engineering Co , 110 East 42nd St , New York, and 115 Townsend Ave , Pelham Manor, N Y.

SAWYER, J. Neal (*J* 1933), Engr , Gustin-Bacon Mfg Co , 1412 West 12th St., Kansas City, Mo

SCANLON, Edward S. (*A* 1934), Utilization Engr , Equitable Gas Co , 427 Liberty Ave., and (for mail), 35 W. Francis Ave , Brentwood, Pittsburgh, Pa

SCARLETT, William J. (*M* 1936), Dist Repr. (for mail), Carrier Corp , 6017 Walnut St., Kansas City, Mo.

SCHECHTER, John P. (*J* 1935), Engr., House Htg Dept , Detroit City Gas Co , 415 Clifford, and (for mail), 1812 Burns Ave , Detroit, Mich.

SCHEIDECKER, Daniel B (*A* 1919), Secy (for mail), Hunter-Clark Ventilating System Co., 2800 Cottage Grove Ave , and 4626 N Kilbourn Ave , Chicago, Ill

SCHERMER, Richard (*S* 1936), 2324 West 111th St , Chicago, Ill

SCHERNBECK, Fred H. (*A* 1930), Salesman (for mail), William Bros Boiler & Mfg. Co , Nicollet Island, and 5045 Portland Ave., Minneapolis, Minn.

SCHERRER, Kenneth C. (*J* 1936), Engr, Natkin & Co, 2020 Wyandotte, and (for mail), 3911 Holmes St, Kansas City, Mo

SCHERRER, Leon B (*J* 1936), Engr, Housing Service, Inc, 5658 Delmar Ave, and (for mail), 6112 Simpson Terrace, St Louis, Mo

SCHICK, Karl W (*A* 1934), Dist Mgr, Minneapolis-Honeywell Regulator Co, 561 Reading Rd, Cincinnati, Ohio, and (for mail), 37 Audubon Place, Ft Thomas, Ky

SCHLICHTING, Walter G (*M* 1932), Mgr, Air Cond Dept, Clarage Fan Co, North and Porter Sts, and (for mail), 1117 W Lovell St, Kalamazoo, Mich

SCHMIDT, Richard H. (*J* 1936, *S* 1934), 2139 Abington Rd, Cleveland, Ohio

SCHMUTZ, Jean (*M* 1933), Administrateur-Delegue, Societe P R S M, 8, Passage de l'Atlas, and (for mail), 18, rue Dufrenoy, Paris 16e, France

SCHNEIDER, William G. (*M* 1932), Engr (for mail), American Brass Co, 25 Broadway, New York, and 45 Wayne Ave, White Plains, N Y

SCHNITZER, Sidney (*S* 1935), 2116 N. Spaulding Ave, Milwaukee, Wis

SCHOENIJAHN, Robert P (*M* 1919), Consulting Engr (for mail), 301-5 Industrial Trust Bldg, and 719 Nottingham Rd, Wilmington, Del

SCHOEPFLIN, Paul H (*M* 1920), Pres (for mail), Niagara Blower Co, 6 East 45th St, New York, and 91 Valley Rd, Larchmont, N Y

SCHUCANY, Oscar W. (*J* 1936, *S* 1935), 4927 Columbia Ave, Dallas, Texas

SCHUETZ, Clyde C (*A* 1936), Research Engr, U S Gypsum Co, 1253 Diversey Parkway, and (for mail), 3425 N Avers Ave, Chicago, Ill

SCHULTZ, Albert W (*M* 1936), Engr (for mail), Grinnell Co, Inc, 210 Seventh Ave S, and 5112 Penn Ave S, Minneapolis, Minn

SCHULZ, Howard I. (*A* 1915), Crane Co, 1223 W. Broad St., Richmond, Va

SCHULZE, Benedict H (*M* 1921), Eastern Sales Mgr (for mail), Kewanee Boiler Corp, 37 West 39th St, and 67 Park Ave, New York, N Y

SCHURMAN, John A., Jr. (*M* 1936, *J* 1935), Regional Air Cond Engr (for mail), York Ice Machinery Corp, 2700 Washington Ave N W, Cleveland, and 1029 Parkside Drive, Lakewood, Ohio

SCHWANTES, Arno R. (*J* 1935), Sales Engr, Waterman-Waterbury Co, 1121 Jackson N E, Minneapolis, Minn, and (for mail), 200 Washington St, Lake Mills, Wis.

SCHWARTZ, Harold (*S* 1936), 3450 Ainslie St., Chicago, Ill.

SCHWARTZ, Jacob (*A* 1936; *J* 1929), Contractor (for mail), Samuel Schwartz & Son, Inc, 30 West 27th St, Bayonne, and 12 Van Houten Ave, Jersey City, N J

SCHWARTZ, Simon B (*A* 1936), Consulting Engr (for mail), 1050 Constance St, and 2417 Broadway, New Orleans, La

SCHWEIM, Henry J. (*M* 1928), Secy and Staff Engr. (for mail), Gypsum Association, 211 W Wacker Drive, and 1637 Estes Ave, Chicago, Ill

SCOFIELD, Paul C. (*J* 1933), Engr. (for mail), Carrier Corp, 748 E. Washington Blvd, and 3879 Edenhurst Ave, Los Angeles, Calif

SCOTT, George M. (*M* 1915), Vice-Pres (for mail), Child & Scott-Donohue, Inc, 153 East 38th St, New York, and 654 King St., Port Chester, N Y

SCOTT, Lloyd G. (*J* 1936), Asst to Supt of Bldgs (for mail), Hudson's Bay Co, Hudson's Bay House, Main St, and Ste 42 Dalkeith Apts., 6 Balmoral Place, Winnipeg, Man, Canada.

SCRIBNER, Eugene D. (*A* 1933; *J* 1929), Purchasing Agt., Quinn Engineering Corp, 151 East 50th St., New York, N Y., and (for mail), 261 Clark St, Westfield, N J.

SCUDDER, Barrett (*A* 1935), Vice-Pres, James P Marsh Co., 2073 Southport Ave, Chicago, and (for mail), Washington Rd, Lake Forest, Ill

SEEBER, Rex R[k] (*M* 1934), Head-Mech Engrg. Dept, Michigan College of Mining and Technology, Houghton, Mich

SEELBACH, Herman (*M* 1931), Pres (for mail), Equipment Sales, Inc, 800 Erie County Bank Bldg, Buffalo, and 31 Central Ave, Hamburg, N Y

SEELERT, Edward H. (*A* 1935), Secy-Treas (for mail), McQuay, Inc, 1600 Broadway N E, and 2927 Ulysses St N E, Minneapolis, Minn

SEELEY, Lauren E.[k] (*M* 1930), Asst Prof Mech Engrg (for mail), Mason Laboratory, Yale University, and 130 Everit St, New Haven, Conn

SEELIG, Alfred E (*M* 1926), Pres and Gen Mgr, L J Wing Mfg Co, 154 West 14th St, and (for mail), 310 Convent Ave, New York, N Y

SEELIG, Lester (*M* 1925), Mech Engr, Museum of Science and Industry, Jackson Park, and (for mail), 725 Irving Park Blvd, Chicago, Ill

SEELY, Irving R. (*J* 1936, *S* 1935), General Electric Co, and (for mail), 1059 Wendell Ave, Schenectady, N Y

SEIDEL, Glenn E. (*S* 1936), Tulane University, and (for mail), 1437 Audubon St, New Orleans, La

SEITER, J. Earl[t] (*M* 1928), Asst Mgr, New Business Dept, Consolidated Gas Electric Light & Power Co, and (for mail), 7117 Bristol Rd, Baltimore, Md

SEKIDO, Kunisuke (*M* 1903), Consulting Engr, 685 Marunouchi Bldg, and (for mail), 19 Momozono, Nakano, Tokyo, Japan

SELIG, Ernest T, Jr (*M* 1936), Treas (for mail), Fuel Savers, Inc, 1501 Herr St., and 3520 Montour St, Harrisburg, Pa

SELLMAN, Nils T. (*M* 1922), Asst Vice-Pres and Asst. Secy (for mail), Consolidated Gas Co of New York, 4 Irving Place, New York, and 56 Walworth Ave, Scarsdale, N Y

SENIOR, Richard L. (*M* 1925), Pres (for mail), R L Senior, Inc, 103 Park Ave, New York, and 10 Cherry Ave, New Rochelle, N Y.

SENNET, Lowell E. (*J* 1936; *S* 1934), Engr, Central Bureau for Heating and Air Conditioning, 309 Euclid 30th Bldg, and (for mail), 1719 East 115th St, Cleveland, Ohio

SEVERNS, William H.* (*M* 1933), Prof. Mech Engrg. (for mail), University of Illinois, and 609 Indiana Ave., Urbana, Ill

SHAER, I. Ernest (*A* 1934), Sales Engr, B F Sturtevant Co, 89 Broad St., Boston, and (for mail), 35 Fessenden St, Dorchester, Mass

SHANKLIN, Arthur P. (*M* 1929), Sales Engr (for mail), Carrier Corp, 12 South 12th St, Philadelphia, and 40 Amherst Ave, Swarthmore, Pa.

SHANKLIN, John A. (*M* 1928), Secy-Treas (for mail), West Virginia Heating & Plumbing Co, 233 Hale St, and 1507 Quarrier St, Charleston, W Va

SHARP, Henry C. (*M* 1935), Mgr Oil Htg Div. (for mail), Smith Oil & Refining Co, and 1928 Rockton Ave, Rockford, Ill

SHAVER, Herbert H. (*A* 1929), Asst Gen Sales Agt. (for mail), Hudson Coal Co, 424 Wyoming Ave, and 1507 Wyoming Ave, Scranton, Pa

SHAW, Burton E.* (*A* 1936, *J* 1934), Research Chief (for mail), Penn Electric Switch Co, 2000 E Walnut St, Des Moines, and 505 Rapids, Adel, Iowa

SHAW, Charles G. (*A* 1936), Engr, Texas Air Conditioning Corp, 407 Capps Bldg, Fort Worth, Texas.

SHAW, Norman J. H. (*M* 1927; *J* 1925), Barnes & Jones, Inc., 128 Brookside Ave, Jamaica Plain, and (for mail), 37 Benjamin Rd, Arlington, Mass

SHAWLIN, Walter C. (*A* 1931), Mgr, Industrial Air Cond (for mail), Northwestern Ventilation Co, 2540 W. Wells St, and 2577 North 46th St., Milwaukee, Wis

SHEA, Michael B. (*M* 1921), Sales Dept (for mail), American Radiator Co, 1344 Broadway, Detroit, and 104 Massachusetts Ave, Highland Park, Mich.

SHEARS, Matthew W. (*M* 1922), Engr (for mail), C A Dunham Co, Ltd, 1523 Davenport Rd, and 39 Sylvan Ave, Toronto, Canada.

SHEFFLER, Morris (*M* 1921), Pres (for mail), Sheffler-Gross Co, 203 Drexel Bldg, and 5451 Lebanon Ave, Philadelphia, Pa

SHELDON, Nelson E. (*M* 1927), Dist Sales Mgr (for mail), Carrier Corp, 620 Reynolds Arcade Bldg, and 41 Lanark Crescent, Rochester, N Y

SHELDON, William D., Jr. (*A* 1936, *J* 1934), Chief Engr, Sheldon's, Ltd, and (for mail), Cedar St, Galt, Ont, Canada

SHELNEY, Thomas (*M* 1931), Pres. (for mail), Pierce Blower Corp, 100 Rhode Island St, Buffalo, and Grand Island, N Y

SHENK, Donald H. (*M* 1934), 106 Forest Lane, and (for mail), Riggs Hall, Clemson College, S. C

SHEPARD, Edward C. (*M* 1932), Mech Engr, Penberthy Injector Co, Detroit, Mich, and (for mail), 2294 University Ave, New York, N Y

SHEPARD, John deB. (*J* 1929), Air Cond. Repr (for mail), Consolidated Gas Electric Light & Power Co, 406 Lexington Bldg, and Tudor Arms Apts, W. University Parkway, Baltimore, Md

SHEPPARD, Frank A. (*M* 1918), Salesman (for mail), Johnson Service Co, 1031 Wyandotte St, and 27 East 70th St, Kansas City, Mo

SHEPPARD, William G. F. (*M* 1922), Partner, Sheppard & Abbott, 119 Harbord St, and (for mail), 1 Clarendon St, Toronto, Ont, Canada

SHERET, Andrew (*M* 1929, *A* 1925), Pres (for mail), Andrew Sheret, Ltd, 1114 Blanshard St, and 1030 St Charles St, Victoria, B C, Canada

SHERMAN, Ralph A. (*M* 1933), Fuel Engr, Head, Fuels Div (for mail), Battelle Memorial Institute, 505 King Ave, and 1893 Coventry Rd, Columbus, Ohio

SHERMAN, Victor L. (*M* 1935), Acting Head, Dept Mech Engrg, Lewis Institute, 1951 W Madison St, Chicago, and (for mail), 643 Hillside Ave, Glen Ellyn, Ill

SHIELDS, Carl D. (*S* 1936), Merchandising Engrg. Dept (for mail), Airtemp, Inc., 1119 Leo St., and 117 Monument Ave, Dayton, Ohio

SHILLING, Howard C. (*A* 1936), Salesman (for mail), Barber-Colman Co, 221 N LaSalle St, and 7068 N Paulina St, Chicago, Ill

SHIVERS, Paul F. (*M* 1930), Chief Engr, Research Div (for mail), Minneapolis-Honeywell Regulator Co, 115 Canal St, and 75 W Maple St, Wabash, Ind

SHODRON, John G. (*M* 1921), Consulting Engr and Research, 419 E Milwaukee Ave., Ft Atkinson, Wis

SHOEMAKER, Forrest F (*A* 1936), Pres and Mgr. (for mail), Air Conditioning Co, Inc, 222 Central Bank Bldg, and 2118 N Denver Blvd, Tulsa, Okla.

SHORB, Will A. (*M* 1909), Treas., Field & Shorb Co, 705 N Pine St, and (for mail), 3 Lincoln Place, Decatur, Ill.

SHOTWELL, Roger W. (*M* 1935), Designing Engr., W J. Spoelstra Co., Inc, 154 Parker Ave, Hawthorne, and (for mail), 97 Franklin St, Verona, N J

SHROCK, John H. (*M* 1924), Vice-Pres (for mail), New York Blower Co, 171 Factory St., and 1701 Indiana Ave, LaPorte, Ind.

SHULTZ, Earl (*A* 1919), Vice-Pres (for mail), Illinois Maintenance Co, 1136-72 W. Adams St., and Edgewater Beach Apts, Chicago, Ill.

SICKERT, Gene D. (*J* 1936), Mgr Htg Div, Perfex Radiator Co, and (for mail), 2178 N Sherman Blvd, Milwaukee, Wis

SIEBS, Claude T. (*A* 1927), Service Systems Engr. (for mail), Western Electric Co, Inc, 195 Broadway, New York, and Ward Estate, 636 Serpentine Rd, Grymes Hill, Staten Island, N. Y

SIGMUND, Ralph W (*M* 1932), Dist Mgr (for mail), B F Sturtevant Co, 913 Provident Bank Bldg, and 304 Oak St, Cincinnati, Ohio

SIMKIN, Milton (*J* 1936, *S* 1933), Assoc Engr, Charles Simkin, 103 Brighton Ave, Perth Amboy, N J.

SIMPSON, Arthur M * (*J* 1935), Chief Engr and Sales Mgr (for mail), Van Kannel Revolving Door Co, 101 Park Ave, New York, and 37-31-85th St, Jackson Heights, N Y

SIMPSON, Donald C (*M* 1932), Mgr, Product Analysis Dept (for mail), Owens-Illinois Glass Co, Case Ave, and 47 Day Ave, Newark, Ohio

SIMPSON, William K. (*M* 1919), Vice-Pres (for mail), Hoffman Specialty Co, 193 Grand St, and 9 Sands St, Waterbury, Conn

SKAGERBERG, Rutcher (*M* 1921, *J* 1921), Market Development Engr (for mail), Brown Instrument Co, Wayne and Roberts St, and 211 Rockglen Rd, Penn Wynne, Philadelphia, Pa

SKIDMORE, John G. (*J* 1930), Air Cond Engr, Carrier Corp, 408 Chrysler Bldg, New York, and (for mail), 5101-39th Ave, Long Island City, N Y

SKINNER, Henry W (*M* 1920), Consulting Engr (for mail), 4816 Dexter St, Fort Worth, Texas.

SKLAREVSKI, Rimma (*J* 1936), Sales Engr (for mail), American Radiator Co, 1512 Merchantile Trust, and 226 E University Parkway, Baltimore, Md

SKLENARIK, Louis (*J* 1928), 305 East 72nd St., New York, N Y

SLAYTER, Games (*M* 1931), Asst Gen Mgr, Industrial Materials Div, Owens-Illinois Glass Co, and (for mail), Box 726, Newark, Ohio

SLUSS, Alfred H. (*M* 1935), Prof Mech. and Industrial Engrg, University of Kansas, and (for mail), 827 Mississippi Ave, Lawrence, Kans

SMAK, Julius R. (*A* 1934), Supt of Service Depts, Crane Co, South Ave, and (for mail), 3135 Park Ave, Bridgeport, Conn

SMALL, Bartlett R. (*J* 1932), Sales Engr (for mail), T. C Heyward, 1408 Independence Bldg, and 326 W Tenth St, Charlotte, N C

SMITH, Elmer G.* (*M* 1929), Asst Prof of Physics (for mail), Agricultural and Mechanical College of Texas, College Station, Texas.

SMITH, Frank J. (*J* 1930), Engr, Northern Air Conditioning Corp, 382 Central Ave, Newark, and (for mail), 1 Fernwood Place, Upper Montclair, N J

SMITH, Gard W. (*M* 1927), Salesman, Premier Warm Air Heater Co, Dowagiac, Mich, and (for mail), 248 Roche St, Huntington, Ind

SMITH, Jared A. (*A* 1933), Distributor (for mail), Bryant Heater Co., 626 Broadway, Cincinnati, and 3817 Indian View Ave, Mariemont, Ohio.

SMITH, J. Darrell (*M* 1933), Mech. Engrg. Dept, Philadelphia & Reading Coal & Iron Co., and (for mail), 317 North 19th St., Pottsville, Pa

SMITH, Lloyd E. (*M* 1935), Mgr, Sales Engrg Dept, Frigidaire Corp., 300 Taylor St, and (for mail), 1327 Amherst Place, Dayton, Ohio.

SMITH, Milton S. (*M* 1919), Treas (for mail), Buensod Stacey Air Conditioning, Inc., 60 East 42nd St, New York, N. Y, and 13 N Terrace, Maplewood, N J

SMITH, Reginald J. (*M* 1936), Mgr., Smith & Elston, 71 Third Ave, and (for mail), P. O. Box 249, Timmins, Ont, Canada.

SMITH, Robert H (*J* 1934, *S* 1933), Field Repr., Plbg-Htg (for mail), Sears Roebuck & Co., 201 Brookline Ave, Dept. 8703-42, Boston, Mass, and 629 N. Grant St, West Lafayette, Ind.

SMITH, Stuart (*A* 1936), Branch Mgr. (for mail), American Radiator Co, 807 Times Star Bldg., and 1188 Herschel Ave, Cincinnati, Ohio.

SMITH, Wilbur F. (*M* 1920), Consulting Engr, W M. Anderson Co., 600 Schuylkill Ave., Philadelphia, and (for mail), 422 Bryn Mawr Ave, Cynwyd, Pa.

SMITH, William D (*A* 1935), Pres (for mail), W D Smith, Inc, 2147 Prospect Ave, and 17707 Winslow Rd, Shaker Heights, Cleveland, Ohio

SMOOT, Theo H. (*M* 1935), Chief Engr, Fluid Heat Div, Anchor Post Fence Co, Eastern Ave and Kane St, and (for mail), 2512 Talbot Rd, Baltimore, Md

SMYERS, Edward C (*A* 1933), Sales Engr, Minneapolis-Honeywell Regulator Co, 1013 Penn Ave, Wilkinsburg, and (for mail), 148 Jamaica Ave, West View, Pittsburgh, Pa

SNAVELY, A. Bowman (*M* 1937), Chief Engr, Hershey Chocolate Corp, Hershey, Pa

SNEED, Richard B (*J* 1936, *S* 1934), 505 West 11th St, Bristow, and (for mail), 320 South Blvd, Norman, Okla

SNELL, Ernest (*M* 1920), 3914 LeMay Ave, Detroit, Mich

SNIDER, Lewis A (*M* 1927), Pres (for mail), L A Snider Engineering Service, Inc, 605 N Michigan Ave, and 619 Buena Ave, Chicago, Ill

SNYDER, Allen K (*J* 1930), Engr in charge of Air Cond, Richmond Air Equipment Co, Inc, 1801 W Broad St, and (for mail), 4309 Grove Ave, Richmond, Va

SNYDER, Jay W. (*M* 1917), McColl-Snyder-McLean, 2301 Penobscot Bldg, Detroit, Mich

SNYDER, Joseph S. (*A* 1925), Sales Repr, Detroit Lubricator Co, 1807 Elmwood Ave, Buffalo, and (for mail), 9 Knowlton Ave, Kenmore (Buffalo), N Y

SODEMANN, Paul (*M* 1926, *J* 1920), Sales Engr, Sodemann Heat & Power Co, 2300 Delmar Blvd, and (for mail), 4136 Farlin Ave, St Louis, Mo

SODEMANN, William C. B. (*M* 1919), Pres (for mail), Sodemann Heat & Power Co, 2306 Delmar Blvd, St Louis, and 7542 Teasdale Ave, University City, Mo

SOLSTAD, Lester L. (*J* 1936), Como Station Route 5, St Paul, Minn

SOMMERFIELD, Sumner S. (*J* 1936), Design Engr, H A Phillips & Co, 155 N Union Ave, and (for mail), 5705 School St, Chicago, Ill

SONNEBORN, Charles (*M* 1930), Vice-Pres in charge of Production, Shaw-Perkins Mfg Co, West Pittsburgh, and (for mail), R D No 3, New Castle, Pa

SONNEY, Kermit J. (*J* 1936, *S* 1934), L B. 136, Wilcox, Pa, and (for mail), 316 W Symmes St, Norman, Okla

SOPER, Horace A. (*M* 1916), Vice-Pres (for mail), American Foundry & Furnace Co, and 1122 E Monroe St, Bloomington, Ill

SOULE, Lawrence C.* (*M* 1908), Secy and Consulting Engr (for mail), Aerofin Corp., 850 Frelinghuysen Ave, Newark, and Cor Stewart and Gordon Rds, Essex Fells, N J

SOUTHMAYD, Richard T. (*J* 1936), Salesman, American Blower Corp. (for mail), 11433 Mayfield Rd, Cleveland, Ohio

SPECKMAN, Charles H. (*M* 1918), Prof Engr., 375 Bourse Bldg, Philadelphia, Pa

SPELLER, Frank N* (*M* 1908), Director, Dept of Metallurgy and Research (for mail), National Tube Co, Frick Bldg, and 6411 Darlington Rd., Pittsburgh, Pa.

SPENCE, Morton R (*J* 1934), Asst. Purch. Agt, Rundle & Spence Mfg Co, 445 N Fourth St, and (for mail), 709 E Lexington Blvd, Milwaukee, Wis

SPENCE, Robert T. (*A* 1935), 1556 South 60th St, West Allis, Wis

SPENCER, J. Boyd (*M* 1935), Owner, Spencer Cooling & Air Conditioning, Co, 413 S. Sixth, Minneapolis, Minn

SPIELMANN, Gordon P. (*A* 1931, *J* 1923), Vice-Pres (for mail), Harrison-Spielmann Co, 408 Milwaukee Ave, Chicago, and 730 N Prospect Ave, Park Ridge, Ill

SPIELMANN, Harold J. (*M* 1933), Air Cond Engr, Vilter Mfg Co, 53 W Jackson Blvd, Chicago, and (for mail), 507 Elmore Ave, Park Ridge, Ill

SPITZLEY, Ray L. (*M* 1920), 1200 W Fort St, Detroit, Mich

SPOELSTRA, William J (*M* 1935), Pres, W J Spoelstra Co, Inc, 154 Parker Ave, Hawthorne, N J

SPOFFORTH, Walter (*M* 1930), Chief Mech Services, U S Penitentiary, McNeil Island, and (for mail), Route 5, Box 398, Day Island, Tacoma, Wash

SPROULL, Howard E (*M* 1920), Div Sales Mgr (for mail), American Blower Corp, 1005-6 American Bldg, and 3588 Raymar Drive, Cincinnati, Ohio

SPURGEON, Joseph H (*M* 1924), Salesman, Spurgeon Co (for mail), 5-203 General Motors Bldg, and 17215 Pennington Drive, Detroit, Mich

STACEY, Alfred E, Jr* (*M* 1914), Buensod-Stacey Air Conditioning, Inc, 60 East 42nd St., New York, N Y, and (for mail), Wootton Rd, Essex Fells, N J

STACK, Arthur E (*A* 1935), Lab Supv, Washington Gas Light Co, 411 Tenth St NW, Washington, D C, and (for mail), 911 Gist Ave, Silver Spring, Md

STACY, Lloyd D (*A* 1936), Sales Engr, Ilg Electric Ventilating Co, 182 N LaSalle St, and (for mail), 2247 Greenleaf Ave, Chicago, Ill

STACY, Stanley C. (*M* 1931), Mech Engr (for mail), Board of Education, 13 S Fitzhugh St, and 1224 Genesee St, Rochester, N Y

STALB, Joseph G. (*A* 1934), Promotion Sales Organizations, Reynolds Corp, 19 Rector St, New York, N Y, and (for mail), 22 Partridge Ave, Ridley Park, Pa

STAMMER, Edward L. (*M* 1919), Supt Htg and Vtg, Board of Education, Ninth and Locust Sts, and (for mail), 4430 Tennessee Ave, St Louis, Mo

STANGER, Ralph B. (*M* 1920), Owner (for mail), Robinson & Stanger, Empire Bldg, Pittsburgh, and Middle Rd, Glenshaw, Pa

STANGLAND, B. F. (*Charter Member*), (2nd Vice-Pres, 1908, Board of Governors, 1905-1906-1909; Board of Mgrs, 1895-1899, Council, 1896-1897), Retired, Kendall, N Y

STANNARD, James M.* (*Life Member*, *M* 1906), Pres and Treas (for mail), Stannard Power Equipment Co, 53 W. Jackson Blvd., Chicago, and 1402 Elinor Place, Evanston, Ill

STARK, W. Elliott* (*M* 1926), (Council, 1932-1936), Air Cond Sales Dept, Bryant Heater Co, 17825 St Clair Ave, Cleveland, and (for mail), 1875 Rosemont Rd, East Cleveland, Ohio

STEELE, John B. (*M* 1932), Chief Operating Engr, Winnipeg School Board, Ellen and William Ave, and (for mail), 184 Waterloo St, Riverheights, Winnipeg, Man, Canada

STEELE, Maurice G. (*M* 1929), Tech Advisor (for mail), Revere Copper & Brass, Inc, 1301 Wicomico St, and 4109 Roland Ave, Baltimore, Md

STEENECK, Kenneth C. (*J* 1935), Htg Engr, John G Kelly, Inc, 210 East 45th St, New York, and (for mail), 9410–211th St, Bellaire, L I., N Y

STEFFNER, Edward F. (*J* 1934), Htg and Air Cond Engr, Henry Furnace & Foundry Co, 3471 East 49th St, Cleveland, and (for mail), 1429 East 133rd St, East Cleveland, Ohio

STEGGALL, Howard B. (*A* 1934), Branch Mgr. (for mail), U S Radiator Corp, 941 Behan St. N S, and 1166 Murray Hill Ave, Pittsburgh, Pa.

STEHL, Howard V. (*A* 1936), Sales Engr. (for mail), Campbell Metal Window Corp, Bush and Hamburg Sts, Baltimore, and 5 Beacon Hill Rd, Woodlawn, Baltimore Co, Md

STEINHORST, Theodore F. (*M* 1919), Pres, Emil Steinhorst & Sons, Inc, 612-16 South St, and (for mail), 1664 Brickerhoff Ave, Utica, N Y.

STEINKELLNER, Edward J (*J* 1936; *S* 1935), 2162 South 32nd St, Milwaukee, Wis

STEINMETZ, C. W. Arthur (*M* 1934), Branch Mgr (for mail), American Blower Corp, 249 High St, Newark, and 50 Oakwood Ave, Bogota, N J

STENGEL, Frank J. (*A* 1935), Secy (for mail), R F Stengel & Son, 76 Rosehill Place, and 30 Elmwood Ave, Irvington, N J

STEPHENSON, Louis A. (*M* 1917), Branch Mgr (for mail), Powers Regulator Co, 409 East 13th St, and 801 West 57th St Terrace, Kansas City, Mo

STERLING, James G, Jr (*S* 1936), 1841 Wilton Rd, Cleveland Heights, Ohio

STERMER, Clarence J (*M* 1936), Engr, Crane Co, 836 S Michigan Ave, and (for mail), 7839 Clyde Ave, Chicago, Ill

STERNBERG, Edwin (*A* 1932, *J* 1931), Air Cond Engr, Armo Cooling & Ventilating Co, 30 West 15th St, and (for mail), 315 East 68th St., New York, N. Y

STERNE, Cecil M. (*A* 1934), Chief Engr (for mail), Metropolitan Refining Co, Inc, 23-28-50th Ave, Long Island City, and 115 Harold Rd, Woodmere, L I, N Y

STERNER, Douglas S (*A* 1936), Air Cond. Specialist (for mail), Tennessee Public Service Co, 626 S. Gay St, and 614 W Hill Ave, Knoxville, Tenn

STETSON, Lawrence R. (*M* 1913), Engr (for mail), McMurrer Co, 303 Congress St, and 35 Bradfield Ave, Roslindale, Boston, Mass

STEVENS, Harry L. (*M* 1934; *A* 1927, *J* 1924), Secy-Treas. (for mail), M M Stevens Co, 108 W Sherman St, and 7 West 22nd St, Hutchinson, Kans

STEVENS, Kenneth M. (*J* 1936), Sales Engr (for mail), Powers Regulator Co, 409 East 13th St, and 1326 E Armour, Kansas City, Mo

STEVENS, William R. (*A* 1934), Partner, L E Stevens Co, 626 Broadway, Cincinnati, Ohio, and (for mail), 159 Tremont Ave, Ft Thomas, Ky

STEVENSON, Melvin J. (*M* 1935), Mgr, Airtemp Div, Sidles Co, 849 Stuart Bldg, and (for mail), 1643 South 20th St, Lincoln, Nebr

STEVENSON, Wilbur W. (*M* 1928), Steam Htg Engr (for mail), Allegheny County Steam Heating Co, 435 Sixth Ave, and 1125 Lancaster Ave, Pittsburgh, Pa.

STEWART, Charles W (*M* 1919, *A* 1918), Asst Secy. (for mail), Hoffman Specialty Co, Waterbury National Bank Bldg, and 21 Yates Ave, Waterbury, Conn

STEWART, Duncan J. (*M* 1936, *A* 1930), Mgr., Electrical Div. (for mail), Barber-Colman Co, and R R. No 3, Rockford, Ill

STEWART, John C. (*A* 1934), Owner, Plbg and Htg (for mail), 1844 Smith St, and 2807 Victoria Ave, Regina, Sask, Canada

STILES, Gordon S. (*J* 1936), Sales Engr, Sidles Airtemp, 208 Davidson Bldg, Des Moines, and (for mail), 229 Stanton Ave, Ames, Iowa

STILL, Fred R.* (*M* 1904), (*Presidential Member*), (Pres, 1918; 2nd Vice-Pres, 1917, Council, 1916-1919), Vice-Pres (for mail), American Blower Corp., 40 West 40th St, and 1 East End Ave, New York, N Y

STILLER, Frederick W. (*J* 1933), Estimator (for mail), F C Stiller & Co, 129 S Tenth St, and 138 West 49th St, Minneapolis, Minn

STINARD, Rutherford L. (*J* 1934), Engr., American Radiator Co, 40 West 40th St, New York, N Y., and (for mail), 1377 Boulevard East, West New York, N. J.

STITT, Arthur B. (*J* 1935, *S* 1933), Mech Engr., F H McGraw & Co, 51 East 42nd St., New York, N Y, and (for mail), 30 Bundy Apts, Middletown, Ohio

STOCK, Charles S. (*M* 1936), Dist. Repr., Herman Nelson Corp, Room 404, 1108-16th St. N.W, Washington, D. C, and (for mail), 6752 Fairfax Rd, Bethesda, Md

STOCKWELL, William R (*M* 1903, *J* 1901), Gen Mgr, Mfg Div, Weil-McLain Co, Michigan City, Ind

STOKES, Alvin D. (*M* 1936), Erection Supt, Coordinator, York Ice Machinery Corp, 215 Investment Bldg, and (for mail), 3535-14th St N W, Washington, D C

STOKES, Arledge (*J* 1936), Branch Mgr (for mail), Sidles Co, Airtemp Div, 818 Stuart Bldg, and 1432 G St, Lincoln, Nebr

STONE, Eugene R (*M* 1913), 78 Woodbine St, Quincy, Mass

STONE, George F. (*Life Member*, *M* 1918), 1519 Cayuga St, Philadelphia, Pa

STORMS, Robert M. (*M* 1936), Consulting Engr, Storms & Gibbs, 604 Architects Bldg, Los Angeles, and (for mail), 354 W Wilson Ave, Glendale, Calif

STRAKOSH, Walter C. (*S* 1935), 911 S Fourth St, Champaign, Ill

STRAUCH, Paul C. (*A* 1934), Sales Engr, Henry Furnace & Foundry Co, 18th and Merriman Sts, and (for mail), Sherwood Hall, Cambridge Court Apts, Edgewood Boro, Pittsburgh, Pa

STRAVITSCH, Joseph J (*J* 1936), Estimator and Salesman, Standard Air, Inc, 10 West 40th St, New York, and (for mail), 9508 Linden Blvd, Ozone Park, L I., N Y

STREVELL, Roger P. (*M* 1934), Secy-Treas (for mail), William R Hogg Co, Inc, 900 Fourth Ave, Asbury Park, and State Highway and Victor Place, Neptune, N J

STRICKLAND, Albert W. (*A* 1929), Htg and Vtg Engr, Big Timber, Mont

STRINGFELLOW, Jack C. (*J* 1936; *S* 1935), A. M Lockett & Co, 305 Magnolia Bldg, Dallas, Texas

STROCK, Clifford (*A* 1929), Associate Editor (for mail), Heating & Ventilating, 148 Lafayette St, New York, and 82-15 Britton Ave, Elmhurst, L. I, N Y

STROUSE, Sidney B. (*M* 1921), Consulting Engr (for mail), S B & B H Strouse, 500-20 Guarantee Trust Bldg, and 22 S Illinois Ave., Atlantic City, N J

STRUNIN, Jay (*J* 1933), Engr, and Contractor (for mail), Strunin Plumbing & Heating Co, Inc, 408 Second Ave, and 54 West 89th St, New York, N Y.

STUART, Milton C.* (*M* 1935), Prof of Mech Engrg. (for mail), Lehigh University, Mech. Engrg Dept, and 505 Norway Place, Bethlehem, Pa

STUBBS, William C. (*M* 1934), Associate Naval Archt, U S Government (for mail), Hull Drawing Room, Norfolk Navy Yard, and 36 Channing Ave, Portsmouth, Va

STURM, William (*S* 1936), 511-12th Ave. E., Duluth, and (for mail), 315-16th Ave. S E., Minneapolis, Minn.

SUDDERTH, Leo, Jr. (*J* 1936), Branch Mgr., Johnson Service Co, 702 Bona Allen Bldg, and (for mail), 1425 Lanier Place N E., Atlanta, Ga.

SUMMERS, Ernest T. (*A* 1930), Pres. (for mail), Summers-Darling & Co, 121 Smith St., and Ste 22 Newcastle Apts., Winnipeg, Man, Canada

SUNDELL, Samuel S. (*J* 1935; *S* 1933), Engr, Larx Co, Inc, 607 S. Fifth Ave, and (for mail), 3040 Longfellow Ave., Minneapolis, Minn.

SUPPLE, Graeme B. (*M* 1934), Sales Engr. (for mail), American Blower Corp., 625 Architects and Builders Bldg., and 6224 Park Ave., Indianapolis, Ind.

SUTCLIFFE, Arthur G. (*M* 1922; *A* 1918), Chief Engr, Ilg Electric Ventilating Co, 2850 N. Crawford Ave, and (for mail), 4146 N. St. Louis Ave, Chicago, Ill

SUTHERLAND, David L. (*A* 1934), Pres and Treas (for mail), Sutherland Air Conditioning Corp, 627 Marquette Ave., and 1815 Colfax Ave. S., Minneapolis, Minn.

SUTTER, Edgar E (*A* 1936), Sales Engr , Mueller Brass Co , Port Huron, Mich , and (for mail), 6705 Sixth St N W , Washington, D C

SWANEY, Carroll R. (*M* 1929, *J* 1921), Sales Engr (for mail), Gilbert Howe Gleason & Co , 25 Huntington Ave , Boston, and 43 Clyde St , Newtonville, Mass

SWANSON, Earl C. (*A* 1935), Research Engr (for mail), Andersen Frame Corp , Bayport, and 616 W Mulberry St , Stillwater, Minn

SWANSON, Nils W. (*A* 1936), Sales Engr (for mail), McDonnell & Miller, 100 N Michigan Ave , Room 1316, and 7138 N Artesian Ave , Chicago, Ill

SWANSON, Rolf G. (*J* 1936; *S* 1935), 321 Walnut St S E , Minneapolis, Minn.

SWANSTROM, Alfred E (*J* 1935; *S* 1932), Construction Foreman, U S Dept of Interior, and (for mail), 1414 Van Buren St , St Paul, Minn

SWEATT, Charles H. (*J* 1936, *S* 1935), 4259 Unity Ave , Robbinsdale, Minn

SWEIVEN, C E. (*J* 1936, *S* 1935), 406 Walnut St S E , Minneapolis, Minn

SWENSON, John E (*A* 1930), Industrial Engr (for mail), Minneapolis Gas Light Co , 800 Hennepin Ave , and 1853 South 14th Ave , Minneapolis, Minn

SWIFT, Paul F. (*A* 1935), Secy , Carl F Scheffer Co , 838 S Main St , and (for mail), 1115 Old Orchard Ave , Dayton, Ohio

SWISHER, Stephen G., Jr. (*M* 1936, *A* 1934), Branch Mgr (for mail), Trane Co , 125 E Wells St , and 1711 E Dean Rd , Fox Point, Milwaukee, Wis

SYDOW, Louis J. (*M* 1936), Owner (for mail), Colonial Heating & Sheet Metal Co, 1423 Hodiamont Ave., and 3841 Maffit Ave , St. Louis, Mo

SYSKA, Adolph G. (*M* 1933), Consulting Engr , Syska & Hennessy, 420 Lexington Ave , New York, N Y.

SZEKELY, Ernest (*M* 1920), Vice-Pres and Gen. Mgr (for mail), Bayley Blower Co , 1817 South 66th St , and 3104 W. Kilbourn Ave , Milwaukee, Wis

SZOMBATHY, Louis R. (*A* 1930), Pres , Ferguson Sheet Metal Works, 34 N Florissant Blvd , Ferguson, Mo.

T

TABOR, Charles B. (*J* 1936, *S* 1935), 1815 University Ave. S E , Minneapolis, Minn.

TAGGART, Ralph C.* (*M* 1912), 14 Lyon Ave , Menands, Albany, N Y.

TALIAFERRO, Robert R.* (*M* 1919), Control Engr , Carrier Corp , 850 Frelinghuysen Ave , Newark, and (for mail), 133 Mountain Ave , Summit, N. J

TALLMADGE, Webster (*M* 1924), Pres. (for mail), Webster Tallmadge & Co, Inc , 255 North 18th St , East Orange, and 7 Claremont Place, Montclair, N. J.

TANKER, George E. (*S* 1936), Mech. Engr , Weatherhead Co , 620 Frankfort Ave , Cleveland, and (for mail), 1303 Virginia Ave , Lakewood, Ohio

TARR, Harold M. (*M* 1931), Htg and Vtg Engr , 21 Montague St , Arlington Heights, Mass

TASKER, C.* (*M* 1935), Research Fellow in Fuels (for mail), Ontario Research Foundation, 43 Queens Park, and 737 Avenue Rd , Toronto, Ont , Canada.

TAVERNA, Frederick F. (*M* 1928, *A* 1927, *J* 1924), Engr., Raisler Heating Co , 129 Amsterdam Ave., New York, N. Y , and (for mail), 106 12th St , Union City, N J.

TAYLOR, Edward M. (*A* 1934), Tech Mgr (for mail), Taylors, Ltd , 32-A Lichfield St , and Cr Kahu Rd , Totara St , Christchurch, New Zealand.

TAYLOR, Harold J. (*M* 1937), Htg -Vtg Contractor, 17514 Greenlawn Ave , Detroit, Mich

TAYLOR, John J. (*M* 1936), Partner (for mail), John Taylor & Son, 15 Hamilton St , and 37 Kay's Ave , Dulwich Hill, Sydney, N S W , Australia

TAYLOR, R. F (*M* 1915), Consulting Engr (for mail), 1305 Santa Fe Bldg , and 6144 Prospect, Dallas, Texas

TAYLOR, William E. (*A* 1934), Factory Sales Engr , Air Cond Div (for mail), Gar Wood Industries, Inc , 7924 Riopelle St , and 2008 W Grand Blvd , Detroit, Mich

TAZE, Donovan L (*M* 1931), Mgr , American Blower Corp , 1302 Swetland Bldg , Cleveland, Ohio.

TEAL, Edwin T. (*J* 1936, *S* 1935), Graduate Asst , Dept of Mech Engrg (for mail), Agricultural and Mechanical College of Texas, College Station, and 6416 Velasco, Dallas, Texas

TEASDALE, Lawrence A. (*M* 1926), Partner (for mail), Office of Hollis French, 20 Ashmun St., and 262 W Rock Ave., New Haven, Conn

TECKMYER, Fred C , Jr. (*S* 1936), 1515 Woodward Ave , Lakewood, Ohio

TEELING, George A. (*M* 1930), Consulting Engr , 11 N Pearl St., Albany, N Y

TEMPLE, Walter J. (*M* 1931), Engr , J A. Temple Co , 120 Red Arrow Parkway, and (for mail), 1215 Reed St , Kalamazoo, Mich

TEMPLIN, Charles L. (*M* 1921), (for mail), Carrier Corp , 404 Bona Allen Bldg., and 781 Sherwood Rd. N E , Atlanta, Ga

TENKONOHY, Rudolph J. (*M* 1923), Vice-Pres. (for mail), Airtherm Mfg Co , 1474 S Vandeventer Ave , and 3650 Shaw Blvd., St. Louis, Mo

TENNANT, Raymond J. J. (*A* 1929), (for mail), Pittsburgh Business Properties, Inc , Oliver Bldg , Room 2237, and 529 Navato Place, Pittsburgh, Pa

TENNEY, Dwight (*M* 1932), Pres. and Chief Engr (for mail), Tenney Engineering, Inc., Bloomfield Ave at Grove St , Bloomfield, and 83 Summit Rd , Verona, N J

TERHUNE, Ralph D. (*A* 1936), Repr , American Gas Products Corp., Fourth and Channing Sts N E., Washington, D. C., and (for mail), 4516 Highland Ave , Bethesda, Md

TERRY, Matson C. (*M* 1936), Chief Engr (for mail), Standard Air Conditioning, Inc , Second and Beechwood Sts , and 138 Calhoun Ave., New Rochelle, N. Y

THEORELL, Hugo G. T.* (*Life Member, M* 1902), Consulting Engr , Hugo Theorells Ingeniorsbyra, Skoldungagatan 4, Stockholm, Sweden

THETFORD, James E. (*J* 1936, *S* 1935), 411 E Healey St , Champaign, Ill.

THINN, Christian A.* (*M* 1921), Chief Engr , C A Dunham Co , 450 E. Ohio St , Chicago, Ill.

THOMAS, Glegge (*M* 1923), Branch Mgr. (for mail), Clarage Fan Co., 816 District National Bank Bldg , Washington, D C , and 4225 Lorcom Lane, Lee Heights, Arlington, Va.

THOMAS, L G. Lee (*M* 1934), Vice-Pres. (for mail), Economy Pumping Machinery Co , 3431 West 48th Place, Chicago, and 426 Forest Ave., Oak Park, Ill.

THOMAS, Melvern F. (*M* 1909), Consulting Engr (for mail), Thomas & Wardell, 863 Bay St., and 24 Rivercrest Rd , Toronto, Ont , Canada.

THOMAS, Norman A. (*M* 1928), Pres (for mail), Thomas Heating Co , 11th and Herrick Ave., and 824 Monroe Ave , Racine, Wis

THOMAS, Richard H. (*Life Member, M* 1920), Pres , Economy Pumping Machinery Co , 3431 West 48th Place, Chicago, Ill

THOMMEN, Adolph A. (*A* 1929), Metal Worker, John W Thomson, 1437 West 103rd St , and (for mail), 3400 West 61st Place, Chicago, Ill

THOMPSON, Donald (*J* 1932), Engrg Dept , Carbide & Carbon Chemicals Corp , and (for mail), 514 Simms St , Charleston, W. Va.

THOMPSON, Edwin F. (*A* 1935), Draftsman, E. I duPont de Nemours Co, Wilmington, Del, and (for mail), Atglen, Pa

THOMPSON, Frank (*M* 1935), Chief Engr, Vulcan Iron Works, Ltd, Sutherland Ave., and (for mail), 543 Newman St, Winnipeg, Man, Canada

THOMPSON, Nelson S.* (*M* 1917, *J* 1897), 1615 Hobart St N W, Washington, D C

THOMSON, Thomas N.* (*M* 1899), Consulting Engr, Plbg and Htg, 37 Irwin Place, Huntington, L I, N. Y

THORNBURG, Harold A. (*M* 1932, *J* 1929), c/o N V Industrieele Mi j Gebr Van Swaay, Societeitstraat 16, Soerabaia, Java

THORNTON, William B.[4] (*M* 1931), Sales Engr, Norge Nestor, Inc, 1024 W Adams St, and (for mail), 3329 Randall St, Jacksonville, Fla

THRUSH, Homer A. (*M* 1918), Pres, H A Thrush & Co, 21-23 Riverside Drive, Peru, Ind

THUNEY, Francis M. (*J* 1936), Application Engr (for mail), William E Kingswell, Inc (Minneapolis-Honeywell Regulator Co), 3707 Georgia Ave N W., and 2700 Que St N W, Washington, D C.

TIBBETS, John C. (*M* 1920), Engrg Dept, B & O R. R Co, and (for mail), P O Box 106, Ellicott City, Md

TILLER, Louin (*A* 1935; *S* 1933), Air Cond Engr, Oklahoma Gas & Electric Co, 321 N Harvey, and (for mail), 1724 Northwest 20th St, Oklahoma City, Okla

TILTZ, Bernard E. (*M* 1930), Pres (for mail), Tiltz Air Conditioning Corp, 230 Park Ave, New York, and 24 Barnum Rd, Larchmont, N Y

TIMMINS, W. W. (*M* 1937), Dist Mgr (for mail), Canadian Powers Regulator Co, Ltd, 314 University Tower Bldg, Montreal, and 351 Brock Ave. N, Montreal West, Que, Canada

TIMMIS, Pierce (*M* 1920), Service Equip Dept (for mail), United Engineers & Constructors, Inc, 1401 Arch St, Philadelphia, and 202 Midland Ave, Wayne, Pa

TIMMIS, W. Walter (*M* 1933, *A* 1925), Mgr, Product Development Sales Engrg (for mail), American Radiator Co, 40 West 40th St, New York, and 32 Oak Lane, Glen Cove, L I, N Y

TITUS, M. S. (*M* 1928), Carbide & Carbon Chemical Corp, and (for mail), 124 Rosemont Ave., South Charleston, W. Va

TJERSLAND, Alf (*M* 1916, *J* 1906), E Sunde & Co, Ltd, Oslo, Norway.

TOBIN, George J. (*M* 1905), Owner (for mail), 187-191 North Ave, and 510 Grant Ave, Plainfield, N J

TOBIN, John F. (*A* 1934), Salesman (for mail), American Blower Corp, 228 N LaSalle St, and 11256 S Artesian Ave, Chicago, Ill

TODD, Meryl L. (*J* 1936), Mech Engr (for mail), 901 Waterloo Bldg, and 1119 Vine St, Waterloo, Iowa

TODD, Stanton W., Jr. (*J* 1935), Sales Repr, American Radiator Co, 1344 Broadway, Detroit, and (for mail), 506 Lyon N E, Grand Rapids, Mich

TOLHURST, George C. (*M* 1936), Htg Expert, Gurney Massey Co, Ltd., Principal St, and (for mail), 142 Blvd St Germain, St. Laurent, near Montreal, Canada.

TONRY, Robert C. (*M* 1936), Mgr. (for mail), Wiedebusch Plumbing & Heating Co, 511 First St, and 507 First St, Fairmont, W Va

TOONDER, Clarence L. (*M* 1933), Air Cond. Engr, Sales Engrg Dept, Kelvinator Corp, Plymouth Rd., and (for mail), 13391 Marlow, Detroit, Mich

TORNQUIST, Earl L. (*A* 1934), Utilization Supv (for mail), Public Service Co of Northern Illinois, 72 W Adams St, Chicago, and 465 Parkside Ave, Elmhurst, Ill

TOROK, Elmer (*M* 1936), Supt of Power (for mail), North American Rayon Corp, Elizabethton, and 708 W Maple St., Johnson City, Tenn

TORR, Thomas W. (*M* 1933), Chief Engr, Rudy Furnace Co, and (for mail), P O Box 73, Dowagiac, Mich

TORRANCE, Henry (*M* 1933), Pres, Carbondale Machine Co, 175 Christopher St, and (for mail), 112 East 17th St, New York, N. Y

TOUTON, R. D. (*M* 1933), Tech Director (for mail), Bayuk Cigars, Inc, Ninth and Columbia Ave, Philadelphia, and 19 Lodges Lane, Cynwyd, Pa

TOWER, Elwood S. (*M* 1930), Engr (for mail), 2422 Koppers Bldg, and 5611 Forbes St, Pittsburgh, Pa

TRAMBAUER, Charles W. (*J* 1936), Sales Engr (for mail), Hoffman Specialty Co, 500 Fifth Ave, and 1051 University Ave, New York, N Y

TRANE, Reuben N* (*M* 1915), Pres (for mail), Trane Co, and 126 South 15th St, LaCrosse, Wis

TRAUGOTT, Mortimer (*A* 1930), East Sales Mgr (for mail), Bryant Heater & Mfg Co, 152 North 15th St, Philadelphia, and 721 Meeting House Rd, Elkins Park, Pa

TREADWAY, John Q (*A* 1936, *J* 1932), Sales Engr (for mail), Clarage Fan Co, 707 Security Bank Bldg, and 3143 Goddard Rd, Toledo, Ohio

TROSKE, Joseph J (*A* 1931), Vice-Pres and Gen Mgr (for mail), Vander-Troske Co, 236 Winter Ave N.W, and 233 Brown St, S.E, Grand Rapids, Mich

TROSTEL, Otto A. (*M* 1935), Engr (for mail), Kern Engineering Co, Inc, 5083 Plankinton Bldg, and 3155 N Seventh St, Milwaukee, Wis

TRUITT, G. Scott (*S* 1936), Sales Promotion Dept. (for mail), Bastian-Morley Co, Inc., La Porte, and Dana, Ind

TRUMBO, Silas M (*A* 1926), Sales (for mail), Buffalo Forge Co, 20 N Wacker Drive, Chicago, and 921 Franklin St, Downers Grove, Ill.

TRUMP, Charles C (*M* 1931), Pres (for mail), James Spear Stove & Heating Co, 1823 Market St, Philadelphia, and 503 Baird Rd, Merion Station, Pa.

TUCKER, Frank N. (*M* 1926), Field Engr, Ilg Electric Ventilating Co., Room 1108, 13 Park Row, New York, and (for mail), 239 Whaley St, Freeport, L I, N Y

TUCKER, Leonard A. (*M* 1935), Dist. Engr., J J Pocock, Inc, 1920 Chestnut St, Philadelphia, and (for mail), 518 Monroe Ave, Ardsley, Pa

TUCKER, Thomas T. (*A* 1936), Engr, Armor Insulating Co., C & S Bank Bldg, and (for mail), 3619 Ivey Rd N E, Atlanta, Ga.

TUCKERMAN, George E. (*M* 1932), Mgr, Air Cond. Dept. (for mail), York Ice Machinery Corp, 1238 North 41th St, Philadelphia, and 502 Rodman Ave, Jenkintown, Pa

TURLAND, Charles H. (*M* 1934; *A* 1930), Mgr, Htg and Vtg Dept., Kipp-Kelly, Ltd., 68 Higgins Ave., and (for mail), 325 Centennial St, Winnipeg, Man, Canada

TURNER, George G. (*A* 1931), Western Repr (for mail), Industrial Press, 228 N. LaSalle St., Chicago, and 714 Hinman Ave, Evanston, Ill

TURNER, Harry S., Jr. (*S* 1936), Dallas Power & Light Co, 1001 Dallas Power & Light Bldg., and (for mail), 226 N. Edgefield, Dallas, Texas

TURNER, John (*M* 1930), Sales Engr. (for mail), Minneapolis-Honeywell Regulator Co., 285 Columbus Ave., Boston, Mass, and Contoocook, New Hampshire.

TURNER, Prescott K. (*J* 1935), Head Engr, Crescent Auto Parts, 62 Webster St., and (for mail), 22 Maplewood Ave, Bradford, Pa.

TURNO, Walter G. W. (*M* 1917; *A* 1912), Secy., H W Porter & Co., Newark, and (for mail), 71 Lafayette Ave, East Orange, N. J.

TUSCH, Walter (*M* 1917), Htg and Vtg Engr., Tenney & Ohmes, Inc., 101 Park Ave., New York, and (for mail), 881 Sterling Place, Brooklyn, N. Y

TUTTLE, George H (*A* 1936; *J* 1934), Htg Engr. (for mail), Detroit Edison Co, 2000 Second Ave, and 11242 Rutherford, Detroit, Mich.

TUTTLE, J. Frank (*M* 1913), Sales Agt (for mail), Warren Webster & Co, 127 Federal St, Boston, and 9 Lewis Rd, Winchester, Mass

TUVE, George L † (*M* 1932), Prof of Heat-Power Engrg (for mail), Case School of Applied Science, and 1291 Cleveland Heights Blvd, Cleveland, Ohio

TUXHORN, David B (*M* 1936), Htg Engr, L P Steuart & Bro, Inc, 138-12th St N E, and (for mail), 1123-13th St N W, Washington, D. C

TWIST, Charles F. (*M* 1921), Pres (for mail), Ashwell-Twist Co, 967 Thomas St, and 2310 Tenth Ave N, Seattle, Wash

TYLER, Roy D. (*M* 1928), Mgr (for mail), Modine Mfg Co, 101 Park Ave, New York, and 15 Highbrook Ave, Pelham, N Y

TYSON, William H (*M* 1928), Mgr of Engrg (for mail), Goodyear Tyre & Rubber Co, Ltd, and "Kipewa" Codsall Rd, Nr Wolverhampton, England

U

UHL, Edwin J. (*M* 1925), Partner (for mail), Uhl Co, 132 S. Tenth St, and 4830 Pleasant Ave S., Minneapolis, Minn

UHL, Willard F. (*M* 1918), Partner (for mail), Uhl Co, 132 S Tenth St, and 4716 Lyndale Ave S, Minneapolis, Minn

UHLHORN, W. J. (*M* 1920), 733 S. Highland Ave, Oak Park, Ill.

ULLMAN, Herbert G. (*A* 1928), American Radiator Co, 40 West 40th St, New York, and (for mail), 107 White Rd, Scarsdale, N Y

URDAHL, Thomas H (*M* 1930), Consulting Engr (for mail), 726 Jackson Place, N W, and 1505 44th St N W, Washington, D C

V

VALE, Henry A L (*M* 1929), Managing Director (for mail), Vale & Co, Ltd, 141-13 Armagh St, and 211 Ilam Rd, Fendalton, Christchurch, New Zealand

VAN ALEN, Walter T. (*M* 1924), Htg Engr and Sales Agt (for mail), 1610 Seventh Ave, and 1300 Darlington Rd, Beaver Falls, Pa

VAN ALSBURG, Jerold H.* (*M* 1931), Engr of Application, Hart & Cooley Mfg. Co., 61 W. Kinzie St, Chicago, Ill

VANCE, Louis G. (*M* 1919), Partner Vance-McCrea Sales Co, 2700 Sisson St, and (for mail), 1402 Maine Ave, Baltimore, Md

VANDERHOOF, Austin L. (*A* 1933), Northern Ohio Dist Mgr. (for mail), Warren Webster & Co, 2311 Carnegie Ave., Cleveland, and 2762 Landon Rd, Shaker Heights, Ohio

VANDERLIP, P. J. (*A* 1935), Consulting Engr (for mail), Oil Heating Co, 206 S. Grand Ave, and 908 W. Shiawassee St, Lansing, Mich

VAN HORN, Howard T. (*A* 1933), Dist. Mgr, Detroit Stoker Co., 1217 McKnight Bldg, and (for mail), 4537 Grand Ave., Minneapolis, Minn

VAN NUYS, Jay C. (*S* 1936), 56 W Cliff St, Somerville, N. J.

VARNER, John L. (*A* 1935), Air Cond and Commercial Engr, Jacksonville Refrigeration, Inc, 35 W. Monroe St., and (for mail), Seminole Hotel, Jacksonville, Fla.

VAUGHAN, John G., Jr. (*J* 1935), Sales Engr, American Radiator Co., Fourth and Channing Sts N E., and (for mail), 3701 Massachusetts Ave., Washington, D C.

VAUGHN, Frank R (*A* 1936), Sales and Engrg. (for mail), Green Foundry & Furnace Works, and 682-38th St, Des Moines, Iowa

VELTMAN, Benjamin M. (*M* 1936), Htg Div (for mail), Brown Sheet Iron & Steel Co, 964 Berry St, and 1818 Goodrich Ave, St Paul, Minn

VERNON, J Rexford (*M* 1928, *A* 1926), (for mail), Johnson Service Co, 1355 Washington Blvd, Chicago, and 733 Brummel St, Evanston, Ill

VERVOORT, Edward L. (*S* 1936), House Htg. Engr, Brooklyn Union Gas Co, 180 Remsen St, Brooklyn, and (for mail), 31 Yale Place, Rockville Centre, L I, N Y

VETLESEN, G Unger (*M* 1930), 3 East 84th St New York, N Y

VIDALE, Richard (*M* 1935), Air Cond Engr (for mail), Quinby Air Conditioning Corp, 618 E Main St, and 572 Flower City Park, Rochester, N Y

VIESSMAN, Warren (*M* 1930), Mech Engr, Public Buildings Branch, Procurement Div, U S Treasury Dept, Washington, D C, and (for mail), 2205 Lake Ave, Baltimore, Md.

VINCENT, Paul J. (*M* 1931), Owner (for mail), P J Vincent Co, 2133 Maryland Ave, and 3807 Beech Ave, Baltimore, Md

VINSON, Neal L. (*J* 1936, *S* 1932), Engr, H P Reger & Co, 1501 East 72nd Place, and (for mail), 6724 Stony Island Ave, Chicago, Ill

VIVARTTAS, E. Arnold (*M* 1910), Consulting Engr, 247 Potter Ave, West New Brighton, S I, N Y

VOGEL, Andrew (*M* 1926), Engr, General Electric Co, 1 River Rd, and (for mail), 1821 Lenox Rd, Schenectady, N Y

VOGELBACH, Oscar (*M* 1923), Consulting Engr, Chamber of Commerce Bldg, Newark, and (for mail), 23 William St, North Arlington, N. J

VOGT, John H. (*A* 1925), Mech Engr (for mail), New York State Dept of Labor, 80 Centre St, New York, and 87 Grant Ave, Brooklyn, N Y

VOISINET, Walter E. (*M* 1930), Sales Repr (for mail), Buckeye Blower Co, 250 Delaware Ave, Buffalo, and 151 Warren Ave, Kenmore, N Y

VOLBERDING, Leroy A. (*A* 1936), Air Cond Engr, Norge Div of Borg-Warner, 670 E Woodbridge, Detroit, and (for mail), 1019 Knox Ave, Birmingham, Mich

VOLK, Joseph H. (*M* 1923), Pres and Treas (for mail), Thos E Hoye Heating Co, 1906 W St Paul Ave, and 2905 South 43rd St, Milwaukee, Wis

VOYE, Vernon J. (*A* 1936), Salesman, Richardson & Boynton Co, 17 Farnsworth St, Boston, and (for mail), 20 Richview St, Dorchester, Mass

VROOME, Albert E. (*M* 1932), Air Cond Engr, Phoenix Engineering Corp, 2 Rector St, New York, N Y., and (for mail), 412 Morton Ave, Rutledge, Pa.

W

WACHS, Louis J. (*A* 1936; *J* 1930), Salesman, Carrier Corp, Chrysler Bldg, New York, and (for mail), 354 East 21st St, Brooklyn, N. Y.

WAECHTER, Herman P (*A* 1930, *J* 1927), Air Cond Engr., W. T Grant Co, 1441 Broadway, New York, and (for mail), 89 Sherman Ave, Staten Island, N Y.

WAGNER, A. M. (*A* 1921), Branch Mgr. (for mail), American Radiator Co, 1741 W St Paul Ave, and 1725 E Kane Place, Milwaukee, Wis

WAGNER, Edward A. (*M* 1937), Pres, Wagner Engineering Corp, 22 Dunham St, and (for mail), 28 Waverly St, Pittsfield, Mass

WAGNER, Frederick H., Jr. (*M* 1934), Vice-Pres (for mail), Niagara Blower Co, 6 East 45th St, New York, and 1126 Post Rd, Scarsdale, N Y

WAHRENBROCK, Orin K. (*J* 1936), Engr, Automatic Appliance Corp, 36 Richmond Hill Ave, Stamford, and (for mail), 118 Lenox Ave, Glenbrook, Conn.

WAITE, Harry (A 1929), Secy-Treas (for mail), Gray Plumbing Co, 1214 Ogden Ave, and 1409–17th St, Superior, Wis

WALDON, Charles D (A 1932), Consulting Engr, Spencer Foundry Co, Penetang, and (for mail), 32 Ferndale Ave, Toronto, Ont, Canada

WALKER, Edmund R (*M* 1934), Sales Mgr (for mail), Fedders Mfg Co, Inc, 57 Tonawanda St, Buffalo, and 365 McKinley Ave, Kenmore, N Y

WALKER, James E. (*S* 1936), 214 Rockwood Ave, Dayton, and (for mail), 2139 Abington Rd, Cleveland, Ohio

WALKER, J. Herbert* (*M* 1916), Engr, Asst to Gen Mgr. (for mail), Detroit Edison Co, 2000 Second Ave, Detroit, and 432 Arlington Rd, Birmingham, Mich

WALKER, William K. (*M* 1935), Development Engr., American Radiator Co, 40 West 40th St, New York, N Y

WALLACE, George J (*M* 1923), Principal, Engr and Contractor, 96-19–35th Ave, Corona, and (for mail), 27-36 Ericsson St, East Elmhurst, N. Y

WALLACE, Harry P., Jr. (*A* 1936), Sales Mgr (for mail), Crane Co, 400 Third Ave N, and 5100–28th Ave. S, Minneapolis, Minn

WALLACE, James B (*A* 1935), Dist Repr, Taco Heaters, Inc, 342 Madison Ave, New York, N Y, and (for mail), 16921 Sorrento, Detroit, Mich

WALLACE, Kenneth S. (*M* 1931), Dist Engr. (for mail), Waterola Southwest Sales Corp., 316-18 S San Pedro St, and 1438 West 83rd St, Los Angeles, Calif

WALLACE, William M., II (*M* 1929), Mgr, Temperature Sales Co, 139 North Mill St, Lexington, Ky

WALSH, Edward R., Jr. (*M* 1936, *A* 1935), Sales Mgr, Automatic Heat & Air Cond, Herman Nelson Corp, Moline, Ill, and (for mail), Mississippi Hotel, Davenport, Iowa

WALSH, James A. (*A* 1932, *J* 1929), Air Conditioning Co, Main at Richmond St, Houston, Texas

WALSH, Malcolm (*M* 1924), Vice-Pres (for mail), Walsh & Wertheim, 504 W. Broadway, New York, and 91 Penbroke Ave, S I, N Y

WALTERS, Arthur L. (*M* 1926, *A* 1925, *J* 1924), Chief Engr. (for mail), Green Foundry & Furnace Works, Third and Elm Sts, and 900–29th St, Des Moines, Iowa

WALTERS, William T. (*M* 1917), Engr, Illinois Engrg Co., Cor 21st St and Racine Ave, and (for mail), 7965 Phillips Ave, Chicago, Ill.

WALTERTHUM, John J. (4 1922), Htg and Vtg Contractor, 212 East 58th St, New York, N. Y, and (for mail), 42-a Van Reipen Ave, Jersey City, N J

WALTON, Charles W., Jr. (*M* 1934), Mech Engr. (for mail), Rockefeller Center, Inc, 30 Rockefeller Plaza, New York, N Y, and 120 Monte Vista Ave, Ridgewood, N. J

WANDLESS, Franklin W. (*M* 1925), Registered Engr. (for mail), 1518 Fairmount Ave, Philadelphia, and Berwyn, Pa

WARD, Frank J. (*M* 1935), Owner, Frank J Ward Co., Cold Spring, Ky.

WARD, Jerry J. (*A* 1921), Pres (for mail), Wenzler & Ward, Inc, 1703 Textile Tower, and 1107–31st Ave, Seattle, Wash

WARD, Oscar G. (*M* 1919), Dist Repr (for mail), Johnson Service Co, 1230 California St, and 1607 Jasmine St., Denver, Colo

WARDELL, Arthur (*M* 1936), Partner (for mail), Thomas & Wardell, 863 Bay St, and 124 Melrose Ave, Toronto, Ont, Canada

WARE, John H., 3rd (*M* 1937), Utilities Contracting (for mail), 45 S. Third St., and "The Woods," Oxford, Pa

WARING, James M. S. (*M* 1932), Consulting Engr., 277 Park Ave, New York, N. Y

WARREN, Charles W (*J* 1936, *S* 1935), Mech Engr, A J Warren (for mail), 2313 Ave E, and 3317-R-½, Galveston, Texas

WARREN, Francis C (*M* 1934), Branch Mgr, American Blower Corp, 200 Division Ave N, Grand Rapids, Mich

WARREN, Harry L. (*M* 1930), Sales Research Engr, Southern California Gas Co, 950 S. Broadway, Los Angeles, and (for mail), 1303 Huntington Drive, South Pasadena, Calif

WASHBURN, Marcus J (A 1931), Insulation Engr (for mail), Eagle-Picher Lead Co, Temple Bar Bldg, and 2211 Park Ave, Cincinnati, Ohio

WASHINGTON, George (*M* 1931), Engr, Hoffman Specialty Co, Waterbury, Conn and (for mail), 4327 Johnson Ave, Western Springs, Ill

WASHINGTON, Laurence W. (*M* 1929), Vice-Pres (for mail), National Regulator Co, 2301 Knox Ave, Chicago, and 778 Laurel Ave, Des Plaines, Ill

WATERMAN, John H. (*M* 1931), Engr (for mail), Charles T Main, Inc, 201 Devonshire St, Boston, and 7 Centre St, Cambridge, Mass

WATERS, Frank A. (*A* 1936), Sales Engr, Automatic Appliance Corp, 36 Richmond Hill Ave, Stamford, Conn, and (for mail), Bedford Hills, N Y

WATERS, George G. (*M* 1931, *A* 1926), Dist Mgr (for mail), American Blower Corp, 1433 Oliver Bldg, and 52 Vernon Drive, Pittsburgh (16), Pa

WATKINS, George B. (*A* 1936), Director of Research (for mail), Libby-Owens-Ford Glass Co, Research Dept, Oakdale and E Broadway, and 3004 Berdan Ave., Toledo, Ohio

WATSON, H. Dalton (*A* 1935), Branch Mgr. (for mail), Linde Canadian Refrigeration Co, 121 King St, and 61 Furby St, Winnipeg, Man, Canada

WATSON, M. Barry (*M* 1928), Consulting Engr, 121 Welland Ave, Toronto 5, Canada

WAUNG, Tsing-fi (*M* 1935, *J* 1933), Htg. Engr. (for mail), Andersen Meyer & Co, Ltd., Yuen Ming Yuen Rd, and 16 Lane, 152 Edinburgh Rd, Shanghai, China

WEATHERBY, Edward P., Jr (*J* 1936, *S* 1935), 813 N Zangs Blvd, Dallas, Texas.

WEATHERLOW, Guy P. (*M* 1936), Htg. Engr, E I. duPont de Nemours & Co, and (for mail), 207 Beaston Ave, Hillcrest, Wilmington, Del.

WEBB, Ernest C. (*M* 1935), Engrg Service Mgr. (for mail), Iron Fireman Mfg. Co., 3170 West 106th St, Cleveland, and 1202 Woodside Drive, Rocky River, Ohio

WEBB, John S. (*M* 1920), Sales Engr, W D. Cashin Co, 69 A St, Boston, and (for mail), 345 Brookline St, Needham, Mass

WEBB, John W. (*M* 1926), Managing Director (for mail), Webb Dust Removing & Drying Co, Ltd., Vinery Works, Town Lane, Denton, and "Ebor" Brinnington, Stockport, England

WEBER, Erwin L. (*M* 1921), Consulting Engr, 534 Medical Arts Bldg, Seattle, Wash.

WEBSTER, E. Kessler (*M* 1915), Warren Webster & Co, 17th and Federal Sts, Camden, N J.

WEBSTER, Warren (*Life Member*, *M* 1906, *A* 1899), Pres, Warren Webster & Co., 17th and Federal Sts, Camden, N. J.

WEBSTER, Warren, Jr. (*M* 1932; *J* 1927), Vice-Pres. and Treas (for mail); Warren Webster & Co, 1725 Federal St, Camden and Washington Ave, and Colonial Ridge Drive, Haddonfield, N. J

WEBSTER, William H., Jr. (*A* 1935), Vice-Pres. (for mail), Hurst Heating Engineers, Inc, 400 York St., and 200 N Shore Rd, (Academy Terrace), Norfolk, Va

WECHSBERG, Otto (*M* 1932), Pres and Gen Mgr, Coppus Engineering Corp, 344 Park Ave., and (for mail), 1006 Main St, Apt. 4, Worcester, Mass.

WEDDELL, George O (*M* 1936), Branch Mgr, York Ice Machinery Corp, 2400 Carson St, Pittsburgh, and (for mail), 3114 Wainbell Ave, Dormont, Pittsburgh, Pa

WEGMANN, Albert (*M* 1918), 6206 North 17th St, Philadelphia, Pa

WEIL, Martin (*A* 1925), Vice-Pres (for mail), Weil-McLain Co, 641 W Lake St., and 1259 Hazel Ave, Chicago, Ill

WEIL, Maurice I. (*A* 1928), Pres (for mail), Chicago Pump Co, 2336 Wolfram St, and 1409 W Elmdale Ave, Chicago, Ill

WEIMER, Fred G. (*A* 1919), Sales, Kewanee Boiler Corp, 1711 W St Paul Ave, and (for mail), 3958 N Stowell Ave, Milwaukee, Wis

WEINFELD, Charles (*J* 1936), Sales Engr (for mail), Ilg Electric Ventilating Co, 415 Brainard St, and 107 Winona St, Detroit, Mich

WEINSHANK, Theodore* (*Life Member, M* 1906), (Board of Governors, 1913), Consulting Engr, 3307 Belden Ave, Chicago, Ill.

WEISS, Arthur P. (*M* 1928), Burnham Boiler Corp, Irvington, and (for mail), 131 Farrington Ave, North Tarrytown, N Y

WEISS, Carl A (*M* 1930; *A* 1921), Supt (for mail), Kornbrodt Kornice Co., 1811-15 Troost Ave, and 29 East 68th St., Kansas City, Mo

WEITZEL, Cameron B (*M* 1930), Owner and Operator, 122 E High St, Manheim, Pa

WEITZEL, Paul H (*J* 1936; *S* 1934), Junior Engr, Cameron B. Weitzel, 122 E High St, Manheim, Pa

WELCH, Louis A., Jr. (*A* 1929), 443 Second St, Schenectady, N Y

WELDY, Lloyd O. (*M* 1930), Branch Mgr. (for mail), Powers Regulator Co, 2341 Carnegie Ave, Cleveland, and 19258 Malvern Ave, Rocky River, Ohio

WELSH, Harvey A. (*A* 1936), Service Supv Engr, Hudson Air Conditioning Corp, 1517 Connecticut Ave, Washington, D C, and (for mail), 4118 Lee Highway, Cherrydale, Va

WELTER, M. A. (*A* 1925), Engr (for mail), Twin City Furnace Co, 410-12 W Lake St, and 4306 Garfield S., Minneapolis, Minn

WENDT, Edgar F. (*M* 1918), Pres (for mail), Buffalo Forge Co, 490 Broadway, and 120 Lincoln Parkway, Buffalo, N Y.

WENDT, Edwin H. (*J* 1936), Estimator and Draftsman (for mail), Wendt & Crone Co, 2124 Southport Ave, and 3809 N Troy St, Chicago, Ill.

WERNER, Richard K. (*M* 1930), Consulting Mech Engr (for mail), 316 W T Waggoner Bldg, and 3671 Monticello Drive, Fort Worth, Texas

WEST, Perry* (*M* 1911), (Council, 1920-1925, Treas., 1924-1925), Consulting Engr. and Prof (for mail), University of Kentucky, College of Engrg., and 185 E. Maxwell St, Lexington, Ky.

WESTOVER, Wendell (*M* 1936), Pres (for mail), Westover-Wolfe, Inc, 170 Washington Ave, Albany, and 1350 Wendell Ave., Schenectady, N. Y.

WETZELL, Horace E. (*M* 1934), Chief Engr. (for mail), Smith & Oby Co., 6107 Carnegie Ave., and 8700 Elsmere Drive, Cleveland, Ohio.

WHALLON, Fletcher (*J* 1936; *S* 1935), 3852 Lyndale Ave S, Minneapolis, Minn

WHELLER, Harry S. (*M* 1916), Vice-Pres, L J. Wing Mfg. Co, 154 West 14th St., New York, N Y., and (for mail), 725 Union Ave, Elizabeth, N. J.

WHITE, Eugene B. (*M* 1934), Arch. and Engr. (for mail), Y M C A., 19 S LaSalle St, Chicago, and 300 N. Taylor Ave, Oak Park, Ill

WHITE, Everett A. (*M* 1921), Engrg Dept, Crane Co, 30 South 16th St., and (for mail), 5244 Nottingham St., St. Louis, Mo.

WHITE, Elwood S. (*M* 1921), Pres (for mail), U. S Radiator Corp., 1056 National Bank Bldg, Detroit, Mich, and Meadowbank Rd, Old Greenwich, Conn.

WHITE, Harry S. (*A* 1936), Mgr Sheet Metal Dept (for mail), Sellers & Marquis Roofing Co, 2201 Broadway, and 20 W. Dartmouth Rd, Kansas City, Mo

WHITE, John C (*M* 1932), State Power Plant Engr, Wisconsin Bureau of Engineering, Power Plant Div (for mail), 624 E Main St, and 622 E Main St, Madison, Wis

WHITE, William R. (*A* 1936), Air Cond Engr. (for mail), Nebraska Power Co, 17th and Harney St, and 4339 Larimore Ave, Omaha, Nebr.

WHITELAW, H. Leigh (*M* 1916), Vice-Pres (for mail), American Gas Products Corp, 40 West 40th St, New York, N Y, and Overbrook Lane, Darien, Conn

WHITELEY, Stockett M. (*M* 1933), Consulting Engr (for mail), Baltimore Life Bldg, and 3931 Canterbury Rd, Baltimore, Md

WHITMER, Robert P. (*M* 1935), Secy (for mail), American Foundry & Furnace Co, McClun and Washington Sts, and 1402 E Washington St., Bloomington, Ill

WHITNEY, C. W. (*M* 1935), Pres, ABC Oil Burner & Engineering Co, 2012-14 Chestnut St., Philadelphia, and (for mail), Apt F-3, Sevilla Court, Bala-Cynwyd, Pa

WHITT, Sidney A. (*J* 1937), Air Cond. and Refrg Engr (for mail), Kelvinator Corp, 14250 Plymouth Rd, and 11950 Ohio Ave, Detroit, Mich

WHITTAKER, Wayne K. (*A* 1935), Bldg. Maintenance Co, 1 Wall St, New York, and (for mail), 221-14–114th Ave, St Albans, L I, N Y.

WHITTALL, Ernest T. (*A* 1933), Vice-Pres, May Oil Burner of Canada, Ltd, 17 Elm St, and (for mail), 11 Cottingham Rd, Toronto, Ont, Canada.

WHITTEN, Horace E. (*M* 1924), Pres and Treas, H E Whitten Co, 9 Federal Court, Boston, and (for mail), 56 Highland Rd, Somerville, Mass.

WHITTINGTON, James A. (*M* 1936), Utilization Testing Engr (for mail), Peoples Gas Light & Coke Co, 3921 S Wabash Ave, Chicago, and 022 Sheridan Square, Evanston, Ill

WIEGNER, Henry B. (*M* 1919), Branch Mgr., Johnson Service Co, 20 Winchester St, Boston, and (for mail), 143 Standish Rd, Watertown, Mass.

WIERENGA, Peter O. (*A* 1931), 231 Brown St S E, Grand Rapids, Mich

WIERIMAN, William J. (*J* 1930), Machinist (for mail), Kearney & Trecker, 1329 South 71st St., West Allis, Wis, and R F D Box 132, Aurora, Minn.

WIGGINS, Oswald J. (*J* 1935; *S* 1933), Chemist, Midwest Oil Co, and (for mail), 2708 East 37th St, Minneapolis, Minn

WIGGS, G. Lorne (*M* 1936, *A* 1932; *J* 1924), Consulting Engr. (for mail), University Tower, and 4797 Grosvenor Ave, Montreal, Que, Canada

WIGLE, Bruce M. (*A* 1926), Pres (for mail), Bruce Wigle Plumbing & Heating Co, 9117 Hamilton Ave, and 18114 Oak Drive, Detroit, Mich.

WILDER, Edward L. (*M* 1915), Mgr, Gas Sales (for mail), Utility Management Corp, 150 Broadway, New York, and 12 Mereland Rd., New Rochelle, N Y

WILEY, Donald C. (*J* 1936), Sales Engr. (for mail), John J Nesbitt, Inc, 205 W Wacker Drive, and 451 Wrightwood, Chicago, Ill.

WILHELM, Joseph E. (*J* 1936, *S* 1934), Office Engr, Avery Engineering Co, 2341 Carnegie Ave, and (for mail), 1355 West 87th St, Cleveland, Ohio

WILKINSON, Arthur (*A* 1936), Mgr (for mail), Wilkinson Engineering Agencies, 1253 McGill College Ave., and 1469 Drummond St, Montreal, Que, Canada

WILKINSON, Farley J. (*M* 1933), Mgr Engrg Service, Montgomery Ward & Co, and (for mail), 18257 Martin Ave, Homewood, Ill

WILLARD, Arthur C.* (*M* 1914), (*Presidential Member*), (Pres, 1928, 1st Vice-Pres, 1927, 2nd Vice-Pres, 1926, Council, 1925-1929), (for mail), President, University of Illinois, and 711 Florida Ave, Urbana, Ill

WILLER, Murray D. (*S* 1936), 10713 Drexel Ave, Cleveland, Ohio.

WILLEY, Earl C. (*M* 1934), Mech Engrg. Instructor, Oregon State College, and (for mail), 1652 "A" St, Corvallis, Oregon

WILLIAMS, Allen W. (*A* 1915), Managing Director (for mail), National Warm Air Heating & Air Conditioning Association, 50 W Broad St, Columbus, and 51 Meadow Park, Bexley, Ohio.

WILLIAMS, Clinton R. (*M* 1936), Draftsman, E I duPont deNemours Co, Wilmington, Del, and (for mail), 155 Primos Ave, Folcroft, Pa

WILLIAMS, Frank H. (*J* 1934), Test Engr (Air, Cond), Frigidaire Div of General Motors, Taylor St., and (for mail), 118 Maplewood Ave, Dayton, Ohio

WILLIAMS, Gordon S. (*S* 1936), Lab Asst, Dept of Mech Engrg, Yale University (for mail), 400 Temple St, New Haven, Conn, and 88-02-193rd St, Hollis, L I, N Y

WILLIAMS, J. McFarland, Jr. (4 1928, *J* 1927), Sales Engr, 1407-35th St N W, Washington, D C.

WILLIAMS, J. Walter (*M* 1915), Pres (for mail), Forest City Plumbing Co, 332 E State St, and 923 E State St, Ithaca, N Y

WILLIS, Leonard L. (*J* 1936, *S* 1935), Engr, Conrad Refrigeration Co, 17 E Hennepin Ave, and (for mail), 5036 Lyndale S, Minneapolis, Minn.

WILMOT, Charles S. (*M* 1919), (for mail), Building Insulation Co, 106 South 16th St, Philadelphia, and 406 Essex Ave, Narberth, Pa

WILSON, Alexander (*M* 1936), Consulting Engr (for mail), 315 New Birks Bldg, and 3750 Cote des Neiges Rd, Montreal, Canada

WILSON, Andrew (*M* 1935), Htg and Estimating Engr, Montgomery Ward & Co, 150th St, and Jamaica Ave, Jamaica, L I, and (for mail), 5523 Seventh Ave, Brooklyn, N Y

WILSON, Eric D. (*M* 1936), Air Cond Engr (for mail), Carrier Co, Ltd, 24 Buckingham Gate, London, S W 1, and Gorof House, Ystradgynlais, Swansea, England

WILSON, George T. (*M* 1925), Sales Engr, Gurney Foundry Co, Ltd, 4 Junction Rd., Toronto, and (for mail), Tyre Ave, Islington, Ont, Canada

WILSON, James W. (*J* 1936; *S* 1935), Route 1, Box 132 A, Irving, Texas

WILSON, Raymond W. (*M* 1934), Member of Firm (for mail), Wilson-Brinker Co, 412 Pythian Bldg, and 429 Creston Ave, Kalamazoo, Mich.

WILSON, Robert A. (*M* 1936), Sales Engr, Minneapolis-Honeywell Regulator Co, 4501 Prospect Ave, Cleveland, and (for mail), 1520 Grace Ave, Lakewood, Ohio

WILSON, W. H. (*A* 1932), Steamfitter Foreman, Pullman-Standard Car Mfg Co, 11001 Cottage Grove Ave, and (for mail), 22 West 110th Place, Chicago, Ill

WILSON, William H. (*A* 1923), Branch Mgr., Johnson Service Co, 507 E. Michigan St, and (for mail), 2023 E Olive St., Milwaukee, Wis.

WILTBERGER, Constant F. (*M* 1935), Mech. Engr, Stewart A. Jellett Co, 1200 Locust St, and (for mail), 2650 N. Ninth St., Philadelphia, Pa

WINANS, Glen D. (*M* 1929), Engr of Steam Distribution (for mail), Detroit Edison Co, 2000 Second Ave, and 16183 Wisconsin Ave., Detroit, Mich

WINQUIST, Walter J. (*A* 1930), Htg and Vtg Engr, 294 Nostrand Ave, Brooklyn, N Y

WINSLOW, C.-E. A.* (*M* 1932), Prof of Public Health (for mail), Yale University, 310 Cedar St, and 314 Prospect St, New Haven, Conn

WINTERBOTTOM, Ralph F. (*M* 1923), Engr, Winterbottom Supply Co, and (for mail), 100 Campbell Ave, Waterloo, Iowa

WINTERER, Frank C. (*M* 1920), Sales Mgr (for mail), Cochran Sargent Co, Broadway and Kellogg Blvd, and 836 Juno St, St Paul, Minn.

WINTHER, Anker (*A* 1936, *J* 1932), Air Cond Engr, York Ice Machinery Corp, 2116 Gilbert Ave, and 3620 Stettinius Ave, Cincinnati, Ohio

WISE, Daniel E. (*S* 1931), 400 Temple St, New Haven, Conn

WISSING, Clement B. (*A* 1936), Secy and Sales Mgr (for mail), Ebner Ice & Cold Storage Co, Locust and Chestnut Sts, and 702 N. Sixth St, Vincennes, Ind

WITHERIDGE, David E. (*J* 1930), Engr, W A Witheridge Co, 746 S Fourth Ave, Saginaw, Mich

WITMER, Charles N. (*J* 1930), Dist Dealer Supv (for mail), Carrier Corp, 2022 Bryan St, and 4154½ Prescott St, Dallas, Texas

WOESE, Carl F. (*M* 1931), Consulting Engr (for mail), Robson & Woese, Inc, 1001 Burnet Ave, and 256 Robineau Rd, Syracuse, N Y

WOHL, Maurice W. (*M* 1931), 100 Linden Blvd, Brooklyn, N. Y.

WOLF, John C. (*M* 1923), Htg and Vtg Engr. (for mail), B F Sturtevant Co, and 76 Beacon St, Hyde Park, Mass

WOLF, Philip (*M* 1935), City Contracting Co, 2025 Fifth Ave, New York, N Y

WOLFF, Peter P. (*M* 1935), Engr, Bell & Gossett Co, 3000 Wallace St, and (for mail), 7250 Champlain Ave, Chicago, Ill

WOOD, Frederick C. (*A* 1936, *J* 1931), Sales Engr, Air Cond (for mail), Airtemp, Inc, Div. Chrysler Motors, 8021 Conant Rd, and 16220 Normandy, Detroit, Mich

WOODMAN, Lawrence E. (*M* 1931), Pres (for mail), Woodman Appliance & Engineering Corp., 203 E Capitol, and 1011 Fairmount, Jefferson City, Mo

WOODS, Edward H. (*M* 1931), Prop (for mail), F H Higgins, 311 E. State St, and Hook Place, Ithaca, N Y.

WOOLLARD, Mason S. (*M* 1934), Draftsman, H H Angus, Consulting Engr, 1221 Bay St, and (for mail), 31 Hillcrest Park Ave, Toronto, Ont, Canada

WOOLLEY, J. Herbert (*A* 1936), Vice-Pres. (for mail), Woolley Coal Co, Inc, 12 Burnett Ave, Maplewood, and 75 Oakview Terrace, Short Hills, N J

WOOLSTON, A. H. (*M* 1919), Woolston-Woods Co., 2132 Cherry St, Philadelphia, Pa

WORLD, Harry P. (*M* 1936), Structural and Mech Designer, Col Mackenzie Waters, Archt, 96 Bloor St W., and (for mail), 102 Geoffrey St., Toronto, Ont, Canada

WORSHAM, Herman (*M* 1925; *J* 1918), National Business (for mail), Delco-Frigidaire Conditioning Division, 1420 Wisconsin Blvd, and 519 W. Norman Ave, Dayton, Ohio.

WORTHING, Stanley L. (*M* 1930), Consulting Engr. (for mail), 433 Kelsey Bldg., Grand Rapids, and Spring Lake, Mich.

WRIGHT, Clarence E (*J* 1935, *S* 1933), Htg Engr, Fairmont Wall Plaster Co., and (for mail), 506 Ogden Ave, Fairmont, W Va

WRIGHT, Harris H. (*M* 1917), Mfrs. Repr (for mail), 320 E Tenth St., and 808 Greenway Terrace, Kansas City, Mo.

WRIGHT, Kenneth A. (*M* 1921), Branch Mgr (for mail), Johnson Service Co, 1113 Race St, Cincinnati, Ohio, and 113 Orchard Rd, Ft Mitchell, Covington, Ky

WRIGHT, M Birney (*A* 1932; *J* 1929), Mech Engr (for mail), E I duPont deNemours & Co, P O Box 1537, and Cedar Ave, S Hills, Charleston, W Va

WRIGHT, William J. (*J* 1936, *S* 1935), 1114 W. Illinois St, Urbana, Ill

WUNDERLICH, Milton S * (*M* 1925), Chairman Research and Test Committee, Insulite Co, 1100 Builders Exchange, Minneapolis, and (for mail), 515 Mt Curve Blvd., St Paul, Minn

WYATT, DeWitt H (*M* 1936), Consulting Engr., 226 Northridge Rd, Columbus, Ohio

WYLIE, Howard M. (*M* 1925, *J* 1917), Vice-Pres in charge of Sales (for mail), Nash Engineering Co., and 51 Elmwood Ave., South Norwalk, Conn

Y

YAGER, John J. (*M* 1921), 425 Woodbridge Ave, Buffalo, N Y

YAGLOU, Constantin P.* (*M* 1923), Asst Prof of Industrial Hygiene (for mail), Harvard School of Public Health, 55 Shattuck St., Boston, and 10 Vernon Rd, Belmont, Mass

YATES, George L. (*J* 1936; *S* 1934), Instructor, Dept Oil & Gas Prod, University of Pittsburgh, Pittsburgh, Pa, and (for mail), 1220 Johnstone, Bartlesville, Okla.

YATES, James E. (*M* 1934), Mgr (for mail), Yates-Neale & Co, 231 Tenth St, and 431-16th St, Brandon, Man, Canada

YATES, James E, Jr. (*J* 1936), Service and Installation Mgr (for mail), Automatic Heating Co., Ltd, 307½ Fort St, and 503 River Ave, Winnipeg, Man., Canada

YATES, Walter (*Life Member, M* 1902), Governing Director (for mail), Matthews & Yates, Ltd, Cyclone Works, and Parksend, Swinton, Man, England

YOUNG, Emil O. (*A* 1935), Pres. (for mail), Young Ventilating Co, 2703 Woodland Ave., and 2040 East 83rd St, Cleveland, Ohio

YOUNG, Forest H., Jr. (*A* 1936), Mgr, Young Heat Engineering Co, 116 North 26th St, Billings, Mont

YOUNG, J. T, Jr. (*A* 1936), Sales Engr (for mail), Crane Co, Box 1410 (307 W. Second So), and 237 D St, Salt Lake City, Utah

Z

ZACK, Hans J (*M* 1928), Pres, Zack Co, 2311 Van Buren St, Chicago, Ill

ZANGRILLI, Albert J. (*S* 1935), 537 Turrett St, Pittsburgh, Pa

ZIBOLD, Carl E. (*M* 1929), Mech Engr, Htg. and Vtg, 13 Chadwick Rd, Westminster Ridge, White Plains, N Y

ZIEBER, William E. (*M* 1935), Asst. Chief Engr. (for mail), York Ice Machinery Corp, Roosevelt Ave, and 112 S Penn St, York, Pa.

ZIESSE, Karl L. (*A* 1931), Secy-Treas (for mail), Phoenix Sprinkler & Heating Co, 115 Campau Ave N W, and 315 Hampton Ave S E, Grand Rapids, Mich

ZIMMERMAN, Alexander H (*A* 1930), Ventilation Engr, Chicago Board of Health, Randolph and LaSalle St, and (for mail), 5449 N St Louis Ave, Chicago, Ill

ZINK, David D (*M* 1931), Consulting Engr (for mail), 320 West 47th St, Kansas City, and Hickman Mills, Mo

ZOKELT, Carl G. (*M* 1921), Consulting Engr., 3810-24th Ave S, Seattle, Wash

ZUHLKE, William R (*M* 1928), Exec Engr, American Radiator Co, 40 West 40th St, New York, and (for mail), 54 Midland Ave, Yonkers, N Y

ZWALLY, August L. (*A* 1937), Air Cond Engr., Interstate Electric Co, 300 Spring St, and (for mail), 908 Elmwood, Shreveport, La

Summary of Membership

(Corrected to January 1, 1937)

UNITED STATES AND ISLAND TERRITORIES

Alabama	3	Nebraska	6
Arizona	2	New Hampshire	1
Arkansas	4	New Jersey	106
California	49	New York	381
Colorado	7	North Carolina	12
Connecticut	30	North Dakota	1
Delaware	9	Ohio	161
District of Columbia	61	Oklahoma	35
Florida	11	Oregon	2
Georgia	16	Pennsylvania	237
Hawaii	1	Phillipine Islands	2
Illinois	253	Puerto Rico	1
Indiana	22	Rhode Island	5
Iowa	14	South Carolina	2
Kansas	9	South Dakota	2
Kentucky	13	Tennessee	9
Louisiana	14	Texas	42
Maine	4	Utah	2
Maryland	30	Vermont	3
Massachusetts	102	Virginia	13
Michigan	120	Washington	26
Minnesota	119	West Virginia	9
Missouri	109	Wisconsin	66
Montana	4		2130

Canada ... 157

FOREIGN COUNTRIES

Australia	4	Mexico	2
Belgium	1	Manchouko	1
Brazil	1	New Zealand	3
China	13	Norway	2
Czechoslovakia	1	Palestine	1
Denmark	1	Scotland	1
England	21	South Africa	2
France	8	Spain	2
Germany	2	Sweden	3
India	1	Trinidad	1
Ireland	1	Turkey	1
Italy	4	U. S. S. R.	1
Japan	5	Venezuela	1
Java	1		85
		Total Membership	2372

SUMMARY OF MEMBERSHIP BY GRADES

Honorary Members	3
Presidential Members	25
Members	1356
Associate Members	540
Junior Members	357
Student Members	91
	2372

LIST OF MEMBERS

Geographically Arranged

UNITED STATES
and
ISLAND TERRITORIES

ALABAMA

Birmingham
Fried, H V
Lichty, C P
Murphree, R L

ARIZONA

Phoenix
Edwards, H. B
Keys, L P

ARKANSAS

Benton
McCoy, C E.

Fort Smith
Herrick, L.

Pine Bluff
Greer, W. R

Siloam Springs
Jones, C R.

CALIFORNIA

Glendale
Moon, F. L.
Stormes, R. M.

Hollywood
Blumenthal, M. I.
Hungerford, L.

Los Angeles
Anderson, C. S.
Berglund, N. W.
Binford, W. M.
Bullock, H. H
Cawby, E. I.
Conrad, R.
Cooper, A. W.
Cranston, W. E., Jr.
Dickinson, T.
Douglas, H. H.
Ellingwood, E. L.
English, H.
Freeman, J. C.
Hendrickson, H. M.
Hess, A. J.
Hill, F. M.
Hogue, W. M.
Kilpatrick, W. S.
Koolstra, J. F.
Lauer, H. B.
Nelson, E. L.
Ness, W. H. C.
Orear, A G.
Ott, O. W.
Park, I F
Phillips, R E
Pierce, F D
Polderman, L H
Scofield, P C
Wallace, K S

Oakland
Cummings, G J
Means, L A.

Pasadena -
Gifford, R. L

San Diego -
Sadler, C B

Sacremento -
Kennedy, M

San Francisco
Boney, A. J.
Cochran, L H
Corrao, J
Haley, H. S
Hudson, R. A.
Krueger, J I
Leland, W. E.

Sausalito
Howe, W W

South Gate
Barnum, W. E., Jr.

South Pasadena
Warren, H. L.

COLORADO

Colorado Springs -
Jardine, D. C.

Denver -
Davis, A. F.
Lindberg, A. F.
McQuaid, D. J.
O'Rear, L R.
Ward, O. G.

Fort Collins -
Cutler, J. M.

CONNECTICUT

Bridgeport —
Smak, J. R.

Fairfield —
Osborn, W. J.

Glenbrook—
Wahrenbrock, O K

Greenwich—
Jones, A. L
Opperman, E F

New Haven—
Blakeley, H. J
Hughes, C E
Rodee, E J
Seeley, L E.
Teasdale, L A
Williams, G S
Winslow, C -E A
Wise, D E.

New London—
Chapin, C. G.
Forsberg, W.
Hopson, W T.

South Norwalk—
Adams H E.
Harvey, A D
Jennings, I C
Lyons, C. J.
Mead, E A
Wylie, H. M.

Stamford —
Hoyt, L. W.

Torrington—
Doster, A.

Wallingford —
Burns, J R.

Waterbury—
Davis, O E
Lincoln, R L
Simpson, W K
Stewart, C. W.

West Hartford —
Corey, G. R.

DELAWARE

Wilmington—
Belt, N. O.
Gawthrop, F. H.
Hayman, A. E., Jr
Kershaw, M G.
Lownsbery, B F.
Ponsell, F. I
Robinson, G. L.
Schoenjahn, R P
Weatherlow, G P

DISTRICT OF COLUMBIA

Washington—
Bennett, C A
Bensinger, M
Brown, W A.
Casey, H F
Connell, H
Cover, R R
Cullen, A G.
Dalla Valle, J M
Danielson, W A
Day, I M
DeWitt, E S
Downes, H H
Eagleton, S P
Ensman, P H
Febrey, E J
Feltwell, R H
Fife, G D
Fineran, E V.
Fisher, J T
France, C N
Frankel, G S
Gardner, S F
Goddard, W. F.
Gregg, S L.
Grimes, F M
Hartline, W R
Hawk, J K
Holmes, P B
Hood, O P
Iverson, H R
Kajuk, A E
Keplinger, W L.
Kicrales, M D
Kingswell, W E.
Liebrecht, W J
Littleford, W. H.
Lloyd, E. H
Lockhart, W R
Loughran, P H
Loving, W H.
Mayette, C. E.
McDonald, A K
Miller, G F
Mullen, T J, Jr
Nest, R. E
Nordine, L. F
Nye, L B, Jr
Ourusoff, L. S.
Riley, E C.
Ritt, C F
Robinson, D. M.
Smallman, E W.
Stokes, A D
Sutter, E E
Thomas, G
Thompson, N S.
Thuney, F M
Tuxhorn, D B.
Urdahl, T H
Vaughan, T G., Jr
Williams, J. M Jr.

FLORIDA

Jacksonville—
Allen, W. W
Thornton, W B
Varner, J. L

Lakeland—
Mozley, R G

Miami—
Burke, W J
Lingo, C K
Munro, E A
Pizie, S G

Miami Beach—
Friedman, D. H, Jr

Orlando—
Porter, C W.

West Palm Beach—
Hodeaux, W L

GEORGIA

Atlanta—
Baker, C T
Beechler, J S
Clare, F W.
Foss, E R
Gouedy, K E
Hahn, R F
Kagey, I B, Jr
Kent, L F
Klein, E W.
Kuempel, L. L
McKinney, W. J.
Moler, W H
Sudderth, L, Jr
Templin, C L.
Tucker, T T.

Augusta—
Arndt, H. W.

HAWAII

Honolulu—
Petersen, S. E.

ILLINOIS

Alton—
Carlock, M. F
Giles, A F

Berwyn—
Brightly, F C, Jr

Bloomington—
MaGirl, W J
Nesmith, O E.
Soper, H A.
Whitmer, R P

Champaign—
Hintz, H P
Logan, T M
Ransom, C F
Strakosh, W C.
Thetford, J E

Chicago—
Adams, B P
Aeberly, J J
Arenberg, M K.
Baker, W H, Jr.
Bamond, M J.
Baumgardner, C M.
Baur, J W
Becker, W A
Beery, C E
Bernstrom, B
Bevington, C H
Bishop, M W
Black, F C.
Boehmer, A P
Bolte, E E
Borling, J R
Boyle, J. R
Bracken, J H
Braun, L T
Brigham, C M
Broom, B A
Brown, A P
Brown, T
Casey, B L
Chapin, H G
Christman, W. F
Christophersen, A E
Clegg, R R.
Close, P D
Crone, C E, Jr
Cunningham, J. S
Cunningham, T. M
Cutler, J A.
Deane, J H
DeLand, C W.
Doherty, R
Dunham, C A
Emmert, L D.
Ericsson, E B.
Evleth, E B.
Fatz, J L
Ficnerski, P
Finan, J J.
Fleming, J P
Fleming, T F
Foster, T R.
Frank, J M
Gardner, W, Jr.
Gaylord, F H
Getschow, R M
Gibbs, F C.
Goelz, A H
Gossett, E J
Gothard, W W
Gotschall, H C
Graham, E W.
Graves, W. B
Haas, S L
Haines, J J
Hale, J F
Hanley, T F, Jr
Hart, H M
Hattis, R E
Hayden, C F
Hayes, J J
Hayward, R. B.
Hebley, H F J.
Herlihy, J J
Hess, D K
Hill, E. V
Hinckley, H B
Hornung, J. C.
Horton, H. F.
Howatt, J.
Howell, L
Hubbard, G W.
Hustoel, A M
Isett, W M
Jenson, J. S
Johns, H B.
Johnson, C. W
Kaufman, C W
Keeney, F P.
Kehm, H S
King, A C
Kreissl, H G.
Krez, L
Kyle, W J
Lagodzinski, H. J.
LaRoi, G. H
Larson, C. P.
Lautenschlager, F.
Lauterbach, H, Jr
Lees, H K
Lewis, S R
Lockhart, H A
Luders, R H
Machen, J T
Malone, D G
Malvin, R C
Manny, J H
Marschall, P. J
Martin, A B
Matchett, J C
Mathis, E
Mathis, H
Mathis, J W
May, M F
McCauley, J H.
McClellan, J E
McDonnell, E N
McDonnell, J E
McIlvaine, J H.
McMahon, J E
Mertz, W A
Miller, F A
Miller, R T
Milliken, J H
Milliken, V D
Moore, C H
Morse, L S, Jr
Mueller, H C
Murphy, E T
Narowetz, L L, Jr
Neiler, S G
Newport, C F
Nightingale, G F
Offen, B
Olsen, C F
Olson B
Paul, L O
Peller, L
Pitcher, L J
Pope, S A
Powers, F W
Prentice, O J
Price, C E
Priester, G B
Rasmussen, R P
Raymond, F. I
Reger, H P
Reid, H P
Reis, L S
Rietz, E W.
Rottmayer, S I.
Runkel, C.
Russell, E A
Scheidecker, D. B.
Schermer, R
Schuetz, C C.
Schwartz, H.
Schweim, H J
Seelig, L
Schilling, H C
Shultz, E
Snider, L A.
Sommerfield, S S
Spielmann, G P
Stacy, L D.
Stannard, J M
Stermer, C J.
Sutcliffe, A. G
Swanson, N W
Thinn, C A
Thomas, L. G L
Thomas, R H
Thommen, A A
Tobin, J F.
Tornquist, E L
Trumbo, S M
Turner, G G
Van Alsburg, J H.
Vernon, J R.
Vinson, N L
Walters, W T.
Washington, L W
Weil, M
Weil, M. I
Weinshank, T
Wendt, E H
White, E B
Whittington, J A
Wiley, D C
Wilson, W H
Wolff, P P
Zack, H J
Zimmerman, A H

Cicero—
Brown, N A
Keppner, H W

Decatur—
Shorb, W A.

Elmhurst—
Jones, D J

Evanston—
Maccubbin, H A
Miller, J E
Moore, R E

Glen Ellyn—
Parsons, L D
Sherman, V L

Homewood—
Wilkinson, F J

Kewanee—
Bronson, C E.
Dickson, R B
Hartman, J M
Pursell, H. E

LaGrange—
Eaton, B K
Linn, H R

Lake Forest—
Scudder, B

Moline—
Beling, E H
Johnson, W G
Nelson, H W
Nelson, R H
Otis, G. E

Mt. Vernon—
Benoist, L L
Benoist, R E.

Oak Park—
Blanding, G H.
Fitzgerald, M J.
May, E. M.
Uhlhorn, W J

Park Ridge—
Heckel, E P
Spielmann, H J

Peoria—
Baird, S A.
Farnsworth, J. G
Fox, E. L.
Meyer, F. L.

Rockford—
Braatz, C. J
Dewey, R. P
Merwin, G E.
Pruden, B
Sharp, H C.
Stewart, D. J.

Urbana—
Bowditch, R. P.
Broderick, E. L
Compton, W. E
Fahnestock, M K
Faulkner, G.
Gallagher, P

Konzo, S
Kratz, A P
Severns, W H
Willard, A C
Wright, W J.

Villa Park
Armspach, O W

Western Springs
Washington, G

Winnetka
Mittendorf, E M

Zion
Baughman, L R

INDIANA

Evansville
Bullett, C R

Huntington
Redrup, W D
Smith, G W

Indianapolis
Ammerman, C. R.
Fenstermaker, S F.
Hagedon, C H
Hayes, J G.
Hildreth, E. S
Ott, R C.
Poehner, R F
Supple, G B

LaPorte
Shrock, J H.
Truitt, G S.

Michigan City
Stockwell, W R

Peru
Thrush, H. A

St. Mary-of-the-Woods
Beach, B J

Vincennes
Waring, C B

Wabash
Shivers, P E.

West Lafayette
Chenoweth, D M.
Hoffman, J D
Moore, D R
Prudden, O. D.

IOWA

Ackley
Nelson, G O.

Ames
Stiles, G S

Cedar Falls
Clay, C. H.

Cedar Rapids
Pappentun, W. G

Davenport
Walsh, E. R , Jr.

Des Moines—
D'Imor, E J.
Shaw, B E.
Vaughn, F. R.
Walters, A. L.

Forest City—
Rossiter, P A

Sioux City—
Hagan, W. V

Waterloo—
Gill, J W
Todd, M L
Winterbottom, R. F

KANSAS

Hutchinson—
Mann, A R
Stevens, H L

Lawrence—
Machin, D W
Sluss, A H

Neodesha
Berzelius, C E

Russell—
Danielson, E. B
Danielson, L C

Salina
Ryan, W. F.

Wichita—
Leverance, H J

KENTUCKY

Gold Spring
Ward, E J

Ft Thomas—
Reik, R C.
Schick, K W
Stevens, W. R

Lexington—
Bozeman, R W
May, J W
O'Bannon, L S
Wallace, W M H
West, P.

Louisville
Fitch, H M.
Hellstrom, J.
Murphy, H C

South Fort Mitchell—
Kennedy, O. A

LOUISIANA

Chaudrant—
Rinehart, W R

New Orleans
Blum, H., Jr.
Gamble, C. B.
Gammill, O. E., Jr.
May, G E
McLaren, F S.
Moses, W. B., Jr
Ryan, J D.
Salzer, A. R , Jr.
Schwartz, S B.
Seidel, G. E.

Shreveport—
Cheatwood, W. H.
Fitzgerald, W. E.
Zwally, A. L

MAINE

Lewiston—
Fowles, H H

Portland—
Fels, A B
Merrill, C J
Mitchell, C H

MARYLAND

Annapolis—
Gale, H A.

Baltimore—
Axeman, J E.
Collier, W I
Crosby, E L
Hall, M S
Hunt, M.
Leilich, R L
McCormack, D
Page, V C.
Posey, J
Powers, E C.
Seiter, J. E
Shepard, J deB
Sklarevski, R
Smoot, T H
Steele, M G
Stehl, H V
Vance, L G
Viessman, W
Vincent, P J
Whiteley, S M

Bethesda—
Goodwin, E W
Stock, E S
Terhune, R D

Brooklyn Park—
Rodgers, J S.

College Park—
Gifford, W R

Ellicott City—
Tibbets, J C

Rockville—
Brunett, A L

Roland Park—
Dorsey, F C

Silver Spring—
Stack, A E

MASSACHUSETTS

Arlington—
Shaw, N J. H

Arlington Heights—
Tarr, H M.

Boston—
Archer, D. M.
Baker, R H
Berchtold, E W.
Boyden, D S.
Brinton, J W
Brissette, L. A.
Brocha, J. F.
Bryant, A G
Bullock, T A
Cummings, C H.
Dickson, G. P
Donohoe, J. B
Drinker, P
Edwards, D J.
Foulds, P A L.
Franklin, R S
Gleason, G H.
Hajek, W J.
Hilliard, C E
Jennings, W G
Keefe, E T.
Kelley, J J
Kellogg, A
Kimball, C W
McCoy, T F.
Merrill, F. A
Millard, J W.
Miller, J F G
Nee, R M
Osborne, M M
Palmer, R T
Plunkett, J H
Rydell, C. A
Smith, R H
Stetson, L R
Swaney, C R.
Turner, J
Tuttle, J F.
Waterman, J H
Yaglou, C P

Cambridge—
Flint, C T
Gerrish, G B
Haddock, I T.
Holt, J W
Hoyt, C W
MacDonald, E A.
Moore, H C
Peterson, C M F
Prince, R F.

Cochituate—
Ahearn, W. J

Dalton—
Dakin, H W

Dorchester—
Goodrich, C F
Hosterman, C O
Shaer, I. E
Voye, V J

Fall River—
Fenner, E M

Fitchburg—
Earley, T J
Illig, W. R
Karlson, A F
McKittrick, P A.

Harwich Port—
Maxwell, G W

Holyoke—
Colby, C W

Hyde Park—
Bartlett, A C.
Ellis, F R
Fritzberg, L H
Wolf, J C

Lawrence—
Bride, W T.

Leominster—
Kern, R T.

Lynn—
Feehan, J B
Oates, W A.

Mattapan—
Ahlberg, H B

Needham—
Park, C D
Webb, J S

Newton Center—
Murray, J J

Newtonville—
Emerson, R R
Jones, W T
McMurrer, L J

Pittsfield—
Wagner, E A

Quincy—
Stone, E R

Reading—
Ingalls, F D B

Roslindale—
Larson, C W

Sharon—
Nelson, A W

Somerville—
Whitten, H E

Springfield—
Cross, R E
Holmes, R E.
Leland, W B
Murphy, W. W
O'Neil, J M

Swampscott—
Knowles, M G

Watertown—
Weigner, H. B

Wellesley Hills—
Barnes, W E
Gilling, W F, Jr

West Roxbury—
Christie, A Y
McPherson, W A

Weymouth—
Clough, L

Winchester—
Brigham, F H

Woburn—
Parker, P

Wollaston—
Hodgdon, H A

Worchester—
Robinson, H C
Wechsberg, O

MICHIGAN

Ann Arbor—
Backus, T H L
Emswiler, J E
Mann, A

Battle Creek—
Christenson, H

Birmingham—
Hadjisky, J N
Root, E B
Volberding, L A.

Dearborn—
King, H K.

Detroit—
Akers, G W
Arnoldy, W F
Baker, I C
Baldwin, W H
Barth, H E
Bishop, F R
Blackmore, F H
Boales, W G
Booth, H N
Brennan, J W
Brown, W. A
Burch, L A
Clark, E H
Connell, R F
Coon, T E
Cummins, G H
Darlington, A P
Dauch, E O
DeBoos, F A
Dickenson, F R
Dubry, E E
Eckert, E K
Epple, A B
Estep, L G
Feely, F J
Feinberg, E
Ford, E F.
Giguere, G H
Hamaker, A C
Hamlin, H. A.
Harrigan, E M.
Heydon, C G
Hill, H H
Hogan, E L
Hutzel, H F
Jelinek, F. R
Keyser, H M
Kilner, J. S
Kincaide, M C
Kirkpatrick, A H.
Knibb, A E
Linsenmeyer, F J
Long, D J
Luty, D J
Maier, G M
McConachie, L L.
McCrea, J B
McIntire, J. F
McLean, D
Miller, R E
Milward, R. K
Morse, C T
O'Gorman, J S, Jr
Paetz, H E
Parrott, L G
Partlan, J W
Patterson, F. H.
Pike, W H.
Purcell, F C
Purcell, R. E.
Randall, W C.
Rickner, C A
Rowe, W A
Sanford, S. S
Sauer, R L
Schechter, J. P
Shea, M B
Snell, E
Snyder, J W
Spitzley, R L
Spurgeon, J H
Taylor, H J.
Taylor, W E
Toonder, C L.
Tuttle, G H.
Walker, J H
Wallace, J B.
Weinfeld, C.
White, E S
Whitt, S. A
Wigle, B M
Winans, G D
Wood, F. C.

Dowagiac—
Firestone, J F
Torr, T W

E. Lansing—
Miller, L G

Grand Rapids—
Bradfield, W W
Morton, C H
Todd, S W, Jr
Troske, J J
Warren, F C
Wierenga, P O
Worthing, S L
Ziesse, K L

Houghton—
Seeber, R R

Kalamazoo—
Brinker, H A
Downs, S. H
McConner, C R
Schlichting, W G
Temple, W J
Wilson, R W

Lansing—
Adams, E I
Chrouch, R B
Haas, R B
McLouth, B. F
Parsons, R A
Vanderlip, P J

Mt. Clemens—
Bailey, E P, Jr.

Muskegon Heights—
Reid, H F

Port Huron—
Blessed, W A

Royal Oak—
Helmrich, G B

Saginaw—
Witheridge, D E.

MINNESOTA

Bayport—
Swanson, E C

Duluth—
Foster, C
Maynard, H R

Hibbing—
Miller, L L

Little Falls—
Haatvedt, S. R.

Minneapolis—
Aiken, J. F
Aikman, J M.
Algren, A B
Anderson, S H
Armstrong, R W.
Bell, E F.
Bensen, C L.
Betts, H. M.
Bredesen, B. P.
Buot, A. V.
Burns, E J
Burritt, C G.
Butts, R. L
Cash, T. T.
Cooper, T E
Copperud, E R
Cumming, F J
Dahlstrom, G A
Davidson, J C.
Dovolis, N J
Edelman, B P
Fergestad, M L
Fortai, D M
Gelb, A
Gerrish, H E
Gordon, E B, Jr
Gordon, W J, Jr
Gross, L C
Hall, J R
Hanson, L P
Harris, J B
Hawkinson, C F
Helstrom, H G
Herman, N B
Hildebrandt, H A
Hitchcock, P C
Huch, A J
Johnson, L H
Jordan, L E
King, R L
Kingsland, G D
Knapp, D S
Kuehn, W C.
Kuns, J W
Kurtz, R W
Lange, F F
Legler, F W.
Lilja, O L.
Lund, C. E
Martenis, J V
Miller, H A
Mills, H. C
Mitchell, J G
Mjolsnes, L. O
Morgan, G C
Morton, H. S
Myers, C R.
Ogard, N L
Orr, G M
Ostrin, A
Pfeifer, O J, Jr
Powell, K A.
Pung, D W
Roberts, H P.
Roberts, J. R
Rowley, F B
Sanford, A L
Schernbeck, F H.
Schultz, A. W.
Seelert, E H
Spencer, J B.
Stiller, F W
Sturm, W.
Sundell, S S.
Sutherland, D. L
Swanson, R. G
Sweiven, C E
Swenson, J E
Tabor, C. B.
Uhl, E. J
Uhl, W. F
Van Horn, H. T
Wallace, H. P, Jr
Welter, M A
Whallon, F.
Wiggins, O J
Willis, L. L.

Owatonna—
Clarkson, W. B

Robbinsdale—
Sweatt, C H

Rochester—
Adams, N. D

St. Louis Park—
Hopkins, F. L.

St. Paul—
Anderson, D B
Ayers, E H
Backstrom, R E
Barnum, C R
Bauer, A E
Bean, G S
Collins, M D
Estes, E C
Fitts, C D
Gausman, C E
Hickey, D W
Hyde, L L
Jones, E F
McNamara, W
Oberg, H C.
Ruff, D C
Solstad, L L
Swanstrom, A. E
Veltman, B M
Winterer, F. C
Wunderlich, M S

Wayzata —
Heberling, C W.

Winona —
Hamerski, F D.

MISSOURI

Clayton—
DuBois, L J.

Ferguson —
Szombathy, L R

Jefferson City—
Woodman, L. E

Kansas City —
Adams, C W
Allan, N J
Allen, D. M
Anspacher, T H
Arthur, J. M
Ball, W
Barnes, A F
Barnes, H P.
Betz, H. D
Bliss, G. L
Caleb, D
Cameron, W. R
Campbell, E K
Campbell, E K, Jr.
Cassell, W. L.
Chase, L. R
Clegg, C
Cook, B F.
Dawson, T. L
Dean, F J, Jr
Dodds, F F
Downes, N. W
Farber, L. M.
Fehlig, J B
Flarsheim, C. A
Forslund, O. A.
Garnett, R. E.
Gilham, W E
Gould, H. E.
Haas, E., Jr.
Harbordt, O. E.
Kell, W. R.
Kitchen, J. H.
Maillard, A L.
Matthews, J. E.
Millis, L. W.
Natkin, B
Nottberg, G
Nottberg, H
Olchoff, M
Pellmounter, T.
Pexton, F S.
Pines, S.
Rivard, M M
Robb, J E
Russell, W A
Sawyer, J N
Scarlett, W J
Scherrer, K C
Sheppard, F A.
Stephenson, L A
Stevens, K M
Weiss, C A
White, H S
Wright, H H
Zink, D D

Normandy—
Dulle, W L

St Joseph—
Flynn, F J
Harton, A J

St. Louis—
Barry, J G, Jr
Bayse, H V
Boester, C. F
Bradley, E P
Carlson, E E
Carter, J H
Chapman, W A
Cooper, J W
Corrigan, J A
Davis, C R.
Edwards, D. F.
Eskin, S G.
Fagin, D J
Foster, J M
Gilmore, L A
Grossmann, H A
Haller, A. L
Hamig, L L
Hamilton, J E.
Hartwein, C E
Hester, T J
Hugomot, V E
Kent, J K
Langenberg, E B
Laufketter, F. C
Malone, J S
Manahan, J E
McLauney, H W
McMahon, T W
Moon, L W
Nelson, C L.
O'Brien, W, N
Reed, P L
Rodenheiser, G B.
Rosebrough, R M
Scherrer, L B
Sodemann, P. W
Sodemann, W C B
Stammer, E L
Sydow, L J
Tenkonohy, R J
White, E. A

Springfield—
James, R E.
Karchmer, J H

University City—
Falvey, J D

Webster Groves—
Fillo, F. B.
Myers, G W. F.

MONTANA

Big Timber—
Strickland, A. W.

Billings —
Cohagen, C C.
Young, F. H., Jr.

Great Falls—
Ginn, T. M

NEBRASKA

Clarks—
Manning, W M

Lincoln—
Prawl, F E
Stevenson, M J
Stokes, A

Omaha—
Olson, G E
White W R

NEW HAMPSHIRE

Manchester—
Meakin, J. B

NEW JERSEY

Arlington—
Adler, A. A
Bock, B A

Asbury Park—
Strevell, R P.

Atlantic City—
Strouse, S B

Bayonne —
Schwartz, J

Bloomfield —
Clericuzio, G P
Faust, F H
Harrington, E
McLenegan, D W
Tenney, D

Camden—
Brown, W. M,
Coward, C. W
Kappel, G W A
Lanning, E K
Plum, L H
Rohlin, K W
Webster, E K
Webster, W
Webster, W, Jr

Cliffside Park -
Butler, P D

Collingswood—
Bolsinger, R C
Mohrfeld, H. H.

Cranford—
Johnson, L O

East Orange—
Atkins, T J
Ferguson, R R
Gombers, H B
Grahn, V F.
Rack, E C
Reilly, J. H.
Tallmadge, W.
Turno, W G W.

Elizabeth—
Bentz, H.
Cherne, R E
Cornwall, G. I
Grant, W A
Kaczenski, C.
Lyman, S E
Wheller, H. S.

Essex Fells—
Stacey, A. E, Jr.

Freehold—
Buck, D. T

Haddonfield—
Dobbs, C E

Hasbrouck Heights—
Goodwin, S L.

Hawthorne—
Spoelstra, W J

Irvington—
Reinke, A G
Stengel, F J

Jersey City—
Hashagen, J B
Jones, H L
Kelly, C J
O'Rourke, H D, Jr
Ritchie, W
Walterthum, J J

Lyndhurst—
Ehrlich, M W

Maplewood—
Kepler, D A
Kylberg, V C
Woolley, J. H.

Merchantville—
Binder, C G.

Newark—
Alt, H. L.
Ashley, C M
Bryant, P J.
Carey, P C
Carner, W. H
Day, V S
Erickson, E. V.
French, D
Hill, C F
Hoehl, E R
Holbrook, F. M
Holton, J H
Ingels, M
Leinroth, J P.
Lewis, L L
Lyle, J I
Macrow, L
Mann, L B
Morehouse, H P
Ray, L. B
Raymer, W. F, Jr
Soule, L. C
Steinmetz, C W. A

North Arlington—
Bermel, A H
Vogelbach, O

Orange—
Crawford, J H, Jr

Paterson—
Bannon, L E
Cox, H F
Pryor, F L

Perth Amboy—
Simkin, M

Plainfield —
Hedges, H B.
Tobin, G J

Princeton—
Dean, M. H.

Ridgefield Park—
Davis, A C.

Ridgewood—
Fitts, J C

River Edge—
Mawby, P

Rochelle Park—
Emery, G W.

Roselle Park—
Kampish, N. S

Somerville—
Van Nuys, J C

South Orange—
Browne, A L.
Hansen, C C.
Lewis, T.

Summit—
Oaks, O O
Taliaferro, R R

Teaneck—
Heebner, W. M.

Union City—
Taverna, F. F.

Upper Montclair—
Smith, F J.

Verona—
Shotwell, R. W

Westfield—
Scribner, E. D.

West Orange—
Adlam, T N

West New York—
Stinard, R L

NEW YORK

Albany—
Bond, H. A.
Dick, A. V
Johnson, H S.
Murray, T. F
Nelson, A W
Ryan, H J
Taggart, R C
Teeling, G A.
Westover, W

Bedford Hills—
Waters, F. A.

Binghamton—
Marum, O.

Buffalo—
Beman, M C
Booth, C A
Cherry, L A
Cheyney, C C
Criqui, A A
Currier, C H.
Davis, J
Day, H C.
Drake, G M
Farnham, R
Farrar, C W.
Gifford, C A
Hamlin, C J, Jr.
Harding, L. A
Heath, W R.
Hedley, P S.
Hexamer, H D
Hirschman, W F.
Jackson, M S
Kaiser, F.
Kamman, A. R
Landers, J J
Lenihan, W O
Lighthart, C H
Love, C H
Madison, R D.
Mahoney, D J
McKinley, C B
McTernan, F J
Messenger, T I
Mosher, C H
Rente, H W
Roebuck, W, Jr
Seelbach, H
Shelney, T.
Smith, M S
Snyder, J S
Voisinet, W E
Walker, E R
Wendt, E F
Yager, J J.

Derby—
Ensign, W A

Elmira—
Davis, B C

Geneva—
Herendeen, F W.

Hamburg—
Graham, C H
Dolson, C N.

Hastings-on-Hudson—
Reynolds, T W

Hudson Falls—
Hollister, E W

Irvington-on-Hudson—
Bastedo, A. E

Ithaca—
Barns, A A.
Elwood, W H
Sawdon, W M
Williams, J. W
Woods, E H

Kendall—
Stangland, B F

Kenmore—
Candee, B C.
Mollenberg, H J.
Quigley, W. J

Larchmont—
Downe, E. R
Gaylor, W S

Lockport—
Bishop, C. R
Saunders, L. P.

Mt. Vernon—
Freitag, F. G.
Northon, L
Obert, C W

New Rochelle—
Abrams, A.
Farley, W F
Giannini, M C.
Lambert, R. D
Rose, H J
Terry, M C

New York City—
Addams, H.
Adler, J C
(Forest Hills, L I)
Ames, C F.
Anderson, G A M.
Andresen, G C
(Brooklyn)
Apt, S R
(Flushing, L I)
Ashley, E E
Atherton, G R
Baker, H L, Jr
Balsam, C. P
(Brooklyn)
Barbieri, P J
Barnum, M C
Baum, A L.
Beebe, F E W.
Bennett, E A
Bennitt, G E
Berman, L K.
Bernhard, G
(Brooklyn)
Bianculli, V A
Blackburn, E C., Jr
(Garden City, L. I.)
Blackman, A O
Blackmore, J J.
Blackshaw, J L
(Brooklyn)
Bloom, L
(Brooklyn)
Bodinger, J H
Bolton, R P
Bonthron, R C
Bowles, P
Brabbee, C W
Broome, J H
(Brooklyn)
Brown, D
Buensod, A C
Burbaum, W A
Burr, K
Buttaravoli, F
(Brooklyn)
Callahan, P J
(Great Kills, S I.)
Campbell, F. B
Campbell, R E
(Coney Island)
Carpenter, R H
(Brooklyn)
Charles, T. J
Charlet, L W.
Chase, C. L
Cross, F G
Cucci, V J.
Dailey, J A
(Astoria, L I.)
Daly, R E.
Darts, J A
Davison, R L
Deely, J J
(Brooklyn)
Denise, J R.
Denny, H R.
Dodge, H A
Donnelly, R
Dornheim, G. A.
(Bronxville)
Downs, C R.
Driscoll, W H
Duff, K
Duncan, J R
Dwyer, T F
(Brooklyn)
Eadie, J G
Elliott, L.
Engle, A.
Etlinger, J M
Everetts, J, Jr
Faile, E H
Farbman, L X
Fay, D P
(Richmond Hill, L I)
Fay, F C
Feldman, A M
Fenner, N P
Fidelius, W R
Fiedler, H W
Fleisher, W L
Flink, C H
Frank, O. E
Frey, G O
Friedman, M.
Frimet, M
(Tompkinsville, S I)
Fritz, C. V.
(Freeport)
Galloway, J F
(Brooklyn)
Gates, R A
(Brooklyn)
Gilmour, A B
(Brooklyn)
Glore, E F
Goldberg, M.
(Brooklyn)
Goldschmidt, O E
Gordon, P B
Gornston, M H
(Brooklyn)
Goulding, W
(Brooklyn)
Graber, E.
(Astoria, L I)
Greben, D
Green, A W
(Jackson Heights., L. I.)
Greenberg, L.
Groves, S A.
(Forest Hills)
Hament, L
Hamje, M C.
Hartman, F. S.
Harsch, R J
(Brooklyn)
Hateau, W M.
Heibel, W E
Henry, A S, Jr.
Hering, A.
Herkimer, H
Herske, A. R
Herty, F. B.
(Brooklyn)
Hicks, H K.
Hiers, C R.
(Great Neck, L I)
Hildreth, L W.
Hinkle, E C
(Hempstead, L I.)
Hinrichsen, A F.
Hoffman, C S.
Hollister, N A.
(Brooklyn)
Hosking, H L
Hotchkiss, C. H. B.
Howell, F B
Hyman, W. M.
d'Issertelle, H. G.
Jacobus, D S.
James, J. W.
Janet, H. L
Jarcho, M. D
Johnson, E. B.
(W. New Brighton, S. I)

Johnson, W A
Johnston, W H
Jordan, W D
Josephson, S
(Brooklyn)
Kastner, G. C
Kessler, J
Kimball, D D
Knopf, C
(Brooklyn)
Koehler, C S
Kubasta, R. W.
Kuhlmann, R
Lane, D D
(Jackson Heights)
L I)
Leupold, H. W
Tottenville,
S I)
Leventhal, B
(Brooklyn)
Lewis, J
Lavar, A P
Lucke, C E
Lyle, E. T
Lyon, P S
Lyons, M. A
Mandeville, E W.
(Brooklyn)
Markush, E U.
Martin, G W
Matzen, H B
(Rockville Centre,
L I)
McClintock, W.
McCloughan, C
(Brooklyn)
McEwan, E E
McGaughey, J E
McKiever, W H
McLeish, W S.
Mehne, C A
Menke, H G.
Meisel, C L.
Merle, A
(Jacksonville, L I)
Meyer, H C, Jr
Milener, E. D.
Miller, C A.
Miller, J
Miller, L B.
Montgomery, O. C.
Moore, R E
(Brooklyn)
Morse, F W
Moss, E
(Brooklyn)
Munier, L. L.
Munson, J. L.
(Valley Stream)
Murphy, C. G
Murphy, J. R
Mytinger, K. L.
Neale, L. I.
Offner, A J
Oldes, W. E.
Olsen, G E.
(Arverne, L I)
Olson, R G
Olvany, W. J
(Woodhaven, L. I)
Pabst, C S
Patorno, S A. S
Penna, A. E
Port Richmond,
S. I.)
Pfuhler, J. L.
(W New Brighton,
S I)
Phillips, F W., Jr
(Brooklyn)
Pietsch, J. A
(New Brighton,
S. I.)
Pihlman, A. A.
Place, C R.

Pohle, K F
Price, E H
(Riverhead, L I)
Purdy, R B
Purinton, D J
Quirk, C H
Raisler, R K
Ramsay, J W
Raynis, T
(Richmond Hill,
L I)
Reynolds, W V
Riley, C L
Riley, R C
(Jamaica, L I)
Ritchie, E J
(Brooklyn)
Ritter, A
Rodman, R W
Rose, A A
Rosenberg, P
Rosenburg, W E
(Locust Valley,
L I)
Ross, J O
Roth, C F.
Ruggles, R. F.
(Randall Manor,
S I)
Salter, E H
Sanbern, E N
(Rockville Centre,
L I)
Sawhill, R V
Schneider, W G
Schoepflin, P H
Schulze, B H.
Scott, G M
Seelig, A E
Sellman, N, T.
Senior, R L
Shepard, E C
Siebs, C I
Simpson, A M
Skidmore, J G
(Long Island City)
Sklenarik, L
Steeneck, K C
(Bellaire, L I.)
Steinberg, E
Sterne, C M.
(Long Island City)
Still, F. R.
Stravitsch, J J
(Ozone Park, L I.)
Strock, C.
Strunin, J
Syska, A G.
Thomson, T N
(Huntington, L. I)
Tiltz, B. E
Timmis, W. W.
Torrance, H.
Trambauer, C W.
Tucker, F N
(Freeport, L. I.)
Tusch, W
(Brooklyn)
Tyler, R D
Vervoort, E. L
Vetlesen, G U
Vivarttas, E. A
(W. New Brighton
S I)
Vogt, J H.
Wachs, L J.
(Brooklyn)
Waechter, H P.
(Tompkinsville,
S. I)
Walker, W. K
Wallace, G. J
(E Elmhurst, L I)
Walsh, M.
Walton, C. W, Jr
Waring, J M S.

Whitelaw, H L
Whittaker, W K.
(St Albans, L I.)
Wilder, E L
Wilson, A.
(Brooklyn)
Winquist, W. J
(Brooklyn)
Wohl, M W
(Brooklyn)
Wolf, P

North Tonawanda—
Conaty, B M

Oriskany—
Oakey, W E

Patchoque—
Blakeslee, D
Jalonack, I G

Pelham Manor—
Crone, T E
Peacock, J K

Rochester—
Betlem, H T
Coe, R T
Cook, R P.
Eschenbach, S P
Hakes, L M
Hewett, J B
Hutchins, W. H
Lee, R T.
Leonard, L W
Sheldon, N E
Stacy, S C
Vidale, R.

Rome—
Lynch, W L.

Scarsdale—
Cumming, R W.
Ullman, H G
Wagner, F H, Jr.

Schenectady—
Hunziker, C E
Seely, I. R
Vogel, A
Welch, L. A., Jr.

Schodack Landing—
Freas, R B

Snyder—
John, V P.

Syracuse—
Acheson, A R
Evans, E C
Hockensmith, F E
Rudio, H M.
Woese, C F.

Tarrytown—
Weiss, A P

Tonawanda—
Karlsteen, G. H

Utica—
Knapp, J. H.
Steinhorst, T F.

White Plains—
Durkee, M E
Zibold, C. E

Yonkers—
Bense, W M
Goerg, B.
Harmonay, W. L.
Hayter, B.
Hopp, H K.

Kelly, J G
Rainger, W F
Zuhlke, W R

NORTH CAROLINA

Burlington—
Kistler, M L

Charlotte—
Brandt, E H
Hodge, W B
Small, B R

Greensboro—
Harding, E R

High Point—
Gray, W E

Raleigh—
Rice, R. B.

Winston-Salem—
Bahnson, F F
Brown, M D
Cornwall, C C
Page, A
Rueger, C A, Jr

NORTH DAKOTA

Grand Forks—
Pesterfield, C H

OHIO

Akron—
Gault, G W

Bellefontaine—
Quay, D M

Cincinnati—
Bird, C
Coombe, J
Doyle, W J
Edwards, A W
Floyd, M
Grabensteder, L
Graham, J M.
Green, W C
Helburn, I. B
Houlis, L D
Houliston, G B
Hudepohl, L F
Hust, C. E
Junker, W H
Kiefer, C J
Kinney, A M
Kitchell, H N
Kramig, R E, Jr.
Lautz, F A.
Little, K. B.
Mills, C A.
Pillen, H A
Pistler, W C
Powers, L. G
Richard, E J
Royer, E B.
Ruff, A. G.
Sigmund, R W.
Smith, J. A.
Smith, S
Sproull, H. E.
Washburn, M. J.
Winther, A
Wright, K A.

Cleveland—
Allmen, N S
Andes, W
Avery, L T
Baker, R H
Barney, W E
Beach, W R
Borkat, P
Brezina, E A
Brueggeman, A R
Cary, E B
Cohen, H
Cohen, P
Conner, R M
Curtis, H. F
Curtis, W A
Davis, J R
Eveleth, C F
Friedman, A A.
Geissbuhler, J O
Geltz, R W
Gottwald, C
Graham, W D
Gray, E W
Harris, J G
Harvey, L C
Kain, E. M
Kalinsky, A G
Kartone, V T
Kitchen, F A
Klie, W
Leslie, D E
Levy, M I
Marotta, J A
Martinka, P D
McKeeman, C A
Miles, J C
Miner, F K
Moeller, R
Morris, F H.
Pogalies, L H
Rather, M. F
Repko, J J
Rhoton, W R
Rowe, W M
Schmidt, R. H
Schurman, J A, Jr
Sennet, L E
Smith, W D
Southmayd, R T.
Taze, D L
Tuve, G L.
Vanderhoof, A. L.
Walker, J E.
Webb, E C
Wetzell, H. E
Wilhelm, J E
Young, E. O.

Cleveland Heights—
Davis, R G.
Richmond, J.
Rodgers, F A
Sterling, J G.

Columbus—
Allonier, H R
Breneman, R. B
Brown, A I
Kimmell, P M.
Poling, D B
Sherman, R A
Williams, A. W.
Wyatt, D H.

Cuyahoga Falls—
Humphrey, D E.
Jennings, H. K
Read, R. R.

Dayton—
Brooks, F W
Buenger, A.
Clarke, J G.
Gibbons, M J
Gonzalez, R A
Hull, H B
LaSalvia, J J
Lewis, C E
MacMillan, A R
Noble, J P
Shields, C D
Smith, L E
Swift, P F
Williams, F H
Worsham, H

East Cleveland—
Nobis, H M
Stark, W E
Steffner, E F

Elyria—
Maynard, J E

Hudson—
Follett, T L

Lakewood—
Fuller, J H
Longcoy, G B
Tanker, G E.
Teckmyer, F C, Jr.
Wilson, R A

Mansfield—
Blair, H A.

Middletown—
Byrd, T
Maier, A H
Stitt, A B.

Newark—
Simpson, D C
Slayter, G.

Norwood—
Braun, J J
Motz, O. W

Painesville—
Hobbs, J C.

Piqua—
Lange, R T

Sedalia—
Middelton, H A

Shaker Heights—
Harvey, R. A.

Springfield—
Hauck, E. L

Toledo—
Baker, H. C
Bichowsky, F R
Jones, S
Keyes, R E
McKitrick, W D
Myers, F L.
Treadway, Q.
Watkins, G B

Waverly—
Armbruster, F T.

Wilmington—
Marconett, V. G.
Sapp, C L

Youngstown—
Boucherle, H. N
Choffin, C. C.

OKLAHOMA

Alva—
Husky, S T

Bartlesville—
Yates, G L

Norman—
Adams, B C, Jr
Bowman, J W
Dawson, E F
Giles, J C
Rauh, E M
Sneed, R B
Sonney, K J

Oklahoma City—
Constant, E S
Dolan, R G
Dugger, E. R
Earl, W
Emmons, N L
Gray, E W.
Hoppe, A A
Howlett, I G
Loeffler, F X
Loeffler, L, Jr
Mayes, C
Meinholtz, H W
Middleton, D
Miller, B R
Miner, M H
Rathbun, P W.
Rolland, S. L
Tiller, L

Tulsa—
Dean, C H.
Disney, M A
Frampton, A C.
Holmes, A D
Jones, E
Jordan, R C
Pauling, R E
Shoemaker, F F

OREGON

Corvallis—
Willey, E C

Portland—
Kroeker, J D

PENNSYLVANIA

Abington—
Park, N W

Allentown—
Korn, C B

Ardmore—
Haynes, C V.

Ardsley—
Tucker, L. A.

Atglen—
Thompson, E. F

Bala-Cynwyd—
Patrick, H M
Whitney, C W.

Beaver Falls—
Van Alen, W T

Beechwood, Del. Co.—
Kipe, J M.

Bethlehem—
Stuart, M. C

Bradford—
Cleveland, C C.
Paterson, F C, Jr
Presdee, C. W
Ruple, P E
Turner, P K

Brookline—
Eastman, C B

Camp Hill—
Beitzell, A E

Chambersburg—
Kottcamp, H A

Cheltenham—
McElgin, J W

Cynwyd—
Smith, W F

Drexel Hill—
Hance, W. W

E. Pittsburgh—
Goodwin, W C.

Elizabethtown—
Dibble, S E

Elkins Park—
Buck, L

Erie—
Joyce, H B
McDowell, B W
Preece, L. W.

Etna—
Park, H. E

Folcroft—
Williams, C R

Germantown—
Mack, L
Marks, A A.
McCullough, H G.

Greensburg—
Burkhart, E. M

Harrisburg—
Eicher, H. C
Geiger, I H
Lutz, J H., Jr
Selig, E. T.

Haverford—
Black, E N, 3rd

Hershey—
Snavely, A. B

Johnstown—
Hunter, L N.
Novotney, T. A

Kingston—
MacDonald, D. B.

Lancaster—
Jones, A.
Lloyd, E C.

Lansdowne—
James, H. R.
Lauer, R F

Lemont—
Hensley, W P

Manheim—
Weitzel, C B
Weitzel, P H

McKeesport—
Dugan, T M

Middletown—
Locke, R A

Midland
Crichton, H C

Narberth—
Dever, H F

New Castle—
Andrews, G H.
McElhaney, G W.
Sonneborn, C.

New Kensington—
Edwards, J D.
Osterle, W. H.

Norristown—
Hucker, J H

Oil City—
Black, W B

Oxford—
Ware, J. H.

Philadelphia—
Adams, B
Ahlitt, A A
Bachman, F.
Barnard, M. E.
Bartlett, C E
Black, H G
Blankin, M. F.
Bogaty, H S
Bornemann, W A
Burke, J J
Caldwell, A. C
Call, J
Cassell, J D.
Clodfelter, J. L
Cody, H G.
Cornell, T C.
Culbert, W. P.
Dambly, A E
Davidson, L. C.
Davidson, P. L.
Dome, A G
Donovan, W J
Elliot, E
Engel, E
Erickson, H. H.
Faltenbacher, H. J
Familetti, A R.
Galligan, A B
Gant, H P.
Gemmill, R. A
Gillett, M. C
Gilman, F W.
Glassey, J W.
Hackett, H. B.
Hibbs, F. C.
Hochstein, G. E.
Hunger, R F
Hynes, L. P
Ickeringill, J. C
Jex, J. Jr
Jopson, J. M.
Kelble, F R
Knebel, A. E.
Leopold, C S.
MacDade, A. H.
Martocello, J. A
Mather, H H
McClintock, A., Jr.
Mellon, J T J
Mensing, F D
Meyer, J W
Moody, L E
Morgan, R C
Morris, A M
Murdoch, J P
Nesbitt, A J
Nesbitt, J J
Newcomb, L B
Nusbaum, L
Plewes, S E
Prybil, P L
Rank, A I
Redstone, A L
Reilly, C E
Rettew, H F
Rhea, C A
Roberts, H L
Rugart, K
Sabin, E R
Shanklin, A P
Sheffler, M
Skagerberg, R
Speckman, C H
Stone, G F
Timmis, P
Touton, R D
Traugott, M
Trump, C C
Tuckerman, G. E.
Wandless, F W.
Wegmann, A
Wilmot, C S
Wiltberger, C F
Woolston, A H

Pittsburgh—
Aston, J
Beighel, H A.
Blackmore, G. C
Brauer, R
Burns, R
Bushnell, C D
Carr, M L.
Cheeseman, E W
Clippinger, J. V.
Collins, J F S, Jr
Comstock, G M
DeMerit, R N
Dorfan, M I.
Edwards, P A.
Eils, L C.
Ellis, G. P
Farbman, S. M.
Frisse, J L
Goodman, D.
Greiner, G E, Jr.
Griest, K
Gunther, F A
Hecht, F H.
Heilman, R H
Houghten, F C
Huettner, H F
Humphreys, C M.
Hyde, E. H
Jacobi, B.
Kennedy, P. V.
Krintzman, H
Maehling, L. S
Maginn, P F
McGinness, J E
McGonagle, A
McGuigan, L A
McIntosh, F. C
McLean, J. E
McMunn, A. H
Miller, R A
Moore, H L.
Nass, A F.
Neis, W A.
Nicholls, P.
Noord, D F
Poucher, R. C
Proie, J.
Reed, I. G.
Reed, V. A.
Riesmeyer, E H., Jr.
Rockwell, T F
Rodgers, W C.
Scanlon, E. S
Smyers, E C
Speller, F N
Stanger, R B
Steggall, H B.
Stevenson, W. W
Strauch, P C
Tennant, R J. J.
Tower, E. S
Waters, G G
Weddell, G O.
Zangrilli, A. J

Pottsville—
Marty, E O
Smith, J D.

Primos, Del. Co.—
Johnson, A J

Reading—
Luck, A W
Nicely, J. E.

Ridley Park—
Stalb, J G.

Rose Valley—
Atkinson, K B.

Rutledge—
Jones, N R
Vroome, A E

Scranton—
Mahon, B B
Shaver, H. H

Springfield—
Grossman, H E
Payne, R E

State College—
Queer, E R.

Stroudsburg—
Kiefer, E J., Jr.

Swarthmore—
Hobbs, W S
Robinson, A. S

Upper Darby—
Aughenbaugh, H E.
Blackmore, J S
Currie, F. J

Villanova—
Barr, G. W.
Carey, J A

Wallingford—
Arnold, R S

Washington—
Frazier, J E.

Wilkinsburg—
Campbell, T F.

Wormleysburg—
Miller, T. G

York—
Hertzler, J R
Zieber, W. E.

PHILLIPINE ISLANDS

Manila—
Hausman, L. M.
Macrae, R B.

PUERTO RICO

San Juan—
Des Reis, J. F.

RHODE ISLAND

Providence—
Coleman, J B.
Gibbs, E W
Hartwell, J C.
McLaughlin, J D
Moulder, A. W

SOUTH CAROLINA

Clemson College—
Shenk, D H

Columbia—
Hartin, W. R, Jr

SOUTH DAKOTA

Lead—
Pullen, R. R

Sioux Falls—
Monick, F R.

TENNESSEE

Elizabethtown—
Torok, E

Knoxville—
Sterner, D S

Memphis—
Campbell, A Q, Jr
Flinn, G. S.
Hoshall, R H
Perkins, R C.

Murfreesboro—
Moore, H W.

Nashville—
Brown, F
Jarratt, P R.

TEXAS

Amarillo—
Burnett, E. S

Bryan—
Griesser, C E

College Station—
Badgett, W. H
Giesecke, F. E.
Long, W. E.
Smith, E G.
Teal, E. T.

Dallas—
Bock, I. I.
Boruch, E. R.
Durning, E. H
Gay, L. M.
Knibs, C L, Jr
Landauer, L L
Meffert, G H.
Mehl, O. H
Renouf, E. P.
Schucany, O. W
Stringfellow, J. C.
Taylor, R. F.
Turner, H. S, Jr.
Weatherby, E. P., Jr.
Witmer, C. N.

Fort Worth—
Shaw, C G
Skinner, H W
Werner, R K

Galveston—
Huff, J M
Warren, C W

Hereford—
Jones, A P

Houston—
Cochran, W B
Groseclose, J B
Kiesling, J A
Rowe, I E
Walsh, J A

Irving—
Wilson, J W

Kingsville—
Richtmann, W M

San Antonio—
Beard, E L
Burks, R H
Cotter, R P
Diver, M L
Ebert, W A
Monier, K A J

Somerville—
Barton, D H

UTAH

Salt Lake City—
Richardson, H G
Young, J T, Jr

VERMONT

Burlington—
Lanou, J E
Raine, J J

North Ferrisburg
Breckenridge, L. P.

VIRGINIA

Arlington—
Marshall, W D

Cherrydale—
Welsh, H A

Danville—
Farley, W S

Lynchburg—
Doering, F L

Norfolk—
Nowitzky, H S
Webster, W H, Jr

Portsmouth—
Stubbs, W C

Richmond—
Carle, W. E
Hankins, R P
Johnston, J A
Schulz, H I
Snyder, A K

University—
Peebles, J K, Jr

WASHINGTON

Kent—
Boyker, R O

Seattle—
Beggs, W E
Bouillon, L
Case, R H
Cox, W W
Daly, C P
Eastwood, E O
Granston, R O
Hauan, M J
MacLeod, K F
Mallis, W
May, C W
Morse, R D
Musgrave, M. N
Peterson, S D
Pollard, A L
Twist, C F
Ward, J J
Weber, E L
Zokelt, C G

Spokane—
Russell, W B

Tacoma—
Foote, E E
Spofforth, W

Washougal—
O'Connell, P M

Yakima—
Leichnitz, R W
McCune, B V

WEST VIRGINIA

Charleston—
Rothmann, S C
Shanklin, J A
Thompson, D
Wright, M B

Fairmont—
Tonry, R C
Wright, C E

Largent—
Donnelly, J. A

South Charleston—
Titus, M S

Wheeling—
Hitt, J C

WISCONSIN

Cuba City—
Kellner, D C

Ft. Atkinson—
Shodron, J G

Green Bay—
Haus, I J

Kohler—
Hvoslef, F W
Kohler, W J, Jr

LaCrosse—
Anderegg, R H
Miller, M W
Trane, R N

Lake Mills—
Schwantes, A R

Madison—
Dean, C L
Larson, G L
Nelson, D W
Plaenert, A B
White, J C

Milwaukee—
Berghoefer, V A
Bowers, A F
Brown, W H
Cochran, C C.
Cook, H R
Elliott, N B
Ellis, H W
Erath, E O
Frentzel, H C
Goldsmith, F W.
Gregg, S H
Hanley, E V
Hessler, L W
Hughey, T M
Jackson, C H
Jepertinger, R C
Jones, E A
Jung, J S
Juttner, O J
Knab, E A
Koch, R G
Krenz, A S
Lofte, J A
Miller, C W
Miller, H M
Mueller, H P
Noll, W F
Page, H W
Peters, H H.
Randolph, C H
Rice, C J
Schnitzer, S
Shawlin, W C
Sickert, G D
Spence, M R.
Steinkellner, E J.
Swisher, S G, Jr
Szekely, E
Trostel, O A
Volk, J H
Wagner, A M
Weimer, F. G.
Wilson, W. H.

Neenah—
Angermeyer, A H
Eiss, R. M

Racine—
Dixon, A G
Menden, P J
Thomas, N. A

Superior—
Waite, H

West Allis—
Erickson, M E.
Spence, R T.
Wierman, W. J

CANADA

Brandon, Man.—
Yates, J E

Brockville, Ont.—
Davenport, R F.

Calgary, Alberta—
Clarke, S. S

Edmonton, Alberta—
Mould, D. E

Fort Erie, N. Ont.—
Gordon, W D

Galt—
Sheldon, W D, Jr

Halifax, N. S—
Eagar, R. F

Hamilton, Ont.—
Barnes, H
Best, M W
Dickenson, M E
Maddux, O L

Islington, Ont.—
Wilson, G T

Kitchener, Ont.—
Beavers, G. R

Lindsay, Ont.—
McCrae, G W.

London, Ont.—
Dobie, T K

Montreal, P. Q.—
Ballantyne, G L
Barnsley, F. R
Darling, A B
Dewar, W G
Dixon, M. F
Dufault, F H
Dupuis, J. R
Dykes, J B
Fraser, J J
Friedman, F J
Garneau, L
Gittleson, H
Givin, A W
Hamlet, F A
Hughes, W U.
Johnson, C W
Laffoley, L H
LaMontagne, A F.
Madely, F J
Marshall, A G
Martin, L
McGrail, T E
Morris, J A
Murray, H. G S.
Nickle, A. J
Osborne, G. H
Perras, G. E.
Phipps, F G.
Robertson, J A M
Roche, I F
Ross, J D
Timmins, W. W
Wiggs, G. L
Wilkinson, A.
Wilson, A.

Montreal, W., P. Q.—
Forrester, N. J
Linton, J P

Ottawa, Ont.—
Allen, A W
Colclough, O. T
Gray, G. A
Pennock, W. B

Preston, Ont.—
Everest, R. H

Quebec—
Paquet, J M

Regina, Sask.—
Stewart, J C.

Rigaud, Que
Forgnty, O A

St. Laurent
Tollhurst, G C

Three Rivers, Que
German, O

Timmins, Ont
Smith, R J

Toronto, Ont –
Alexander, S W
Allsop, R P
Angus, H H
Anthes, L L
Arrowsmith, J O
Baker, G R
Burrell, A L
Blackhall, L C
Blackhall, W R
Boddington, W P
Cairns, J H
Carter, A W
Chambers, F W.
Church, H J
Cole, G E
Cornish, D. F
Dickey, A J
Duncan, W. A
Eaton, W G M
Ellis, F E
Fitzsimons, J P
Fox, E
Fox, J H
Gauley, E R
Goodram, W E
Gurney, E H
Gurney, E R
Harrington, C
Heard, R G
Hemon, H D
Hills, A H
Hopper, G H
Hughes, L K
Jeffrey, T G
Jenney, H B
Jennings, S A
Jones, A T
Kelly, W C
Lawlor, J J
Ledgett, F D
Leitch, A S
Libby, R S
MacDonald, D J
Marriner, J M S
McCrimmon, A M
McDonald, T
Moore, H S
Oke, W C
O'Neill, J W
Paterson, J S
Paul, D I
Playfair, G A
Price, D O
Purdy, A K
Ritchie, A G
Roth, H R
Shears, M W
Sheppard, W G F
Tasker, C
Thomas, M F
Waldon, C D
Wardell, A
Watson, M. B
Whittall, E T
Woollard, M S
World, H P

Vancouver, B. C.—
Dawson, G S
Hale, F J
Johnston, R E
Leek, W
McCreery, H J

Victoria, B. C.—
Sheret, A

Wellington, Ont —
Johnston, H D

Westmount, P. Q —
Pratt, J C

Windsor, Ont —
Hare, W A

Winnipeg, Man.—
Aitken, J
Argue, E J
Davis, G C
Eade, H R
Glass, W
Jones, B G
Kent, R L
Lochhead, K Y
Michie, D F
Miller, E R
Munn, E F
Scott, L G
Steele, J B
Summers, E T
Thompson, F
Turland, C H.
Watson, H D
Yates, J E

Woodstock, Ont.—
Karges, A

AUSTRALIA

Sydney
Hunt, N P.
Robinson, J. A
Sands, C C
Taylor, J. J

BELGIUM

Brussels
Mautsch, R

BRAZIL

Rio de Janeiro
Darby, M. H

CHINA

Nanking
Loo, P Y

Shanghai
Bradford, G G
Carter, D
Chen, S. T.
Doughty, C J
Figgis, T. G
Hart-Baker, H W
Kwan, I. K.
Loh, N S.
Merritt, C J.
Morrison, C. B.
Rachal, J. M.
Waung, T F.

CZECHOSLOVAKIA

Praha
Brust, O.

DENMARK

Copenhagen –
Reck, W. E.

DUTCH EAST INDIES

JAVA

Soerabaia – –
Thornburg, H. A.

ENGLAND

Buckinghamshire —
Russell, J N

Cheshire -
Adshead, B J

Leeds –
Jennins, H H

London
Bailey, W. M
Chester, T
Greenland, S F
Haden, G N
Herring, E
Nobbs, W W
Wilson, E. D

Middlesex
Butt, R E W
Case, W G.
Chipperfield, W H
Gill, E. F

Stockport -
Webb, J W.

Sutton -
Casperd, H. W H.

Swinton -
Yates, W

Trowbridge
Haden, W. N

Westminster
Faber, Dr O
Kraminsky, V.

Wolverhampton – –
Tyson, W. H

FRANCE

Dijon -
Bur, J R C

Lille -
Neu, H. J E

Lyon-
Goenaga, R C.

Paris
Beaurrienne, A
Downe, H S
Modiano, R.
Nessi, A
Schmutz, J.

GERMANY

Berlin—
Brandt, O H

Stuttgart—
Klein, A

INDIA

New Delhi -
Heard, J A E

IRELAND

Cork –
Barry, P. I

ITALY

Milan—
Donzelli, E
Gini, A
Hauss, C F

Torino -
Baldi, G.

JAPAN

Osaka -
Fukui, K

Tokyo—
Kitaura, S
Kozu, T
Saito, S.
Sekido, K

MANCHOUKUO

Hsinking—
Kawase, S

MEXICO

Mexico, D. F.—
Gilfrin, G F
Martinez, J J

NEW ZEALAND

Christchurch—
Taylor, E M
Vale, H. A L

Dunedin—
Davies, G W.

NORWAY

Oslo—
Alfsen, N
Tjersland, A

PALESTINE

Tel-Aviv—
Roos, E. B J

SCOTLAND

Angus—
Knox, J R

SOUTH AFRICA

Johannesburg—
Carrier, E G
Overton, S H

SPAIN

Madrid—
Alfageme, B
Jimenez, J G

SWEDEN

Lidingo—
Rosell, A, F

Stockholm—
Gille, H.
Theorell, H G T

TRINIDAD

Port of Spain—
Cox, T M. Jr.

TURKEY

Istanbul—
Karakash, T

U.S.S.R.

Lenigrad –
Sakouta, M L

VENEZUELA

Caracas—
Blas, R J

PAST OFFICERS

AMERICAN SOCIETY *of* HEATING *and* VENTILATING ENGINEERS

1894

President .... Edward P. Bates
1st Vice-President .... Wm M Mackay
2nd Vice-President .... Wiltsie F Wolfe
3rd Vice-President .... Chas S Onderdonk
Treasurer .... Judson A Goodrich
Secretary .... L H Hart

Board of Managers

Chairman, Fred P. Smith

Henry Adams — A A Cary
Hugh J Barron — James A Harding
Edward P Bates, *Pres* — L H Hart, *Secy*

Council

Chairman, R C Carpenter

Albert A Cryer — Chas. W Newton
F. W Foster — Ulysses G Scollay, *Secy*

1895

President .... Stewart A. Jellett
1st Vice-President .... Wm M. Mackay
2nd Vice-President .... Chas S Onderdonk
3rd Vice-President .... D M Quay
Treasurer .... Judson A Goodrich
Secretary .... L. H Hart

Board of Managers

Chairman, James A. Harding

Geo B Cobb — Ulysses G. Scollay
Wm McMannis — B F. Stangland
Stewart A Jellett, *Pres.* — L H Hart, *Secy*

Council

Chairman, R. C. Carpenter

Henry Adams — T. J. Waters
Edward P Bates — Albert A Cryer, *Secy*

1896

President .... R C Carpenter
1st Vice-President .... D. M. Quay
2nd Vice-President .... Edward P Bates
3rd Vice-President .... F. W Foster
Treasurer .... Judson A. Goodrich
Secretary .... L H. Hart

Board of Managers

Chairman, Wm. M Mackay

Hugh J Barron — Stewart A. Jellett
W. S. Hadaway, Jr — Wiltsie F. Wolfe
R. C. Carpenter, *Pres.* — L. H. Hart, *Secy*

Council

Chairman, A. A. Cary

Albert A. Cryer — B. F. Stangland
Wm. McMannis — J. J. Blackmore, *Secy.*

1897

President .... Wm M. Mackay
1st Vice-President .... H D Crane
2nd Vice-President .... Henry Adams
3rd Vice-President .... A. E Kenrick
Treasurer .... Judson A Goodrich
Secretary .... H M Swetland

Board of Managers

Chairman, R C Carpenter

Edward P Bates — Stewart A Jellett
W S Hadaway, Jr — Wiltsie F Wolfe
Wm M Mackay, *Pres.* — H M Swetland, *Secy*

Council

Chairman, Albert A Cryer

John A Fish — James Mackay
Wm McMannis — B. F Stangland

1898

President .... Wiltsie F. Wolfe
1st Vice-President .... J H Kinealy
2nd Vice-President .... A. E Kenrick
3rd Vice-President .... John A Fish
Treasurer .... Judson A Goodrich
Secretary .... Stewart A Jellett

Board of Managers

Chairman, Wm. M. Mackay

Thomas Barwick — A C Mott
John A. Connolly — Francis A. Williams
Wiltsie F Wolfe, *Pres* — Stewart A. Jellett, *Secy.*

Council

Chairman, R. C. Carpenter

Henry Adams — W S Hadaway, Jr.
Albert A Cryer — Wm McMannis
Wiltsie F. Wolfe, *Pres* — Stewart A. Jellett, *Secy*

1899

President .... Henry Adams
1st Vice-President .... D M. Quay
2nd Vice-President .... A. E Kenrick
3rd Vice-President .... Francis A. Williams
Treasurer .... Judson A Goodrich
Secretary .... Wm M. Mackay

Board of Managers

Chairman, Stewart A Jellett

B. H. Carpenter — Wm. Kent
A. A. Cary — Wiltsie F. Wolfe
Henry Adams, *Pres* — Wm M. Mackay, *Secy*

Council

Chairman, R. C. Carpenter

John Gormly — Wm. McMannis
W. S. Hadaway, Jr. — B. F. Stangland
Henry Adams, *Pres* — Wm M. Mackay, *Secy.*

1900

President ... D M Quay
1st Vice-President ... A E Kenrick
2nd Vice-President ... Francis A Williams
Treasurer ... Judson A Goodrich
Secretary ... Wm. M Mackay

Board of Governors

Chairman, D M Quay

Wm Kent, *Vice-Chm*
R C Carpenter
John Gormly
D M Nesbit
C B J. Snyder
Wm M Mackay, *Secy*

1901

President ... J H Kinealy
1st Vice-President ... A E Kenrick
2nd Vice-President ... Andrew Harvey
Treasurer ... Judson A Goodrich
Secretary ... Wm M Mackay

Board of Governors

Chairman, J. H Kinealy

Wm Kent, *Vice-Chm.*
R C. Carpenter
R P. Bolton
John Gormly
C B J Snyder
Wm. M Mackay, *Secy*

1902

President ... A E Kenrick
1st Vice-President ... Andrew Harvey
2nd Vice-President ... Robert C Clarkson
Treasurer ... Judson A Goodrich
Secretary ... Wm M. Mackay

Board of Governors

Chairman, A E. Kenrick

John Gormly, *Vice-Chm*
R. C. Carpenter
Wm. Kent
J H. Kinealy
C B J Snyder
Wm M. Mackay, *Secy.*

1903

President ... H. D. Crane
1st Vice-President ... Wm. Kent
2nd Vice-President ... R. P Bolton
Treasurer ... Judson A. Goodrich
Secretary ... Wm M. Mackay

Board of Governors

Chairman, H. D Crane

C. B J. Snyder, *Vice-Chm.*
R C. Carpenter
John Gormly
A. E Kenrick
Geo. Mehring
Wm M. Mackay, *Secy.*

1904

President ... Andrew Harvey
1st Vice-President ... John Gormly
2nd Vice-President ... Robert C. Clarkson
Treasurer ... Ulysses G Scollay
Secretary ... Wm. M Mackay

Board of Governors

Chairman, Andrew Harvey

John Gormly
Robert C. Clarkson
J. J Blackmore
R. C. Carpenter
H. D Crane
A. E Kenrick
C. B. J Snyder
Wm. M Mackay, *Secy.*

1905

President ... Wm Kent
1st Vice-President ... R P Bolton
2nd Vice-President ... C. B. J Snyder
Treasurer ... Ulysses G Scollay
Secretary ... Wm M. Mackay

Board of Governors

Chairman, Wm Kent

R. P. Bolton
C. B J Snyder
B. H. Carpenter
A B Franklin
James Mackay
B F Stangland
J. C F. Trachsel
Wm M Mackay, *Secy.*

1906

President ... John Gormly
1st Vice-President ... C B J Snyder
2nd Vice-President ... T. J Waters
Treasurer ... Ulysses G Scollay
Secretary ... Wm M Mackay

Board of Governors

Chairman, John Gormly

C. B J. Snyder, *Vice-Chm.*
R C Carpenter
Frank K Chew
A. B Franklin
James Mackay
B. F Stangland
T J Waters
Wm M Mackay, *Secy.*

1907

President ... C B J. Snyder
1st Vice-President ... James Mackay
2nd Vice-President ... Wm G. Snow
Treasurer ... Ulysses G. Scollay
Secretary ... Wm M Mackay

Board of Governors

Chairman, C B. J Snyder

James Mackay, *Vice-Chm.*
R. E Atkinson
R. C. Carpenter
Edmund F Capron
Frank K. Chew
A. B. Franklin
Wm. G. Snow
Wm. M Mackay, *Secy*

1908

President ... James Mackay
1st Vice-President ... Jas. D Hoffman
2nd Vice-President ... B F. Stangland
Treasurer ... Ulysses G Scollay
Secretary ... Wm. M. Mackay

Board of Governors

Chairman, James Mackay

Jas D. Hoffman, *Vice-Chm.*
B. F. Stangland
R. C Carpenter
Frank K. Chew
John F. Hale
August Kehm
C B. J Snyder
Wm M. Mackay, *Secy.*

1909

President ... Wm. G. Snow
1st Vice-President ... August Kehm
2nd Vice-President ... B. S Harrison
Treasurer ... Ulysses G. Scollay
Secretary ... Wm M. Mackay

Board of Governors

Chairman, Wm. G. Snow

August Kehm, *Vice-Chm*
John R. Allen
R. C. Carpenter
B. S. Harrison
Samuel R. Lewis
James Mackay
B. F. Stangland
Wm M. Mackay, *Secy.*

1910

President .... Jas D Hoffman
1st Vice-President .... R P. Bolton
2nd Vice-President .... Samuel R Lewis
Treasurer .... Ulysses G Scollay
Secretary .... Wm M Mackay

Board of Governors

Chairman, Jas D. Hoffman

R. P. Bolton, *Vice-Chm.*
Geo. W Barr
R. C Carpenter
Judson A Goodrich
John F Hale
Samuel R Lewis
James Mackay
Wm. M Mackay, *Secy*

1911

President .... R P. Bolton
1st Vice-President .... John R Allen
2nd Vice-President .... A B Franklin
Treasurer .... Ulysses G Scollay
Secretary .... Wm W Macon

Board of Governors

Chairman, R. P. Bolton

John R Allen, *Vice-Chm.*
John T Bradley
R C Carpenter
James H Davis
A B. Franklin
Jas D Hoffman
August Kehm
Wm W. Macon, *Secy.*

1912

President .... John R Allen
1st Vice-President .... John F. Hale
2nd Vice-President .... Edmund F. Capron
Treasurer .... James A. Donnelly
Secretary .... Wm W. Macon

Board of Governors

Chairman, John R Allen

John F. Hale, *Vice-Chm.*
Edmund F. Capron
R P Bolton
Jas D Hoffman
Dwight D Kimball
Samuel R Lewis
Wm M. Mackay
Wm. W. Macon, *Secy*

1913

President .... John F. Hale
1st Vice-President .... A. B Franklin
2nd Vice-President .... Edmund F. Capron
Treasurer .... James A. Donnelly
Secretary .... Edwin A Scott

Board of Governors

Chairman, John F. Hale

A B. Franklin, *Vice-Chm.*
John R. Allen
Edmund F Capron
R P. Bolton
Frank T Chapman
Ralph Collamore
James A. Donnelly
Dwight D Kimball
Wm. W Macon
James M. Stannard
Theodore Weinshank
Edwin A. Scott, *Secy.*

1914

President .... Samuel R. Lewis
1st Vice-President .... Edmund F Capron
2nd Vice-President .... Dwight D. Kimball
Treasurer .... James A Donnelly
Secretary .... J J. Blackmore

Council

Chairman, Samuel R Lewis

E. F. Capron, *Vice-Chm*
Dwight D. Kimball
John R. Allen
Frank T Chapman
Frank I Cooper
James A. Donnelly
John F. Hale
Harry M Hart
Frank G McCann
Wm. W Macon
James M. Stannard
J J. Blackmore, *Secy.*

1915

President .... Dwight D. Kimball
1st Vice-President .... Harry M Hart
2nd Vice-President .... Frank T Chapman
Treasurer .... Homer Addams
Secretary .... J J Blackmore

Council

Chairman, Dwight D Kimball

Harry M Hart, *Vice-Chm*
Homer Addams
Frank T Chapman
Frank I Cooper
E Vernon Hill
Wm M Kingsbury
Samuel R Lewis
Frank G. McCann
J. T J Mellon
Henry C Meyer, Jr
Arthur K Ohmes
J. J Blackmore, *Secy*

1916

President .... Harry M. Hart
1st Vice-President .... Frank T Chapman
2nd Vice-President .... Arthur K Ohmes
Treasurer .... Homer Addams
Secretary .... Casin W Obert

Council

Chairman, Harry M Hart

F T Chapman, *Vice-Chm*
Homer Addams
Charles R Bishop
Frank I Cooper
Milton W Franklin
E. Vernon Hill
Dwight D Kimball
Henry C. Meyer, Jr.
Arthur K. Ohmes
Fred R. Still
Walter S Timmis
Casin W. Obert, *Secy.*

1917

President .... J. Irvine Lyle
1st Vice-President .... Arthur K. Ohmes
2nd Vice-President .... Fred R. Still
Treasurer .... Homer Addams
Secretary .... Casin W. Obert

Council

Chairman, J. Irvine Lyle

A K. Ohmes, *Vice-Chm.*
Homer Addams
Davis S Boyden
Bert C. Davis
Milton W. Franklin
Charles A. Fuller
Harry M. Hart
E. Vernon Hill
James M Stannard
Fred R Still
Walter S. Timmis
Casin W. Obert, *Secy.*

1918

President .... Fred R. Still
1st Vice-President .... Walter S. Timmis
2nd Vice-President .... E. Vernon Hill
Treasurer .... Homer Addams
Secretary .... Casin W. Obert

Council

Chairman, Fred R. Still

W. S. Timmis, *Vice-Chm.*
Homer Addams
William H Driscoll
Howard H Fielding
H P. Gant
C W Kimball
J. Irvine Lyle
E. Vernon Hill
Frank G. Phegley
Fred. W. Powers
Champlain L. Riley
Casin W. Obert, *Secy.*

1919

President .... Walter S Timmis
1st Vice-President .... E. Vernon Hill
2nd Vice-President .... Milton W. Franklin
Treasurer .... Homer Addams
Secretary .... Casin W. Obert

Council

Chairman, Walter S. Timmis

E. Vernon Hill, *Vice-Chm.*
Homer Addams
Howard H. Fielding
Milton W Franklin
Harry E Gerrish
George B. Nichols
Frank G. Phegley
Fred. W. Powers
Robt. W. Pryor, Jr.
Champlain L. Riley
Fred R. Still
Casin W. Obert, *Secy.*

1920

President E Vernon Hill
1st Vice-President Champlain L Riley
2nd Vice-President Jay R McColl
Treasurer Homer Addams
Secretary Casin W Obert

Council

Chairman, E Vernon Hill

C L Riley, *Vice-Chm*
Homer Addams
Jos A Cutler
Wm H Driscoll
A C Edgar
Alfred Kellogg
Jay R McColl
George B Nichols
Robt W. Pryor, Jr.
W S Timmis
Perry West
Casin W Obert, *Secy*

1921

President Champlain L Riley
1st Vice-President Jay R McColl
2nd Vice-President H P. Gant
Treasurer Homer Addams
Secretary Casin W Obert

Council

Chairman, Champlain L Riley

Jay R McColl, *Vice-Chm*
Homer Addams
Jos A Cutler
Samuel E Dibble
Wm H Driscoll
H. P. Gant
E S Hallett
E Vernon Hill
Alfred Kellogg
E E McNair
Perry West
Casin W Obert, *Secy*

1922

President Jay R. McColl
1st Vice-President H P Gant
2nd Vice-President Samuel E Dibble
Treasurer Homer Addams
Secretary Casin W. Obert

Council

Chairman, Jay R. McColl

H. P. Gant, *Vice-Chm.*
Homer Addams
Jos. A. Cutler
Samuel E. Dibble
Wm H. Driscoll
E. S. Hallett
L. A. Harding
E. E McNair
H. J. Meyer
C. L Riley
Perry West
Casin W. Obert, *Secy.*

1923

President H. P Gant
1st Vice-President Homer Addams
2nd Vice-President E. E. McNair
Treasurer Wm. H. Driscoll
Secretary C W. Obert

Council

Chairman, H. P. Gant

Homer Addams, *Vice-Chm*
W. H. Carrier
J. A Cutler
S. E. Dibble
Wm. H. Driscoll
E. S Hallett
Alfred Kellogg
Thornton Lewis
E E McNair
Perry West
Casin W. Obert, *Secy*

1924

President Homer Addams
1st Vice-President S E Dibble
2nd Vice-President William H Driscoll
Treasurer Perry West
Secretary F C Houghten

Council

Chairman, Homer Addams

S. E Dibble, *Vice-Chm.*
F Paul Anderson
W H Carrier
J A Cutler
William H Driscoll
H P. Gant
W. E Gillham
L A Harding
Alfred Kellogg
Thornton Lewis
Perry West
F. C Houghten, *Secy*

1925

President S E Dibble
1st Vice-President Wm. H Driscoll
2nd Vice-President F. Paul Anderson
Treasurer Perry West
Secretary F C Houghten

Council

Chairman, S. E. Dibble

Wm H Driscoll, *Vice-Chm.*
Homer Addams
F Paul Anderson
W. H Carrier
J. A Cutler
W E Gillham
W T. Jones
Thornton Lewis
J H Walker
Perry West
A. C Willard
F C. Houghten, *Secy.*

1926

President W. H. Driscoll
1st Vice-President F Paul Anderson
2nd Vice-President A C Willard
Treasurer W E Gillham
Secretary A V Hutchinson

Council

Chairman, W. H. Driscoll

F. Paul Anderson, *Vice-Chm.*
W. H Carrier
J A Cutler
S. E Dibble
W. E Gillham
C V Haynes
W T. Jones
E B Langenberg
Thornton Lewis
J F. McInture
A. C. Willard

1927

President F. Paul Anderson
1st Vice-President A C Willard
2nd Vice-President Thornton Lewis
Treasurer W E Gillham
Secretary A. V. Hutchinson

Council

Chairman, F. Paul Anderson

A. C Willard, *Vice-Chm.*
H H. Angus
W. H Carrier
W H. Driscoll
Roswell Farnham
H H Fielding
W. E Gillham
C. V. Haynes
John Howatt
W T. Jones
J J. Kissick
E. B. Langenberg
Thornton Lewis
J F. McInture
H. Lee Moore
F B. Rowley

1928

President A. C Willard
1st Vice-President Thornton Lewis
2nd Vice-President L A. Harding
Treasurer W. E Gillham
Secretary A. V. Hutchinson

Council

Chairman, A. C. Willard

Thornton Lewis, *Vice-Chm.*
F. Paul Anderson
H H Angus
W. H. Carrier
N. W Downes
Roswell Farnham
W E. Gillham
C. V. Haynes
John Howatt
W. T Jones
J J. Kissick
E B Langenberg
J F. McIntire
H. Lee Moore
F. B. Rowley

1929

President .... Thornton Lewis
1st Vice-President .... L A Harding
2nd Vice-President .... W H Carrier
Treasurer .... W E Gillham
Secretary .... A V Hutchinson
Technical Secretary .... P D Close

Council

Chairman, Thornton Lewis

L. A Harding, *Vice-Chm*
H. H Angus
W H Carrier
N. W. Downes
Roswell Farnham
W E Gillham
C. V Haynes
John Howatt
W T Jones
E. B Langenberg
G L Larson
F C McIntosh
W A Rowe
F B Rowley
A. C Willard

1930

President .... L A Harding
1st Vice-President .... W H Carrier
2nd Vice-President .... F B. Rowley
Treasurer .... C. W Farrar
Secretary .... A. V Hutchinson
Technical Secretary .... P D Close

Council

Chairman, L A Harding

W. H. Carrier, *Vice-Chm.*
H H. Angus
D. S Boyden
R. H. Carpenter
J. D Cassell
N. W Downes
Roswell Farnham
C W. Farrar
John Howatt
W. T Jones
E B Langenberg
G L Larson
Thornton Lewis
F C. McIntosh
W A Rowe
F B. Rowley

1931

President .... W H Carrier
1st Vice-President .... F B Rowley
2nd Vice-President .... W T. Jones
Treasurer .... F. D Mensing
Secretary .... A V. Hutchinson
Technical Secretary .... P D Close

Council

Chairman, W. H Carrier

F. B. Rowley, *Vice-Chm.*
D S Boyden
E K Campbell
R. H. Carpenter
J. D Cassell
E O Eastwood
Roswell Farnham
E. H. Gurney
L A Harding
John Howatt
W. T. Jones
E. B Langenberg
G L. Larson
F. C. McIntosh
F. D Mensing
W A. Rowe

1932

President .... F. B. Rowley
1st Vice-President .... W. T Jones
2nd Vice-President .... C. V. Haynes
Treasurer .... F D. Mensing
Secretary .... A V. Hutchinson
Technical Secretary .... P. D. Close

Council

Chairman, F B. Rowley

W. T Jones, *Vice-Chm.*
D. S. Boyden
E. K. Campbell
R H Carpenter
W. H Carrier
John D Cassell
E. O. Eastwood
Roswell Farnham
F. E Giesecke
E. H. Gurney
C. V Haynes
John Howatt
G L. Larson
J F McIntire
F. D Mensing
W. E Stark

1933

President .... W T Jones
1st Vice-President .... C V. Haynes
2nd Vice-President .... John Howatt
Treasurer .... D S Boyden
Secretary .... A. V Hutchinson

Council

Chairman, W. T Jones

C V Haynes, *Vice-Chm*
D S Boyden
E K Campbell
R. H Carpenter
J D Cassell
E O Eastwood
R Farnham
F. E Giesecke
E H Gurney
John Howatt
G L. Larson
J F McIntire
F C McIntosh
L W Moon
F B Rowley
W E Stark

1934

President .... C V Haynes
1st Vice-President .... John Howatt
2nd Vice-President .... G. L. Larson
Treasurer .... D S. Boyden
Secretary .... A. V. Hutchinson

Council

Chairman, C. V Haynes

John Howatt, *Vice-Chm.*
M C Beman
D S. Boyden
Albert Buenger
R H Carpenter
J D Cassell
F E Giesecke
E H Gurney
W T Jones
G L. Larson
J F McIntire
F. C. McIntosh
L. Walter Moon
O W. Ott
W A Russell
W E Stark

1935

President .... John Howatt
1st Vice-President .... G. L. Larson
2nd Vice-President .... D. S. Boyden
Treasurer .... A J Offner
Secretary .... A. V Hutchinson

Council

Chairman, John Howatt

G. L Larson, *Vice-Chm.*
M C Beman
D S Boyden
Albert Buenger
R. H. Carpenter
J. D Cassell
F. E Giesecke
E. H. Gurney
C V. Haynes
J F. McIntire
F. C McIntosh
L Walter Moon
A. J Offner
O W Ott
W. A. Russell
W E. Stark

1936

President .... G L Larson
1st Vice-President .... D. S. Boyden
2nd Vice-President .... E H Gurney
Treasurer .... A. J. Offner
Secretary .... A. V. Hutchinson

Council

Chairman, G. L Larson

D. S. Boyden, *Vice-Chm.*
M. C. Beman
R C Bolsinger
Albert Buenger
S H Downs
W. L. Fleisher
F. E. Giesecke
E. H Gurney
John Howatt
C. M Humphreys
L Walter Moon
J. F McIntire
A. J. Offner
O. W Ott
W. A. Russell
W. E. Stark